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STUDY OF LOW GRAVITY PROPELLANT TRANSFER

FINAL REPORT

GENERAL DYNAMICS
Convair Aerospace Division

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FOREWORD

This report was prepared by the Convair Aerospace Division of General Dynamics under contract NAS8-26236, "Study of Low Gravity Propellant Transfer," for the George C. Marshall Space Flight Center. The overall contract period covered by this report is 23 June 1970 through 31 May 1972. The NASA-MSFC project manager is Mr. Hugh Campbell, S&E-ASTN-PFA.

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NOMENCLATURE

A	area
A_{cs}	cross sectional area
A_p	total paddle area in the direction of motion
a	acceleration
Bo	Bond number $= \rho \ a \ R^2/\sigma$
c	distance from neutral axis in bending
C	constant used in various calculations
CD	coupling disconnect
C_D	drag coefficient which is a function of Reynolds number for the paddle vortex system
CF	collapse factor for pressurant calculations
C_p	specific heat at constant pressure
C_1	constant used in flow equation
C_2	constant used in flow equation
D	diameter
E	modulus of elasticity
F_I	force at docking impact
F_{BL}	boost phase latch loads
F_D	total drag force
F_L	liquid fraction, V_L/V_T
F_v	ullage or vapor fraction, V_u/V_T
f	Darcy-Weisbach friction factor
g_c	gravitational constant taken as 32.2 ft/sec^2
h	specific enthalpy of fluid
h_c	pull through height for liquid outflow
h_f	heat transfer coefficient between fluid and wall
I	moment of inertia

K	radius of gyration (for solid body rotation of a sphere $K^2 = 2/5\ R^2$), coefficient for calculating head losses in fluid flow hardware
KE	kinetic energy
K_{eff}	effective thermal conductivity of high performance insulation
k	thermal conductivity, ratio of specific heat capacities
L	length
M	tank fluid mass
m	mass
$\dot{m}$	mass flow rate
n	polytropic expansion coefficient, number of screens
P	absolute pressure
$\dot{P}$	power
Q	heat transfer
$\dot{Q}$	heat transfer rate
R	radius
R_G	ideal gas constant for helium pressurant
S	ullage volume %, constant in pressurant calculations, stress
S/O	shutoff
s	distance
T	absolute temperature
t	thickness
V	volume
$\dot{V}$	volume flow rate
V_e	velocity
$(V_e)_p$	velocity of paddle taken at the outer periphery where $(V_e)_p = R_p\omega$
v	fluid specific volume
W	weight, work

Δ	differential change
λ	latent heat of vaporization
ω	angular speed
ρ	density
σ	surface tension
θ	time
$\approx$	proportional

<u>Subscripts</u>

a	allowable
b	base
BP	bubble point
C	compressor
e	environment
eq	equivalent
f	final
G	pressurant gas
He	helium
i	insulation, initial, inner
in	inlet, in
I	impact
j	jacket
L	liquid
ℓ	line
m	model
o	outlet
P	power supply
p	paddle, prototype
R	required, reservoir
s	tank surface area, saturation
T	total, turbine

TR	transferred
t	tank
u	ullage
v	vapor
w	wall

<u>Superscript</u>

o	pressurant inlet condition

SUMMARY

This report presents the results of a program to perform an analytical assessment of potential methods for replenishing the auxiliary propulsion, fuel cell and life support cryogens which may be aboard an orbiting space station. The fluids involved are cryogenic H_2, O_2, and N_2.

A complete transfer system was taken to consist of supply storage, transfer, and receiver tank fluid conditioning (pressure and temperature control). In terms of supply storage,the basic systems considered were high pressure (greater than critical), intermediate pressure (less than critical), and modular (transfer of the tanks).

The baseline resupply requirement, taken from the North American Rockwell space station concept, consists of the transfer of 1096 lb of H_2, 2480 lb of O_2 and 3150 lb of N_2 to eight H_2, two O_2 and two N_2 bottles located on the station. The standard receiver tank was 42.5 ft^3 and was designed to contain liquid at 100 psia for station use. The station life was 10 years with resupply nominally every 6 months, but with the capability for resupply every 90 days. Boiloff of fluid on the station, from environmental heating, was to be a maximum of 50 percent of LO_2 and LH_2 and 100 percent of LN_2 over a 180 day period. For redundancy in the event of meteoroid impact, half of the cryogens will be stored at one end of the station and half at the other. The maximum time allowed for the transfer operation is 24 hours and the maximum time between final supply fluid loading in the shuttle and start of transfer is seven days. Transfer line lengths from 20 to 200 feet are considered, with the nominal length being 100 ft. Maximum disturbing accelerations at the station are 10^{-4} g's.

Where applicable, both the use of individual supply tanks for each receiver and the use of a single supply for each fluid, were considered for the baseline transfer requirement. In order that other potential transfer requirements could be assessed, parametric weight and performance data were generated over a range of bottle diameters from 25 to 150 inches and fluid quantities from 500 to 5,000 lb of H_2 and 1,000 to 10,000 lb each for O_2 and N_2. The resupply of supercritical as well as subcritical receivers was analyzed.

Considerable work pertinent to the current program was performed under the Convair Aerospace 1970 and 1971 Independent Research and Development (IRAD) programs. This work is reported herein for reference only, and consisted of;

a. Analysis of high pressure or supercritical transfer to determine the feasibility of such a system for low-gravity resupply. In this system, the transfer process is analogous to the transferring of a single phase gas from one tank to another.

The major concern with the high pressure systems is that significant energy must normally be added to the supply in order to effect transfer and this energy must be removed from the receiver tank or a significant reduction in receiver fluid storage density will result. Analysis of various supply heating and blowdown and receiver cooling schemes showed that the only concept worthy of further consideration was one employing simple supply bottle heating with increased receiver volume to allow for a reduction in receiver fluid density.

b. Screening of various subcritical liquid orientation and/or collection methods to determine those most applicable to transfer of cryogenic fluids in space. On the basis of safety, weight and development potential, surface tension, metallic bellows, metallic diaphragm and paddle vortex systems were selected.

c. Detail definition and development of parametric weight data for the subcritical systems selected under (b). General heat transfer, pressurization, fluid residual and transfer line and receiver tank childown analyses were included.

Specific work under the contract consisted of;

a. Detailed analysis and development of parametric weight and performance data for the high pressure transfer concept considered to be feasible for the space station resupply application.

b. Overall system definition, analysis and development of parametric data for the modular transfer concept.

c. Weight, reliability, complexity and crew requirement comparisons and resulting recommendations for promising high pressure, subcritical and modular transfer schemes.

d. Definition of future theoretical and experimental investigations required to verify performance of the most promising systems as determined from the results of (c) above.

Significant study conclusions are presented below.

a. In the case of the high pressure supply, the main disadvantage is that energy rates required to accomplish transfer in a reasonable time are high. This is especially true for oxygen and nitrogen transfer. Also, it was found that only the supply of supercritical receivers is practical. Condensation of fluid in the receiver tank to allow storage and subsequent use of a liquid results in unreasonable power and hardware requirements. The high pressure system should not be considered further for the space station application unless station configurations with highly efficient power systems become available and transfer times approaching 24 hours are desirable.

b. For subcritical transfer the surface tension and paddle systems were determined, on the basis of low weight and cost and high reliability and reusability, to have the best potential for the present application.

A surface tension system having a double screen liner installed in a locked-up (non-vented) tank was chosen for simplicity. In this system the liner is isolated from the tank wall and wicking is relied upon to maintain the screens wetted at all times, to prevent vapor from entering the liquid outlet. Uncertainties with this system requiring further demonstration are associated with structural integrity of large size screens, cleanliness over repeated flow cycles, maintenance of proper wicking at seams, corners, supports, and outlets and overall cryogenic flow performance.

The paddle vortex system operates by creating a centrifugal acceleration on the supply liquid to maintain it at the tank outlet for transfer. For the present case an electric motor operating through a hermetically sealed flex spline was chosen to drive the paddle. Fluid residuals, however, are high for this system unless special sump designs are incorporated. Also the power requirements are uncertain and subscale model testing is needed to demonstrate confidence in the system.

c. A significant problem with respect to the prediction of overall subcritical system operation during low-g transfer is associated with transfer line and receiver tank chilldown. There are two basic methods of filling a receiver tank. One is to maintain the tank in a locked-up (no-vent) condition and design the tank to withstand any resultant pressure rise. The other is to maintain a specified maximum pressure by venting. In the case of a locked-up tank, high inflow rates and/or correspondingly low heat transfer rates are required to minimize receiver tank pressure. Even under optimum conditions it was shown that for certain tank sizes and design pressures, H_2 chilldown of a locked-up tank was not feasible. Calculations for the 42.5 ft^3 station receivers showed a non-vent fill to be questionable and use of a vent is recommended.

Where receiver tank venting is accomplished during tank chilldown, it was found that the condition of the fluid actually being vented overboard had a significant effect on the vent quantity required. In any case, fluid inflow dynamics and heat transfer at low-g must be known in order to define optimum fill methods and performance as to maximum pressure and/or quality of fluid vented. Development of a numerical technique based on the Marker-and-Cell (MAC) method is recommended for solving the cryogenic receiver liquid inflow problem. Subscale cryogenic 1-g and drop tower testing should also be performed and the data correlated with analytical models.

d. In the case of modular transfer, the main weight penalty for the individual modules is the requirement to insulate the supply for long term storage in the space station.

Even with this insulation penalty the modular systems have the lowest weight and the highest reliability when large quantities of propellant are to be transferred on a single tank to tank basis. This neglects the weight of remote manipulation systems required for modular transfer. The primary consideration in the final choice of modular versus subcritical transfer is associated with the satisfactory development of a suitable cargo handling system and the crew participation involved.

e. In all cases it was found that the fewer bottles for a given fluid transfer requirement, the lower the weight, cost and crew requirements and the higher the reliability. Safety, with respect to separation of station fluids, and redundancy requirements would determine the minimum number of bottles which could actually be used.

1

INTRODUCTION

This report presents the results of a program to perform an analytical assessment of potential methods for replenishing the auxiliary propulsion, fuel cell and life support cryogens which may be aboard an orbiting space station. The fluids involved are cryogenic H_2, O_2, and N_2. The replacement of storage bottles aboard a space station is complicated by the difficulties of performing mechanical operations in space, and transfer of fluids through lines is complicated by the absence of natural orientation of liquid and vapor in a tank. This lack of natural orientation results in the requirements for special systems to provide orientation and/or collection of the liquid to be transferred as well as receiver tank vent systems that can prevent excessive liquid loss. In this regard schemes for the transfer of the fluid in a single phase or supercritical state were also considered.

A complete transfer system for purposes of this study was taken to consist of supply storage, transfer, and receiver tank fluid conditioning (pressure and temperature control). In terms of supply storage the basic systems considered were high pressure (greater than critical), intermediate pressure (less than critical), and modular (transfer of the tanks). Detailed ground rules used to accomplish the overall program are presented in Section 2.

Considerable work pertinent to the resupply of cryogenics aboard an orbiting space system was performed under the Convair 1970 and 1971 Independent Research and Development (IRAD) programs. This work is reported herein for reference only as it compliments the present propellant transfer study. Important work under the IRAD program consisted of:

a. A screening of various intermediate pressure (subcritical) liquid orientation and/or collection methods to determine those most applicable to the transfer of cyrogenic fluids in space. Details are presented in Section 3. Also, basic tank and line weight data applicable to the overall study were developed under the IRAD program and are presented in Appendix A.

b. Analysis of various high pressure or supercritical transfer methods to determine the feasibility of such a system for low-gravity resupply. In this system the transfer process is analogous to the transferring of a single phase gas from one tank to another and the expulsion of the supply fluids is nominally accomplished by heating or blowdown. It is noted that a major concern with the high pressure systems is the fact that significant energy must, in general, be added to the supply in order to effect transfer and this energy must somehow be removed from the receiver tank or a significant reduction in receiver fluid storage density allowed for. Details of the analyses and tradeoffs performed are presented in Section 4.1. It is noted that in the case of the high pressure systems the fluid state and overall transfer performance is an integral part of the supply, transfer line and receiver tank thermodynamic and flow control processes. In order to properly determine heating, cooling and control requirements, a computer program, described in Appendix B, was developed under the Convair Aerospace 1970 IRAD program.

c. Detailed system definition, analysis and development of parametric weight data were accomplished with respect to the most promising subcritical systems as determined by the screening. Thermodynamic, heat transfer, pressurization and fluid residual calculations were included. Details are presented in Section 5. An important consideration with respect to subcritical transfer is the possible requirements for chilldown of warm transfer lines and receiver tanks. This chilldown process can be complex and computer programs for both line and tank chilldown and fill were developed and are described in Reference 1-1.

Work performed under the basic contract consisted of the following tasks.

a. Detailed analysis and development of parametric weight and performance data for high pressure transfer concepts considered to be feasible for the space station resupply application. Results are presented in Section 4.2.

b. Overall system definition, analysis and development of parametric data associated with the modular transfer concept. Details are presented in Section 6.

c. Weight, reliability, complexity and crew requirement comparisons and resulting recommendations were made with respect to the most promising high pressure, subcritical and modular transfer schemes. These data are presented in Section 7.

d. Definition of future theoretical and experimental investigations required to verify systems as determined from the results of (c.) above. This information is presented in Section 8.

The overall program conclusions are presented in Section 9.

STUDY GROUND RULES

The study ground rules were chosen to provide a scope of work such that a reasonably general investigation of low gravity fluid transfer could be accomplished within the specific requirement to replenish any cryogens which may be considered for use aboard an earth orbiting space station.

Basic space station data obtained from References 2-1 and 2-2 and which are pertinent to the present program are presented below.

A sketch of the space station, pointing out its major features, is shown in Figure 2-1. The space station is being designed for a lifetime of ten years with a circular earth orbit of 240 - 246 nautical miles at a 55° inclination angle and must be capable of independent operation for periods up to six months. Possible cryogens to be required by the space station are N_2, O_2 and H_2. Cryogens may be replenished from the shuttle docked to any of the five docking ports shown.

For redundancy in the event of meteoroid impact, half of the cryogens will be stored at one end of the space station and an equal quantity of each cryogen will be stored at the

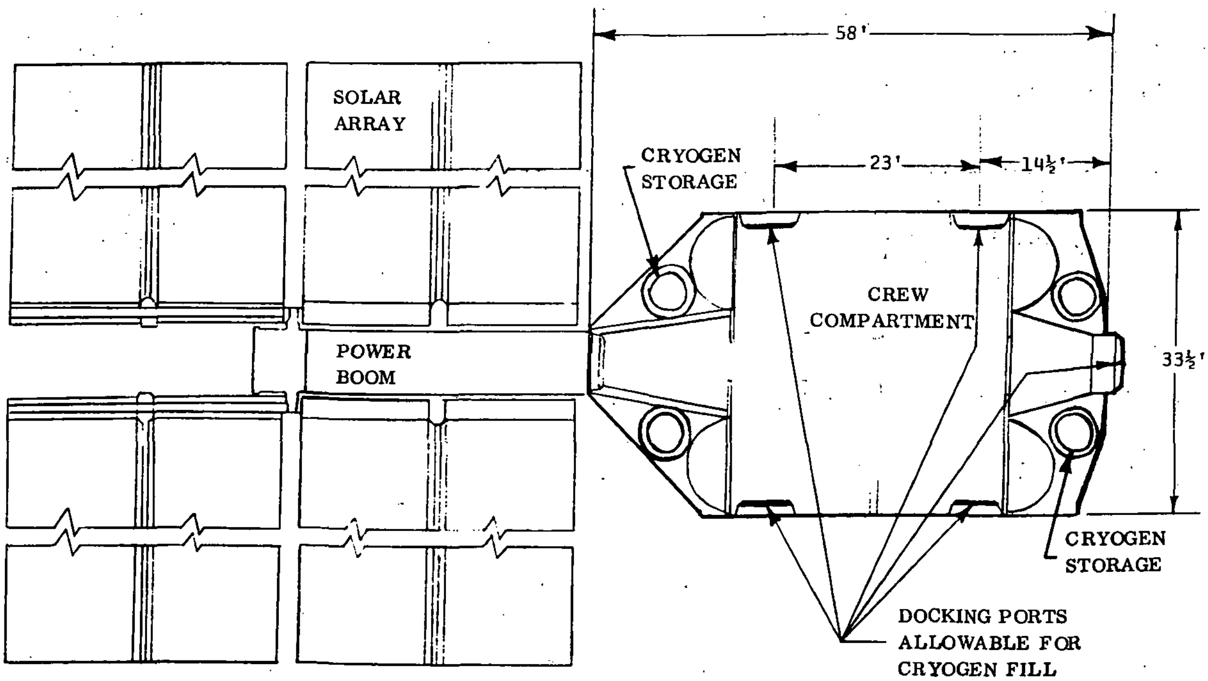


Figure 2-1. Space Station Major Features

opposite end of the station. All cryogen storage containers must be capable of being replenished from a vehicle docked to any one of the five docking ports. All cryogen storage containers are assumed to be spherical containers of a common size with volumes of 42.5 ft^3 each. Eight tanks are used for hydrogen storage, two tanks for oxygen storage and two tanks for nitrogen storage. All such storage is subcritical. The normal operating pressure is 100 psia with a maximum design pressure of 150 psia. Heat leak to the LN_2 tanks must be no greater than that required to boil off 100% of the LN_2 in six months. LO_2 and LH_2 tank heat leak must be no greater than required to boil off 50% of the respective fluids in six months.

From Reference 2-2 the total fluid quantities are 1096 lb of LH_2, 2480 lb of LO_2 and 3150 lb of LN_2. In all cases the LH_2 and LO_2 will be propellant grade fluids. Resupply is assumed to occur at a maximum of every six months. All tanks are of a common design and are assumed to be insulated with approximately 8 inches of high performance insulation (HPI). Also, the tanks are located external to the pressurized cabins in an uncontrolled environment. It is noted, however, that the station system design is such that tank equipment such as valving and control hardware are accessible in a "shirt sleeve" environment. In all cases fluid may be supplied to the user function as a saturated vapor or liquid. The individual tank weight is 225 pounds including insulation and internal hardware.

The shuttle supply vehicle will have a maximum acceleration of 3 G's. The thermal environment within the cargo compartment is undefined. It can be assumed that the propellant tanks for space station resupply are located inside the shuttle cargo compartment and can be topped off until 1/2 hour before liftoff. The fluid would be transferred from the shuttle to the space station at some time interval within seven days after docking the shuttle to the space station.

The maximum disturbing acceleration during the transfer operation is 10^{-4} G's. Based on the above data, information from References 2-3 and 2-4 and a desire to cover the full range of possible future station type resupply applications the following specific ground rules were used in generating the data contained in subsequent sections of this report.

1. The resupply fluids will be H_2, O_2 and N_2.

2. In each case both subcritical and supercritical receiver or space station tanks will be considered.

3. Transfer line lengths will range from 20 to 200 feet.

4. The maximum disturbing acceleration which can occur in any direction during transfer is 10^{-4} G's.

5. Crew tasks will be minimized and crew safety is a prime consideration.

6. Time from final topping of the supply bottles on the ground to start of transfer will be anywhere from 4 hours to 7 days.

7. Maximum boost acceleration is 3 G's.

8. The shuttle cargo compartment during boost may be either pressurized or unpressurized.

9. The space station bottles are in unpressurized areas as shown in Figure 2-1 and allowable heat leaks are such as to require venting of 100% of LN_2 and 50% of LO_2 and LH_2 when maintaining a constant storage pressure over the system operating life.

10. Heat sinks below 500°R are not available on the station for bottle cooling, and any such cooling must be a part of the supply system or proposed as an addition to the space station.

11. The station life is 10 years.

12. Full replenishment will be assumed to occur every six months with the capability of 50% replenishment every 90 days.

13. The individual receiver bottles will be assumed to be between empty and one-half full at the initiation of transfer.

14. Parametric studies of basic receiver or space station bottle sizes will range from internal diameters of 25 to 150 inches.

15. The number of bottles to be filled for each fluid will range from one to sixteen.

16. The total fluid quantities considered will range from 500 to 5000 lb of LH_2 and 1000 to 10,000 lb each for LO_2 and LN_2.

17. Any modifications or design features required of the receiver bottles for efficient operation of a particular supply mode will be defined during the study.

18. The supply bottles are assumed fixed in the shuttle cargo module during fluid transfer.

19. In all cases propellant grade fluids will be supplied.

20. The maximum allowable time for the transfer operation is 24 hours.

It is noted that Items 14, 15 and 16 of the above ground rule list represent in some cases, conflicting boundaries. As an example, for normal fluid load densities, the use of sixteen bottles of 150 inches diameter would greatly exceed the maximum fluid capacity requirements of 5,000 and 10,000 lb.

Therefore, in order to provide reasonable limits on the variables to be considered in the overall study, the data envelopes presented in Figures 2-2 and 2-3 were developed. These are based on maximum loading densities of 4.2, 67.5 and 47.5 lb/ft^3 for H_2, O_2 and N_2 respectively and corresponding minimum densities of 3.15, 57.8 and 39.6 lb/ft^3. Maximum densities are based on loading 15 psia saturated liquid to 95% full and minimum densities are based on 90% full saturated liquid at 100 psia. Fluid capacity ranges are per Item 16 of the ground rules list.

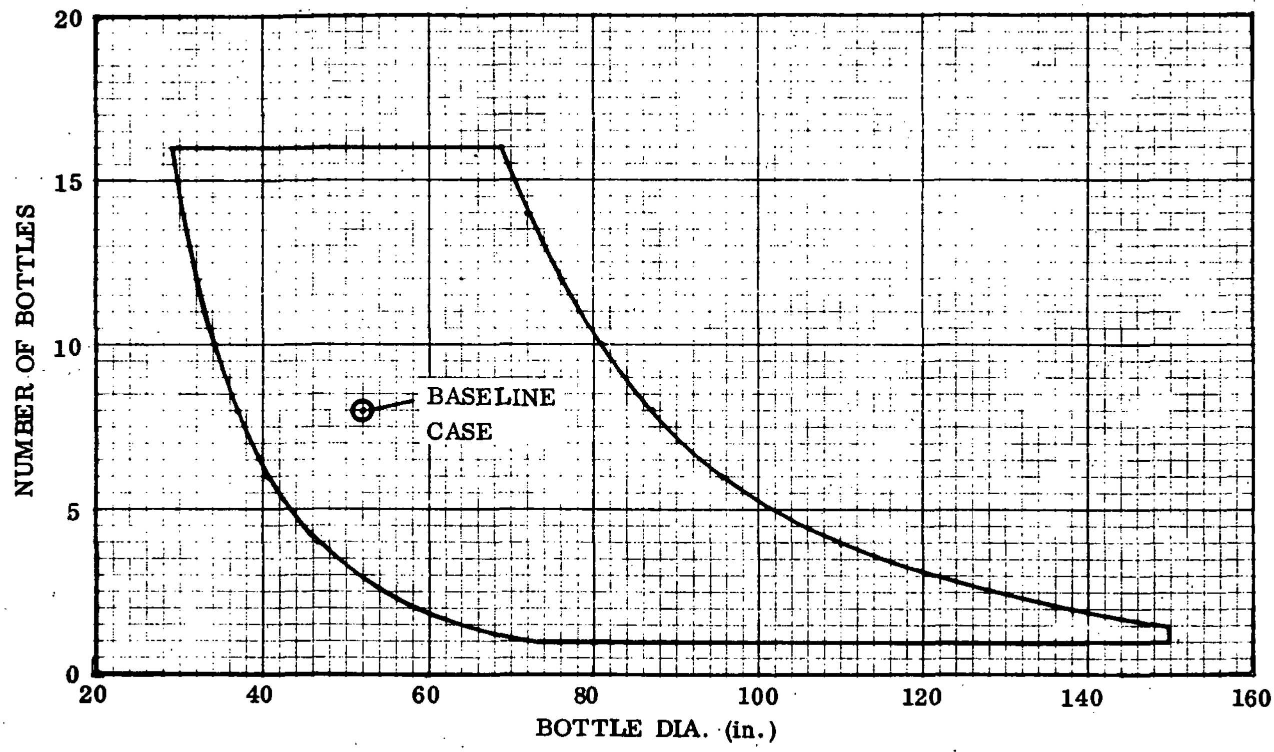


Figure 2-2. H_2 Systems Area of Interest

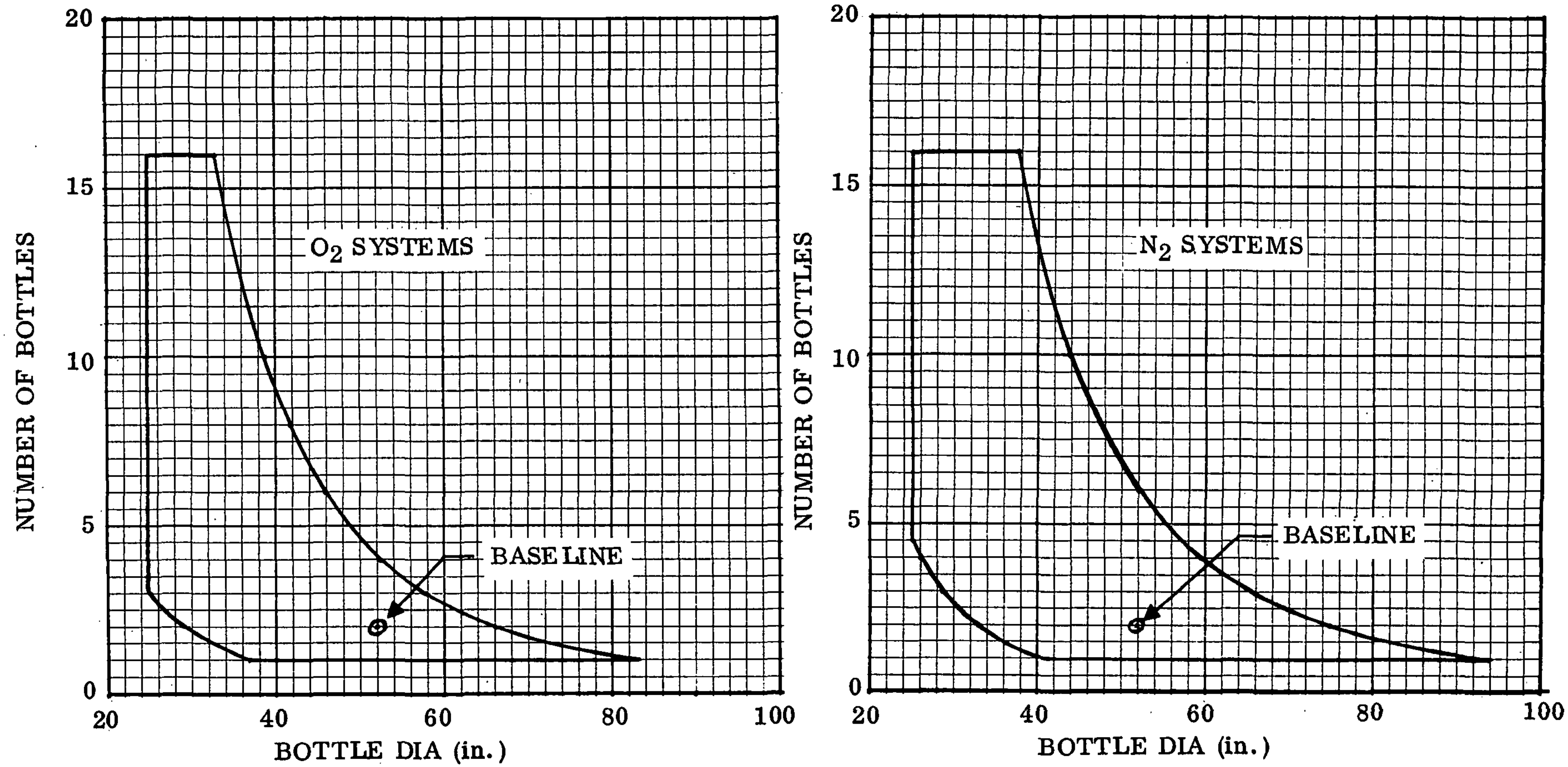


Figure 2-3. O_2 and N_2 Study Areas of Interest

3

PRELIMINARY SYSTEM SCREENING

A screening analysis was accomplished under the Convair Aerospace 1970 Independent Research and Development (IRAD) program to determine transfer system configurations most applicable to the transfer of cryogenic liquids to an orbiting system such as a space station. This work is reported below for reference as it relates to the present propellant transfer study.

The basic system elements considered in the analysis are illustrated in Figure 3-1. The screening was accomplished on the basis of eliminating systems and/or operations having low safety, excessive weight and/or low development potential.

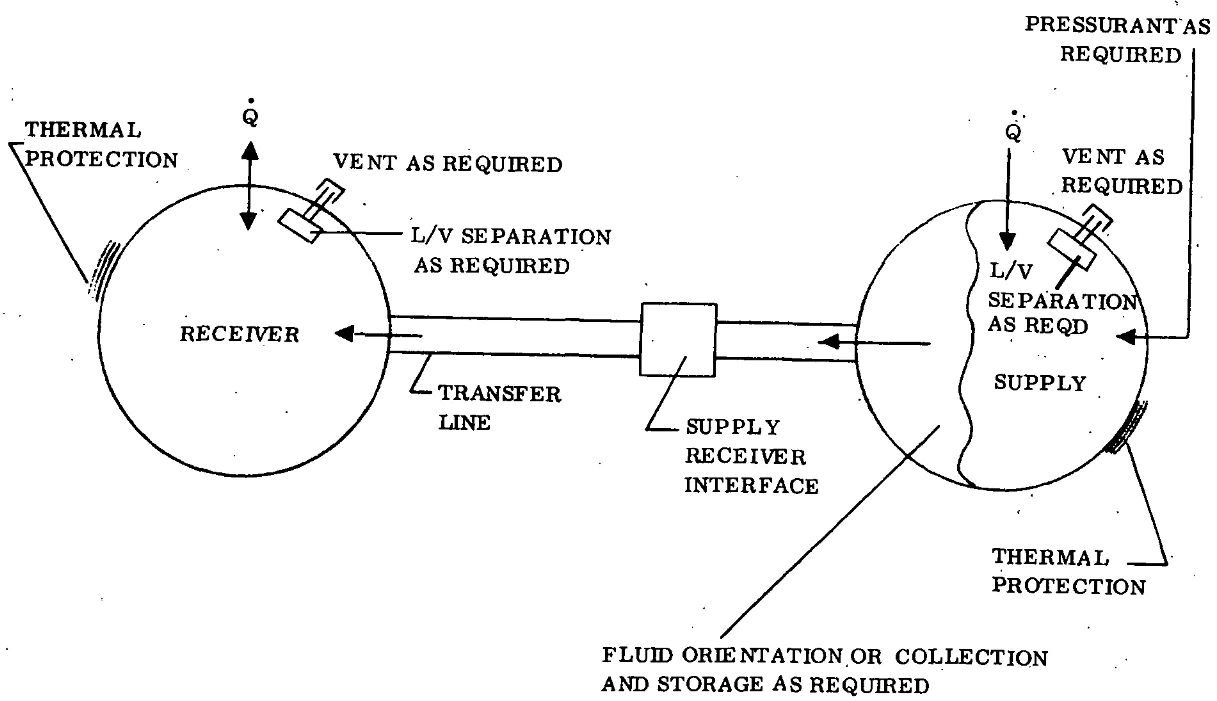


Figure 3-1. Fluid Transfer Elements

The primary problem with the intermediate or subcritical supply methods is that under low gravity some manner of orientation or collection of the liquid for transfer must be employed. Therefore, the subcritical systems are characterized by the method used for liquid orientation. In performing the comparison analysis, orientation methods were categorized as follows:

1. Surface Force Systems - Relying on surface tension or electrical force differences between the liquid and vapor to orient or collect the liquid for transfer.

2. Positive Expulsion Systems - Providing an essentially impermeable barrier between the pressurant and fluid to be transferred.

3. Dynamic Force Systems - Where the fluid is forced to move in a manner such that the liquid orientation is known and transfer can be accomplished.

An additional system was also considered where the liquid is vaporized and transferred as a saturated gas.

It is noted that for each of the system types listed above, the auxiliary system requirements and design problems relative to pressurization and/or pumping, thermal protection and venting, line and receiver tank chilldown and general receiver tank fluid conditioning will be similar within each category. Thus in performing the screening analysis such auxiliary system requirements were not included in the data developments. The data generated are therefore primarily used for system comparisons within each category.

In performing the screening, a nominal supply tank volume of 42.5 ft^3 was assumed and the desirable operating life of the system was taken to be 40 cycles. Systems having less capability were considered on the basis of periodic replacement. Both LO_2 and LH_2 were considered as the transfer fluids and the maximum transfer time was taken as 24 hours.

Results are presented in the following paragraphs. General state-of-the-art and safety discussions and background data developments are presented in Paragraphs 3.1 through 3.4. Paragraph 3.5 presents overall weights, efficiencies and relative evaluations of all the systems considered.

3.1 SURFACE FORCE SYSTEMS

Two such systems were considered; (1) a capillary collection system employing screens and (2) a dielectrophoretic system using dielectric properties of the fluid to effect collection. The systems are individually discussed below.

3.1.1 CAPILLARY SYSTEM. The system analyzed is shown in Figure 3-2. The liquid collector channels are designed to maintain continuous contact with the tank liquid while the cylindrical reservoir provides liquid flow in the event a disturbing acceleration forces the channels to be momentarily uncovered from the main liquid pool. The basic concept used in the present comparisons was developed under Contract NAS8-21465. The detailed configuration design data are presented in Reference 3-1. The weight of this system, as applied to a 52-inch LO_2 tank, is estimated to be on the order of 30 lb. Expulsion efficiency is estimated to be 98%, and the volumetric efficiency 99.5%.

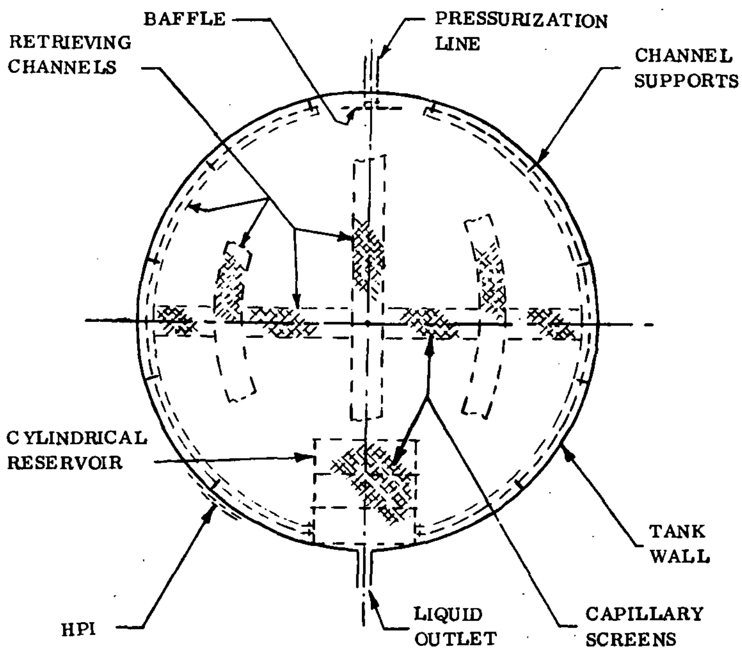


Figure 3-2. Channel-Reservoir Capillary Control Device

The primary advantages of such devices are that they are lightweight, generally passive and can be used for a number of cycles of operation. Such devices have previously been applied to non-cryogenic propellant acquisition for engine restart purposes. Using cryogenic fluids introduces thermodynamic and heat transfer problems which can significantly affect system design. The primary problem is to prevent vapor generation within the capillary device from causing a breakdown in the capillary barrier such that a direct vapor path is formed between the ullage and the tank outlet. Also, in the case of a continuous collection system operating at low gravity, communication between the tank outlet and liquid pool must be maintained throughout the transfer in order to minimize residuals. In general the residuals for the present collection system will be greater than that of an orientation system used for engine start since bottoming acceleration build-up will not occur in the present transfer application. The above problems were analyzed under contract NAS8-21465 and the overall results are presented in References 3-1, 3-2 and 3-3.

The resulting design utilizes the vent fluid to control the heat leak into the capillary device such that harmful vapor formation is prevented and the reservoir shown in Figure 3-2 assures continuous communication of the liquid with the tank outlet.

Analysis indicates that such design approaches will result in a reliable system for the transfer application with system life expectancy for the full number of transfers desired. Fluid cleanliness must, however, be maintained at a high level in order to prevent screen clogging from occurring over a period of time. In this regard periodic cleaning may be necessary.

3.1.2 DIELECTROPHORETIC SYSTEM. Dielectrophoresis is defined as the motion of matter caused by polarization effects in a nonuniform electric field. This electrical phenomenon may be used to orient and control a large class of dielectric fluids, including cryogenic hydrogen, oxygen and nitrogen. For the present application this orientation is accomplished by locating electrodes (electrostatic condensers) within the transfer tank such that the liquid is moved and drawn into the space between the high potential and ground electrodes.

A considerable amount of analytical and test data have been generated on this concept for use with LH_2 and LO_2 as applicable to the present program. The most complete information was found in References 3-4 through 3-7. Based on the information obtained from these references the use of a ribbon electrode configuration, as shown in Figure 3-3, is considered the most promising. The system weight (including power supplies), expulsion efficiency, and volumetric efficiency were estimated to be 40 lb, 99% and 99.5% respectively. Weight data were determined from the information contained in Reference 3-5.

The primary advantage of this system over the capillary screen device is that with the dielectrophoretic system, positive orientation is applied to the liquid such that vapor bubbles are forcibly expelled from the drain. Thus, vaporization within the electrode or expulsion channels is not a potential problem as it is with the surface tension device.

The main concern with the dielectrophoretic system is the complexity associated with required high voltage feed-throughs and power supplies. Also, there is some question of LO_2 compatibility where electrical discharges may occur. Furthermore the potential arcing of electrodes is forever present. Several NASA studies have been performed in order to demonstrate the safety of such systems. The data are presented in References 3-5 and 3-7. The latest testing most applicable to the present program was performed under Contract NAS8-20553 and reported in Reference 3-7. This report was reviewed by Dr. S. Kaye of the Convair Scientific Research Department who has developed extensive experience with respect to O_2 and H_2 combustion under NASA Contracts NAS8-11405 and NAS8-20350 and under the Convair IRAD program. A summary of his comments is presented below.

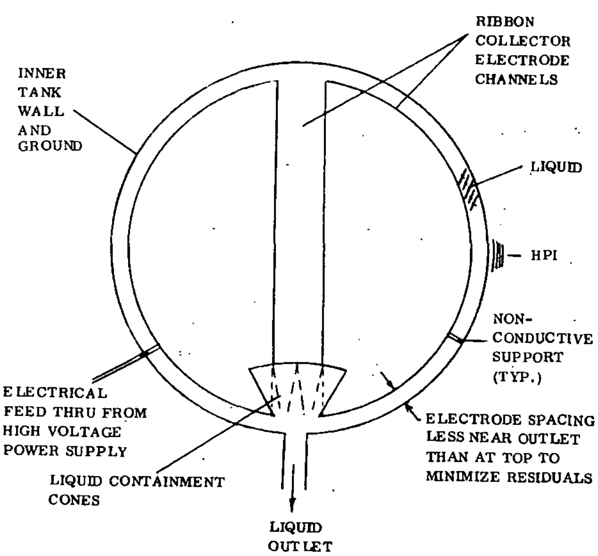


Figure 3-3. Dielectrophoretic Collection System

1. The data contained did not demonstrate that the operational safety of full size tanks can be predicted by subscale tests. The scaling was done only on the basis of breakdown voltage with respect to electrode spacing, pressure and

temperature. An important scaling parameter which was not considered in the analysis is the ratio of enclosed fluid mass or volume to surface area and tank mass. The important consideration here is the rate at which heat to the enclosed mass can be dissipated to the surroundings. A small tank is less likely to accumulate an explosive amount of heat (that required to start combustion of the metal electrodes and/or container) than a large tank.

2. The method of testing was not conducive to obtaining known mixture ratios of propellant gases and helium since the assumption of complete gas mixing, especially with helium and O_2, would be far from true as their densities are significantly different and no attempt was made to provide mixing. This would, however, only serve to shift the breakdown data curves and should not significantly affect the overall safety analysis.

3. The small sample testing of materials would not necessarily be valid for demonstrating complete safety, for the same reasons as stated in comment (1). Also, arcing time was not given or discussed and this would be an important factor in whether or not a fire could be initiated. Given enough energy, eventually ignition of the electrode and container material could occur.

4. The O_2 testing is considered to be by far the most critical and, due to the failure of the high voltage feedthrough, the test series was cut short such that the tests were not conclusive in proving O_2 system safety.

5. Another factor which was not completely analyzed, tested or otherwise accounted for was the specific shape and mass (as affecting system heat up) of the electrodes themselves. These factors can have a significant effect on ignition. Also, electrode shape and manufacture can have a strong effect on breakdown voltage and where it will occur. As an example a burr type of defect can cause a voltage or charge concentration which will cause premature breakdown or arcing at that point.

3.2 POSITIVE EXPULSION SYSTEMS

Such systems are designed to provide a positive barrier between the pressurant and fluid to be transferred. Bladder, bellows and diaphragm systems were considered and are discussed below. Pistons were not considered due to their combination of high weight and moving seal problems.

3.2.1 BLADDERS. A significant amount of development work has been accomplished on such systems. The folding type of non-metallic bladder, as shown in Figure 3-4, is considered most applicable to the present program. Such systems have been satisfactorily demonstrated for the expulsion of earth storable fluids and are presently in use with such fluids.

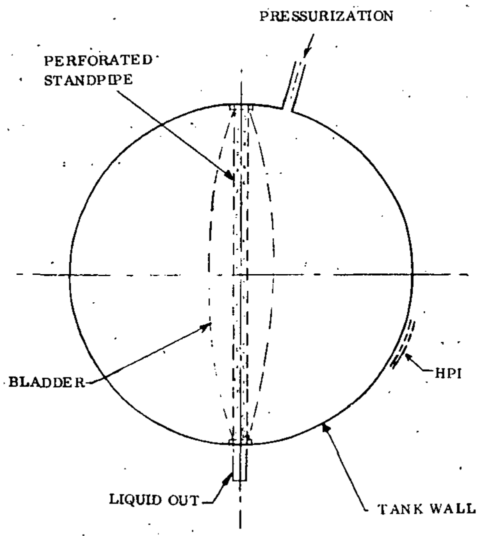


Figure 3-4. External Pressurized Bladder

The primary problem with the use of bladders for cryogenic fluid expulsion is in finding materials which are flexible at cryogenic temperatures and which can be incorporated into a satisfactory design. In considering use with LO_2, considerable work has been done on developing materials and adhesives which are both flexible and LOX compatible under the required operating conditions. The most recent and applicable data with respect to the development of such systems for use with cryogenics was obtained from References 3-8 through 3-17. In summary the development has progressed in the following manner.

Materials and complete system testing was performed in order to determine satisfactory materials and bladder fabrication techniques for the cryogenic application. Satisfactory systems were determined to consist of thin plys (on the order of 0.5 mils) of either Mylar or Kapton laminated together into a multi-ply configuration.

LO_2 compatibility testing was then performed on the individual materials using the ABMA sensitivity criteria that detonation shall not occur when the material is subjected to an impact of less than 72 ft-lb. The mylar did not meet this criteria. However, with proper baking during the fabrication process, the Kapton did meet it (Reference 3-15). In any case, it was thought it would be impossible to add enough energy to the bladder material itself for detonation to occur.

Subsequent full scale testing was performed with a 30-06 non-ferrous bullet fired into the bladder tank at high velocity. Both Mylar and Kapton multi-ply bladders were tested. In these tests the Mylar charred and the Kapton burned. It is believed that these failures were primarily due to use of non-LOX compatible adhesives.

Subsequent investigation and material testing was performed in order to develop a LOX compatible adhesive (Reference 3-11). This program was considered successful and full-scale bladder testing was then continued using the new adhesive. During these tests, again using a non-ferrous projectile, the Mylar did not react and the Kapton showed one reaction out of 12 tests; and in this case (Reference 3-15) the aluminum tank itself also burned.

This is essentially the current state-of-the-art of LO_2 expulsion bladders. According to Reference 3-15, bladders for LO_2 service have been satisfactorily demonstrated. Reference 3-15 indicated that such systems should ultimately be good for up to 25 cycles on a reliable and repeatable basis and test cycles on the order of 50 would be necessary in order to guarantee such a repeatable 25-cycle life.

An evaluation of all the data, however, indicates that there is still uncertainty as to whether present safety requirements would allow the use of such bladders for LO_2 service, since the Mylar is not basically LOX compatible and the Kapton system did burn in one test. Also, the length of exposure to LO_2 has an effect on increasing the material impact sensitivity, and full quantitative evaluation of this effect has not been demonstrated to date.

The main problem with developing suitable bladder systems for LH_2 has been permeation and inter-ply inflation. Interply inflation is caused by gas being trapped within the plies at cryogenic temperatures such that, when the system is warmed back to ambient, the gas expands and separation of the plies causes failure.

Initially, development of a single ply bladder for the LH_2 service was attempted but was not successful. Following this, work was initiated by the Boeing Co. under contract (Ref. 3-13) to NASA LeRC to develop an impermeable but flexible membrane to be used in conjunction with the polymeric materials. According to Reference 3-15, this program is promising and it is believed that a satisfactory system can be developed; however, the total number of predictable cycles for this hydrogen system is estimated at two or three with an ultimate of five.

It is noted that the bladder system shown in Figure 3-4 is of the collapsing type where pressurization is applied external to the bladder and the liquid to be expelled is internal. This is preferred for the following reasons, as summarized from Reference 3-17.

1. Internal pressurization would tend to trap propellant between the bladder and the tank wall thus reducing expulsion efficiency.

2. At initial loading, the internally pressurized bladder is folded around the standpipe, resulting in a folded and creased bladder that is then subjected to the booster vibrational environment such that bladder flexural failure may occur.

3. Significantly more analytical and operational knowledge is available on externally pressurized bladders.

At one stage in the development, it was thought that an expanding bladder would eliminate the interply inflation problem, however, subsequent testing (Reference 3-12) showed this to be untrue and the latest recommendation is for the collapsing type.

Based on a perusal of the data from References 3-8 through 3-17, the expected weight, expulsion efficiency and volumetric efficiency for the 42.5 ft^3 tank application were estimated to be 40 lb, 98% and 98% respectively. Weight is based on using a bladder consisting of 10 plies of 0.5 mil Kapton where the standpipe and associated hardware represent 85% of the total system weight.

3.2.2 METALLIC BELLOWS. Although these systems are in general heavier than other systems, they have the advantage of operation over a significant number of expulsion cycles. Also, by measuring the stroke of the bellows the fluid quantity remaining can be quite accurately determined, even at low-gravity. A typical system which was considered in the present study is shown in Figure 3-5.

The pressurant storage is assumed to be contained within the overall tank envelope in order to increase the volumetric efficiency of the system. This configuration is similar to that tested under Contract NAS3-12017 (Reference 3-18).

The two main types of bellows presently available are welded and formed. The main disadvantages of the welded type are that the extensive welding required makes these bellows difficult to clean, inspect and seal. Also, the life expectancy is in general less predictable and they are more expensive than the formed type. Initially, the primary advantage of the welded type was a high expulsion efficiency. New formed bellows designs have, however, been developed having essentially as high expulsion efficiencies as the welded types. Pertinent data used in estimating weights, expulsion and volumetric efficiencies and state-of-the-art of these systems were obtained from References 3-15 through 3-24.

The design illustrated in Figure 3-5 utilizes the nested type of formed bellows for potentially high reliability and high expulsion efficiency. These bellows are described in References 3-19 and 3-22 and are similar to those used in the test program described in Reference 3-18.

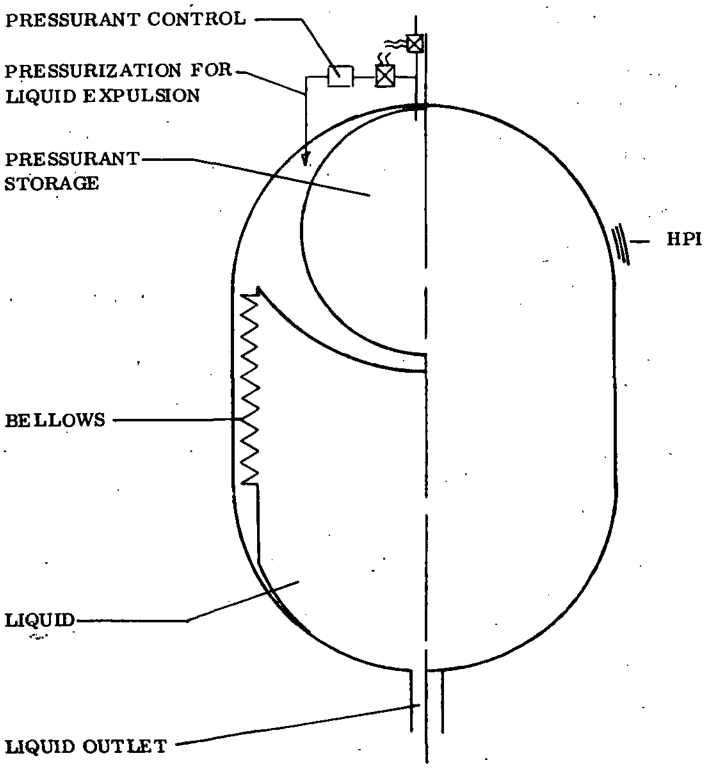


Figure 3-5. Bellows Expulsion System

Such bellows have previously been used fairly extensively for expulsion of storable fluids, as in the Minuteman Program (Reference 3-23). In addition, several development programs have been accomplished or are under way to provide reliable bellows systems for use with cryogenics. As an example, testing was performed on a 7 × 11 inch bellows where more than 100 cycles were accomplished with LN_2 (Reference 3-21). It is noted that this particular bellows was of the welded type.

Testing was also accomplished on the nested formed type of bellows described in Reference 3-19 using LH_2. This test program, performed on a 13.5 inch diameter bellows, is described in Reference 3-18. The first bellows tested developed a leak during LN_2 and LH_2 checkout testing. A second bellows was tested and successfully completed 50 expulsion cycles (100 reversals) before a leak was detected, (2×10^{-5} scc He/sec). Failure was assumed to occur when the leak rate exceeded 10^{-6} scc He/sec. The program target was 100 complete expulsion cycles. A failure analysis of the bellows correlated the leaks with regions of corrosion found within the bellows. These are believed to be the result of inadequate cleaning procedures, and/or the use of tap water in some of the test operations. Indications are that these deficiencies can be overcome in the future.

At present it is considered that the main problem needing further testing is associated with rubbing or impacting of the bellows convolutions on the container wall under vibration and expulsion dynamic conditions. Such interference conditions tend to reduce the bellows life. Several programs were initiated to provide further cryogenic design data for the bellows system. One such program is being conducted by Bell Aerosystems under Contract NAS3-13327. Under the program, bellows fatigue and life data will be generated under realistic dynamic loadings to be expected in operation. It is thought that such data are still needed for the proper design of cryogenic bellows, however the feasibility of such systems for greater than 100 cycles should not be a basic problem (Reference 3-23).

At present, bellows diameters up to 4 ft are projected without significant difficulty (Ref. 3-24). Total volumes of such bellows would be on the order of 40 to 50 ft^3. Significant tooling developments would be needed, however, for larger size systems. According to Reference 3-20 bellows diameters up to 50 ft are feasible.

It is noted that a significant weight penalty is paid by this system due to its fairly low volumetric efficiency and the fact that a cylindrical rather than a spherical tank is required as the basic container. Based on a direct comparison of cylindrical versus spherical tank weights, it was estimated that the use of the cylindrical tank would increase the container weight by approximately 25 per cent over that of the basic spherical system.

3.2.3 DIAPHRAGMS. Development work has been accomplished on both non-metallic and metallic diaphragms. Data applicable to the general evaluation of such systems has been obtained primarily from References 3-15, 3-16, 3-17 and 3-25 through 3-29. Testing accomplished by Boeing under Contract NAS3-12204 on polymeric positive expulsion diaphragms for cryogenics was not successful in that the configuration tested did not collapse properly to expel the LH_2 (Reference 3-15). The diaphragm collapsed into a cone shape and then became rigid. Data from Reference 3-15 indicated that a satisfactory shape could have been developed, however due to high flange weights and sealing problems, such systems would not be practical beyond diameters of about 24 inches.

The system considered to be most applicable to the present program is one using a metallic diaphragm and operating as shown in Figure 3-6. Significant development and

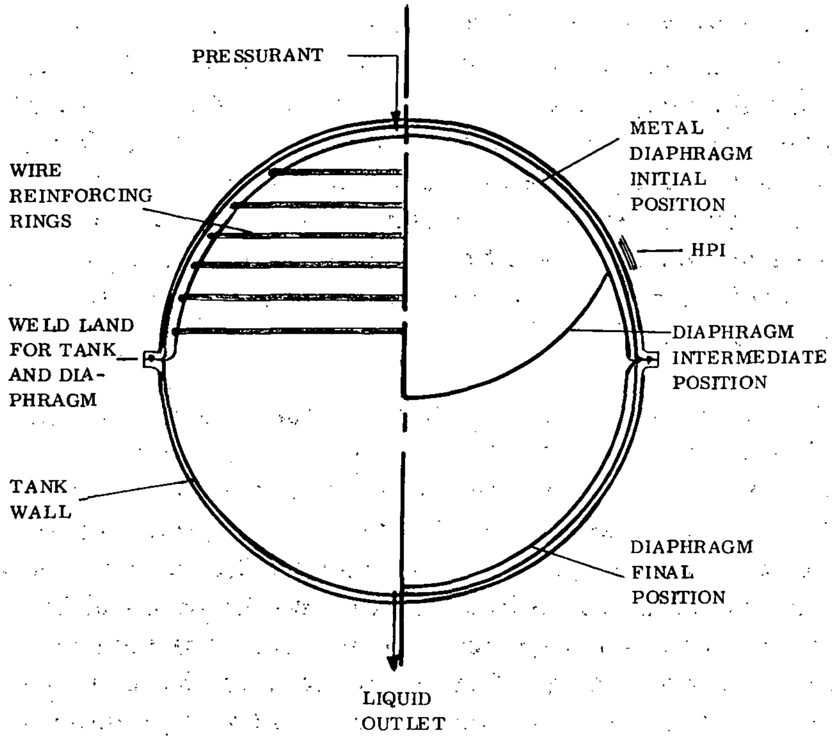


Figure 3-6. Metallic Diaphragm System

testing of such systems has been accomplished by Arde Inc. of Paramus, N.J. Pertinent data are presented in References 3-26 through 3-29. The system basically consists of a thin metallic shell with wire rings brazed circumferentially at controlled intervals on the shell. The wire rings are used to stabilize the folding pattern in order to allow multiple expulsions to be accomplished. The diaphragm is integrally welded into the storage bottle.

Testing to date has shown that approximately the same number of reversals can be obtained at cryogenic temperatures as at ambient. Seven reversals have been obtained with LH_2 using a 24 inch diameter system and a program is presently underway under Contract NAS3-12026 to improve the design to allow an increase in the number of cycles (Ref. 3-29). Up to eleven reversals have been accomplished to date on an 13.5 inch diameter system at room temperature. Additional fabrication and satisfactory testing have been performed on systems up to 6 ft in diameter. Data from Reference 3-29 indicated that with present technology, diameters up to 90 inches would be attainable.

It is noted that with the proper plumbing arrangement each reversal could be designed to accomplish a liquid expulsion cycle. For purposes of the present comparisons, the maximum number of repeatable cycles was estimated to be five.

3.3 DYNAMIC FORCE SYSTEMS

The fluid vortexing method of dynamic liquid control was chosen as that most applicable to the space station resupply from a shuttle vehicle. Linear acceleration or rotation of the entire shuttle and station was not considered practical. Rotation of the bottle within the payload is possible, but was not considered desirable in comparison with fluid vortexing due to the requirement for a stationary to rotating connection.

Two basic methods can be considered for applying a vortex motion to the tank fluid. These are illustrated in Figure 3-7 and 3-8. The system shown in Figure 3-7 uses an internal paddle to provide a positive vortexing action to the fluid while the system of Figure 3-8 removes fluid from the tank and tangentially injects a portion of it back into the tank to provide the required vortexing.

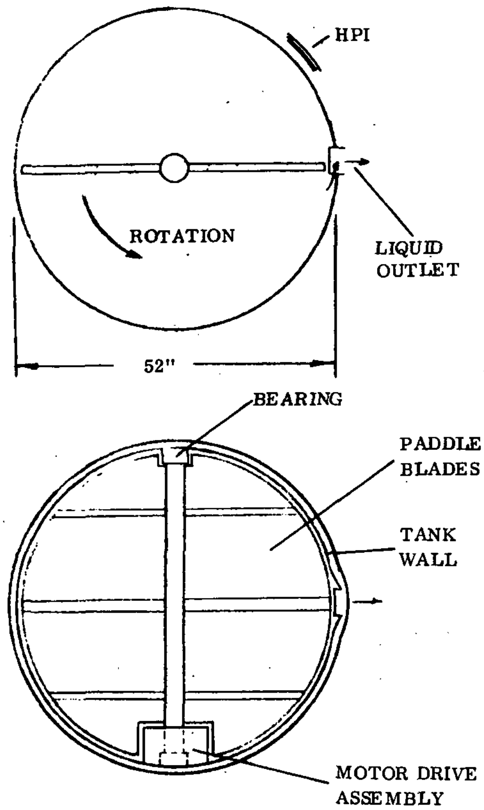


Figure 3-7. Paddle Type Vortex System

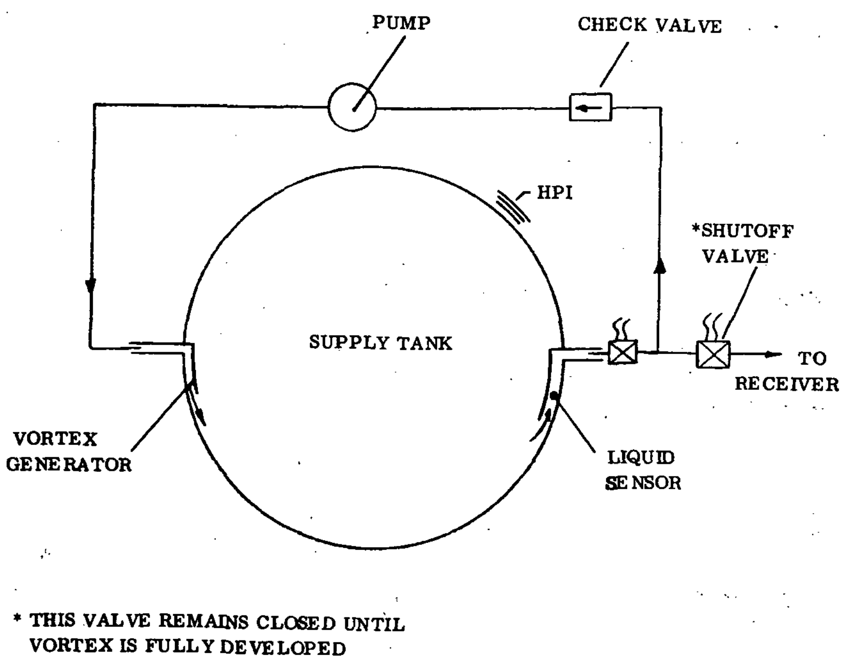


Figure 3-8. Jet Type Vortex System

The paddle system has the advantage of reduced residuals and the disadvantage of larger hardware weight with the requirement for an internal tank motor or a tank pass-through for applying rotation motion to the paddle. The injection pumping scheme will have minimum hardware weight, but with a likely increase in total electrical power and residual liquid. Also, start up of this sytem may be relatively slow due to the fact that most of the initial fluid being injected back into the tank may be vapor.

Significant quantitative analysis has not been performed on these particular systems. Some analysis has been reported on propellant tank rotation which does give an indication of the energies and forces associated with the fluid rotation problem. Typical data of this nature are contained in References 3-30 through 3-32.

In general the fluid dynamics of the process are quite involved and a detailed analysis of the system was not within the scope of the screening task. Weight, power and fluid residuals were, however, estimated on an order of magnitude basis in order to determine whether the vortexing system could in any way be competitive with other methods of transfer considered. Due to its relative analytical simplicity, the system shown in Figure 3-7 was chosen for analysis. It is realized that this system does not necessarily represent an optimum design, but should be representative of the total weight to be expected for the vortex concept.

The basic approach was to determine at what rate the fluid must be rotated in order to insure liquid at the wall and then to determine the power and subsequent hardware weight required to accelerate and maintain the fluid at this rotation. Oxygen was taken as the critical fluid due to its large mass as compared to hydrogen.

Based on the data contained in Reference 3-32 the rotational speed of the fluid required to maintain liquid at the wall can be estimated from the following equation.

$$\Omega_c^2 = \frac{\omega^2 \rho R_t^3}{\sigma g_c} \tag{3-1}$$

where Ω_c = a critical rotational Weber No. below which liquid will not be forced to the outer wall, assuming solid body rotation of the fluid.

Ω_c is a function of the Bond No., $Bo = \rho_L a R_t^2/\sigma$ and the liquid to solid contact angle. For the present case the contact angle is taken to be zero and based on

$$\rho_L = 70 \text{ lb/ft}^3, \; a = 10^{-4} \text{ g's}, \; R_t = 13 \text{ in., and } \sigma = 8.9 \times 10^{-4} \text{ lb}_f\text{/ft}$$

the Bond No. (Bo) = 9.25

It is noted that in this case R_t is based on the average radius of a 52 inch diameter tank in order to be conservative, since the data of Reference 3-32 was generated for a cylinder rather than a sphere. It is noted that from Reference 3-32 the required angular speed decreases as the radius is increased. Based on an extrapolation of the data contained in Reference 3-32, $\Omega_c^2 = 4 + Bo/0.2 = 50$. Then from Equation 3-1

$$\omega = 0.13 \text{ rad/sec} = 1.25 \text{ rpm}$$

A further design criteria was also considered where it was assumed that liquid must be pumped from the inner radius of the paddle to the outer radius against 10^{-4} g's. From a simple force balance on a fluid element, the following equation determines the required rotation rate.

$$\omega = \sqrt{a/R_i} \tag{3-2}$$

Assuming an inner paddle diameter of 3 inches, which is reasonable for the supporting shaft, then from Equation 3-2 at $a = 10^{-4}$ g's

$$\omega = .16 \text{ rad/sec} = 1.54 \text{ rpm}$$

Using this as the final design criteria and applying a safety factor of two, the design rotation rate for determining power requirements was taken as 0.32 rad/sec or 3.08 rpm.

Assuming a clearance between the rotating paddle and the tank wall such that form drag on the paddle is the major retarding force then

$$F_D = C_D \rho_L A_p \frac{(Ve)_p^2}{2g_c} \tag{3-3}$$

In the present case the Reynolds No. was calculated to be sufficiently in the turbulent region such that $C_D \cong 1.0$.

Analysis indicated that friction drag or that due to boundary layer shear between the paddle and wall was negligible at these low rotation rates for reasonable paddle to wall clearances.

Assuming the drag force to act entirely at the outer periphery, again a conservative assumption, the required power is determined from $\dot{P} = F_D R_p \omega_p$ to be 7.3 watts.

Taking the total fluid mass as 2830 lb, which assumes a 95% full 42.5 ft^3 tank, the energy required to accelerate to 3.08 rpm was determined from

$$KE = \frac{1}{2} W_L K^2 \omega^2 \tag{3-4}$$

to be 8.5 ft-lb or 11.5 watt-sec. Assuming the steady state requirement of 7.3 watts of power continuously applied to the paddle then approximately 1.6 seconds would be required for start up, which is a very small time when considering the overall transfer process. The basic power supply was assumed to be fuel cells with the following weight assessments.

$$(W_T)_P = 94 \text{ lb/kw plus } 2.9 \text{ lb/kw-hr} \tag{3-5}$$

Based on a motor-drive efficiency of 50% (14.3 watts of continuous motor input power required) and a maximum transfer time of 24 hours, the total power supply weight was calculated to be only 2.4 lb. Therefore, the power requirements of the system are low for this case and the main weight associated with this system can be attributed to fluid residuals and motor, gear box and paddle assembly weights. It is noted that the large speed reduction and associated gearing required could result in the main weight penalty associated with the motor-drive system. The basic volumetric efficiency of the system for use in a 52 inch diameter tank was estimated to be 98% and the hardware weight, including motor and gearing was estimated to be approximately 75 lb.

For the configuration shown in Figure 3-7 it is assumed that fluid residuals will consist of that liquid which can be located between the paddle and the tank wall. Using a 1/2-inch clearance between the wall and paddle, an expulsion efficiency of 94% results. Calculations were also made for the liquid hydrogen case showing a slightly lower power requirement and overall system weight.

3.4 EVAPORATION SYSTEM

This system relies on evaporation of the stored liquid by the input of heat to effect transfer. In this case the fluid transferred is assumed to be in the form of a vapor. In the basic system initially considered this vapor must then be condensed to restore the fluid to its original supply condition. The system is illustrated schematically in Figure 3-9.

Weight and power calculations were initially made for liquid hydrogen transfer which takes place in 24 hours. Assuming a single 42.5 ft^3 tank containing 95% liquid, the mass of hydrogen to be transferred is calculated to be 177 lb. Based on a simple evaporation energy balance, where the heat of evaporation (λ) equals 189 Btu/lb, the required tank heating is determined to be 9,800 watt-hrs or 408 watts.

Based on the power supply penalties, as expressed in the previous paragraph, the weight penalty for such heating would be 67 lb. In order to determine system weights for cooling the receiver, the data from References 3-33 and 3-34 were used. It is assumed that a refrigeration system operating between 40°R and 500°K is required to remove the heat added due to the heating above. Solar cells aboard the space station were assumed as the power source. Assuming a completely independent closed cycle refrigeration

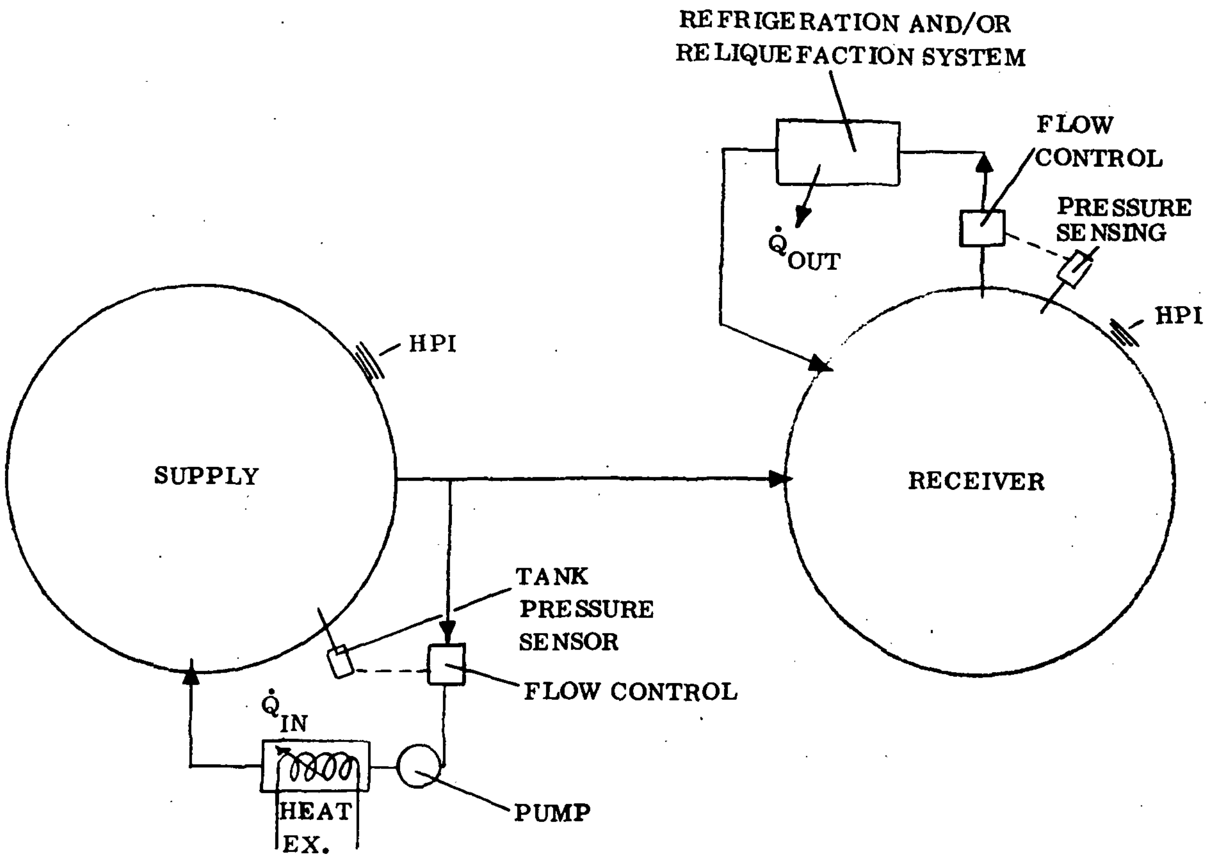


Figure 3-9. Liquid Vaporization System

system, then from the data of Reference 3-33 a total weight, including radiator and solar cells, was estimated to be 20,400 lb. This is considered to be an intolerable penalty.

Assuming that the hydrogen being transferred is used in the refrigeration cycle, a reliquefaction system requirement was determined from the data of Reference 3-34. For reliquefaction of 177 lb of GH_2, where the GH_2 is initially at 40°R, a minimum total weight penalty, as estimated from Reference 3-24, was found to be 2,660 lb. This is still considered excessive as compared to other liquid transfer systems being considered.

Further analyses were then made to determine the possibility of not condensing the transferred fluid and letting the pressure buildup in the supply and receiver bottles. Calculations for this condition indicated a potential pressure rise on the order of 1,000 psi. This puts the system in the supercritical pressure range and therefore this system was not considered further for liquid transfer.

In the case of LO_2 reliquefaction, for a single tank where the mass to be transferred is 2830 lb, a system weight of approximately 5,600 lb was estimated.

3.5 OVERALL SYSTEM COMPARISONS AND RECOMMENDATIONS

Based on the discussions of the previous paragraphs and an analysis of the pertinent data contained in References 3-1 through 3-35, representative comparison data were generated on the various systems. These data are presented in Tables 3-1 and 3-2. The main objective was to provide data applicable to relative evaluations of the various systems within each major category rather than to obtain absolute magnitudes.

Data are presented for both oxygen and hydrogen transfer fluids. In each case a single 42.5 ft^3 supply tank was assumed. The hardware weight is taken to include only the fluid orientation or collection device and power supply weights required for the orientation or collection. Auxiliary system weights such as required for venting, pressurization and pumping were not included. Basic tank weights are not included, however, differences in tank weight are estimated between the various systems. As an example, the increase in tank weight due to volumetric efficiencies less than 100% as well as that due to additional flanging required for certain systems is presented. The tank weight is taken to be proportional to the total tank volume required. The reference spherical tank weight is taken to be 225 pounds including high performance insulation. Also, as is the case with the bellows system, required tank shapes other than spherical will result in larger weight and are taken account of in the analysis.

It is noted that for each of the major system types listed in Tables 3-1 and 3-2, auxiliary system requirements and design problems relating to pressurization and/or pumping, thermal protection and venting, line and receiver tank childown and general receiver tank fluid conditioning will be similar within each category. It was therefore deemed desirable to choose a representative system from each category for further overall detail system definition, analysis and comparisons.

Based on the data presented in Tables 3-1 and 3-2, the following systems were chosen for further detail definition and analysis.

1. Surface tension or capillary containment system using screens.

2. Metallic bellows for positive expulsion.

3. Fluid vortexing within a restrained tank in order to orient the liquid at the outer periphery for transfer.

The surface tension system was chosen over the dielectrophoretic surface orientation system primarily on the basis of potential safety. Weights and state-of-the-art of the two systems are comparable; however for use in oxygen there is still some question of electrical breakdown and associated combustion hazard associated with the dielectrophoretic system.

Table 3-1. Subcritical System Comparison Data (42.5 Ft^3 LO_2 Tank)

	Surface Force Sys.		Positive Expulsion				
	Capillary Screens	Dielectro-phoresis	Bladders	Bellows	Diaphragms	Fluid Vortexing	Liquid Vaporization
* Hardware Weight, Lb	30	40	15	72	70	75	5,600
Δ Expulsion Efficiency, %	98	99	98	98	99	94	99
** Fluid Residuals, Lb	56	28	56	56	28	168	14
+ Volumetric Efficiency, %	99.5	99.5	98	90	98	98	99.5
*** Increased Tank Weight, Lb	1	1	4	87	16	5	1
Total Weight, Lb	87	69	75	215	114	248	5,615
Total Life, Expulsion Cycles	>40	>40	25	>40	5	>40	>40
† Safety	2	4	4	2	2	2	2
† Complexity/Reliability	2	4	3	3	3	3	3
† Development Potential/ State-of-the-Art	2	3	3	2	2	2	2

Δ Expulsion efficiency defined as [(Fluid Loaded) - (Fluid Remaining)]/ (Fluid Loaded)

\+ Volumetric efficiency defined as [(Total Tank Volume) - (Unusable Volume)]/(Total Tank Volume)

† Relative ratings; 1 through 5 where 1 represents best. No absolute value significance intended.

* Includes only the orientation or collection device and power supply for direct system operation.

** Assumes initially 95% full tank of LO_2.

*** Represents additional tank weight due to less than 100% volumetric efficiency plus any increase due to required flanges or tank shapes other than spherical. Reference spherical tank weight taken to be 225 lb.

Table 3-2. Subcritical System Comparison Data (42.5 Ft^3 LH_2 Tank)

	Surface Force Sys.		Positive Expulsion				
	Capillary Screens	Dielectro-phoresis	Bladders	Bellows	Diaphragms	Fluid Vortexing	Liquid Vaporization
* Hardware Weight, Lb	30	40	15	72	70	70	2,660
Δ Expulsion Efficiency, %	98	99	98	98	99	94	99
** Fluid Residuals, Lb	4	2	4	4	2	11	2
+ Volumetric Efficiency, %	99.5	99.5	98	90	98	98.5	99.5
*** Increased Tank Weight, Lb	1	1	4	87	16	5	1
Total Weight, Lb	35	43	23	163	88	86	2,663
Total Life, Expulsion Cycles	>40	>40	5	>40	5	>40	>40
† Safety	2	2	2	2	2	2	2
† Complexity/Reliability	2	4	3	3	3	3	3
† Development Potential/ State-of-the-Art	2	3	4	2	2	2	2

Δ Expulsion efficiency defined as [(Fluid Loaded) - (Fluid Remaining)]/ (Fluid Loaded)

\+ Volumetric efficiency defined as [(Total Tank Volume) - (Unusable Volume)]/(Total Tank Volume)

† Relative ratings; 1 through 5 where 1 represents best. No absolute value significance intended.

* Includes only the orientation or collection device and power supply for direct system operation.

** Assumes initially 95% full tank of LO_2.

*** Represents additional tank weight due to less than 100% volumetric efficiency plus any increase due to required flanges or tank shapes other than spherical. Reference spherical tank weight taken to be 225 lb.

In the case of the positive expulsion methods, the metallic bellows was chosen as the only system potentially capable of meeting the number of expulsion cycles desired (40 cycles) for the station resupply application. Also, even though somewhat heavier than other methods, the potential of developing a reliable and predictable system for use with cryogenics is believed to be higher with the bellows system. It is noted that other methods such as the metallic diaphragm can be considered and compared with the bellows system even though not having a total life comparable to that of the station. This comparison would be on the basis of total cost and would take account of replacing such systems or expulsion components after a number of flights.

The fluid vortexing method of dynamic liquid control is seen to represent a weight penalty in the case of LO_2 transfer when compared to other systems. This is due primarily to the potentially high residuals associated with this system. When considering overall safety and development potential or state-of-the-art, this system is considered representative enough to justify further detailed analysis.

Due to the very high weight involved the liquid vaporization system was not considered further.

4

HIGH PRESSURE SYSTEM DEFINITION AND ANALYSIS

Due to the difficulties associated with orienting and/or collecting liquid at low-gravity it was anticipated that the transfer of fluid under single phase or high pressure (above critical) conditions might show promise for space transfer applications. The normal method of expelling supercritical fluids from a supply tank is by heating. This is accomplished with either an internal heater and mixer or by an external heater and closed-loop circulation pump. The major concern with this type of high pressure transfer is that significant energy must be added to the supply in order to effect transfer and this energy must then be effectively removed from the transferred fluid or a significantly reduced receiver tank fluid storage density accepted.

In order to determine the feasibility of high pressure transfer various conventional and non-conventional schemes were investigated under the Convair Aerospace IRAD program. The IRAD data obtained are reported in Reference 4-1 and repeated in Paragraph 4.1 below for reference as it relates to the present propellant transfer study.

Paragraph 4.2 presents a detailed analysis and development of parametric data of the best high pressure system as determined from the Convair IRAD program.

4.1 SYSTEM DEFINITION

The work presented in this section was performed under the Convair Aerospace 1971 Independent Research and Development (IRAD) program and is reported herein only for reference as it relates to the present cryogenic propellant transfer study.

Work reported in this section was directed toward the conception, definition (sizing) and feasibility of supercritical (high pressure) cryogen transfer systems under low-g conditions. The systems or methods of high pressure cryogen transfer presented in this section are: (a) simple blowdown, (b) expulsion by heating, (c) pumping, (d) heating and pumping, (e) cooling receiver-heating supply, (f) cooling receiver-heating supply plus regeneration, and (g) vortex tube assisted transfer.

The definition analyses for these systems was performed with the Plumber Computer code as described in Appendix B. All work was based on hydrogen and, except for systems (e) and (f), initial receiver pressure was assumed to be 100 ± 20 psia. It was further assumed that the state of the fluid in the supply system must be supercritical or gaseous at all times to insure expulsion of a homogeneous fluid. Tank weights were determined from Appendix A and insulation weights are based on the use of the Convair

Aerospace high performance insulation "Superfloc" with an overall $K_{eff} = 6.5 \times 10^{-5}$ Btu/hr-ft°F and $\rho = 1.3$ lb/ft^3. The allowable receiver heat leak was taken from the ground rules of Section 2.

Initial fluid densities for a depleted receiver were assumed to vary between 0.1 and 0.5 lbs/ft^3. Required mass transfer to replenish a depleted receiver is assumed to be 137 lbs of hydrogen. A receiving station is assumed to be comprised of 8 such receivers.

4.1.1 TRANSFER BY SIMPLE BLOWDOWN. The system is shown schematically in Figure 4-1. Fluid is transferred from supply to receiver by pressure differential between the two bottles. Transfer is terminated when pressure equilibrium is attained between the bottles. Control or crew involvement beyond connecting the bottles and operating the shutoff valves is not required. Two-phase flow in the line is eliminated by taking the major pressure drop at the receiver inlet and keeping the line at essentially the supply tank pressure.

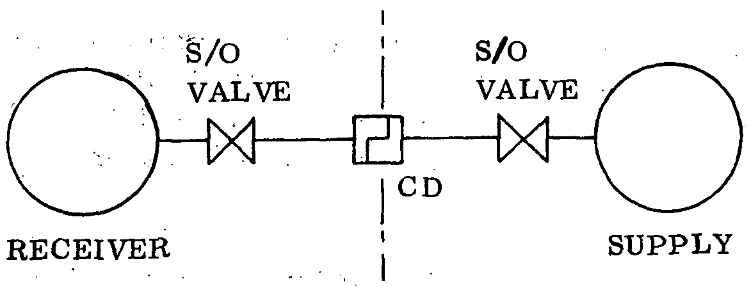


Figure 4-1. Schematic - Simple Blowdown System

The sizing and feasibility analysis for this method of transfer was performed under the following conditions.

Equivalent Orifice Diameter of Transfer Line	= 0.5 inch	= constant
Heat Through Transfer Line	= 0	= constant
Heat to Supply Fluid	= 0	= constant
Heat to Receiver Fluid	= 0	= constant
Initial Fluid Mass in Supply	= 168 ± 2 lbs	= constant
Initial Receiver Pressure	= 100 psia	= constant
Initial Receiver Density	= 0.05 to 0.5 lb/ft^3	= variable
Receiver Volume	= 42.5 to 600 ft^3	= variable
Initial Supply Pressure	= 300 to 5000 psia	= variable
Supply Volume	= 42.5 to 200 ft^3	= variable

Figure 4-2 shows the effect of receiver volume on total mass transferred for the conditions listed. This figure indicates that the mass transferred for the listed conditions is far short of the desired quantity. In examining the entire range of data generated, no combination of supply and receiver sizes was found that would transfer 137 lbs of hydrogen. The trend is obvious however, that even if a combination can be found, the weights of the bottles will be prohibitively high.

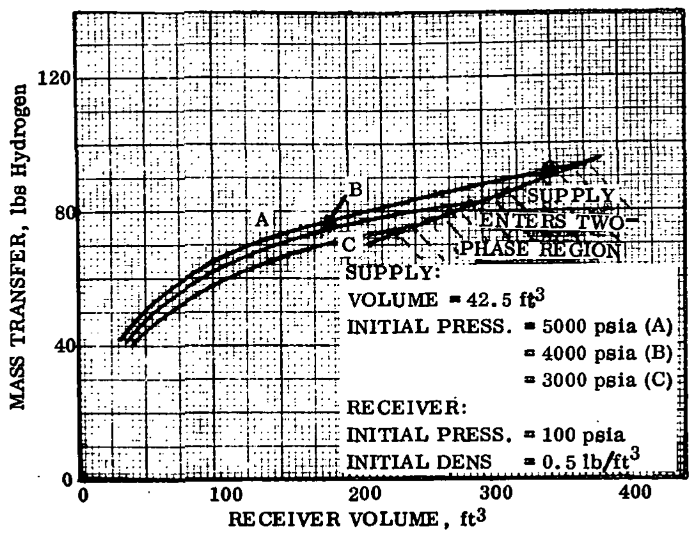


Figure 4-2. High Pressure Fluid Transfer by Simple Blowdown

A modification to the transfer procedure was made in an attempt to increase mass transfer. In this procedure the depleted receiver bottle was blown down to 5 psia or less before initiating the transfer of fluid from the supply. This doubled fluid transfer under some conditions, but in all cases where the fluid state in the supply bottle remained super-critical or gaseous, the increase in transferred mass did not offset mass discarded during the initial receiver blowdown. In the final analysis the high-pressure low-temperature fluid acts much like an incompressible liquid during the blowdown.

The following is a weight breakdown for the simple blowdown system transferring the most fluid, as determined from Figure 4-2. These weights should be used only for comparison with other systems discussed herein.

Single Receiver Bottle (Bare), 350 ft^3 @ 200 psia peak) =	709 lbs
Single Supply Bottle (Bare), 42.5 ft^3 @ 5000 psia =	2348 lbs
Weight of System per 90 lbs Transferred Excluding Insulation, Instrumentation, etc.	3057 lbs

Based on the above discussion and weight analysis, this system is not considered feasible and no further consideration was given to it.

4.1.2 TRANSFER BY HEATING SUPPLY FLUID. The system is shown schematically in Figure 4-3.

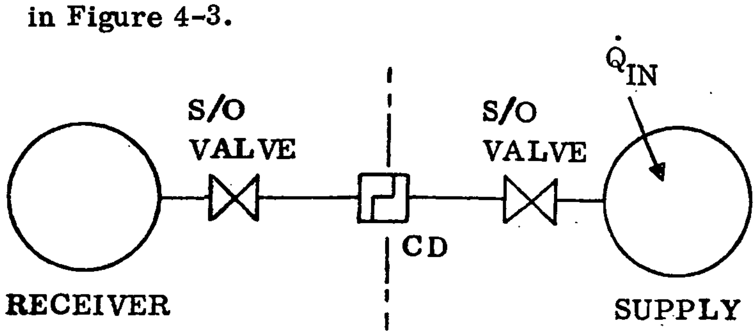


Figure 4-3. Schematic - Supply Heating System

Fluid is transferred from supply to receiver by pressure differential. The supply bottle pressure is maintained at a constant level by the addition of heat energy. Receiver bottle pressure is allowed to increase until it reaches the level of the supply pressure. Fluid transfer is terminated at this point.

Supply Bottle Volume	= 42.5 ft^3	= constant
Initial Supply Density	= 3.9 lb/ft^3	= constant
Receiver Bottle Volume	= 42.5 to 300 ft^3	= variable
Initial Receiver Density	= 0.1 to 0.5 lb/ft^3	= variable
Equivalent Orifice Dia of Transfer Line	= 0.06 to 0.12 in.	= variable
Heat Through Transfer Line	= 0	= constant
Heat Through Receiver	= 0	= constant
Heat to Supply Fluid = as required to maintain supply bottle pressure constant		= variable
Supply Bottle Pressure	= 225 psia	= constant

It is noted that analysis showed that the lower the supply pressure the lower the overall energy required for the transfer. This is also illustrated by the data presented in Ref. 4-2. However, in order to prevent instabilities in the supply tank pressure and control system, it is necessary to keep the pressure slightly above critical at all times. A value of 225 psia was chosen as a reasonable compromise between these two requirements for the hydrogen case.

Figure 4-4 shows the effect of receiver volume on total fluid mass transferred for the conditions stated. The required rate of heat energy addition to the supply fluid is

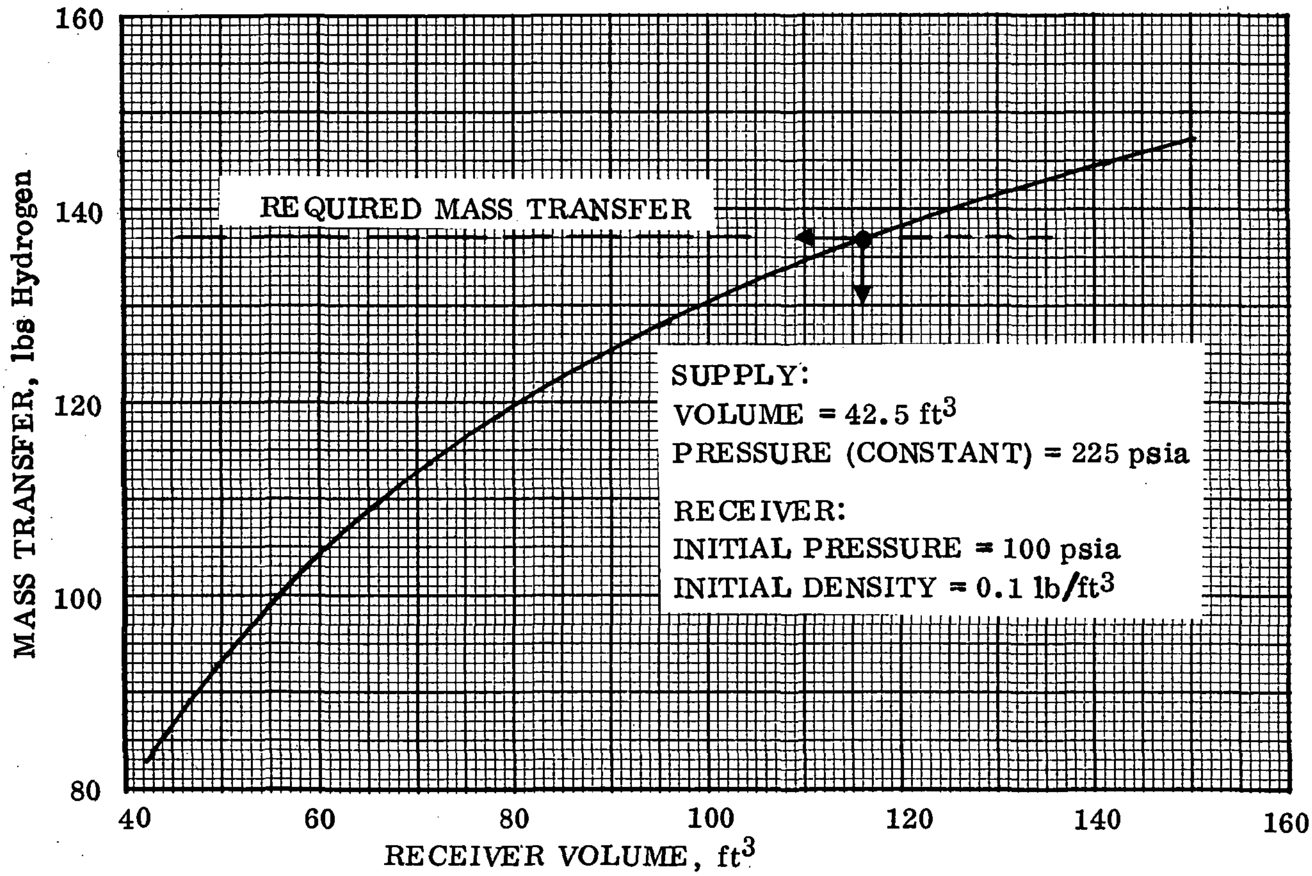


Figure 4-4. Supercritical Fluid Transfer by Heating Supply Bottle

dependent on the transfer line equivalent orifice diameter. Diameter increases require heat rate increases. The total energy required to transfer 137 lbs of hydrogen under the conditions stated in Figure 4-4 is constant at approximately 18,000 Btu. The peak rate of energy addition was approximately 4 Btu/sec for a 0.06 inch transfer orifice diameter. At this rate the elapsed time to transfer was approximately 3.5 hours.

Weight analysis for this system must consider the source of heat energy. If the required energy is obtainable from the receiver stations existing power sources or waste processes, weight of the energy source would be minimized. The following weight summary, however, assumes an additional fuel cell type power supply is required. Furthermore its weight is shared with seven other supply bottles. Fuel cell weight is assumed to be 94 lbs/KW + 2.9 lbs/KW-hr.

The system weight summary follows:

Single Supply Bottle (Bare), 42.5 ft^3, 225 psia	= 130 lbs
Single Receiver Bottle (Bare), 116 ft^3, 225 psia (design)	= 278 lbs
Receiver Insulation	= 300 lbs
Power Supply (see computation below)	= 65 lbs
Weight per 137 lbs Hydrogen Transferred (Excluding Instrumentation, etc.)	= 773 lbs

Power supply weight was determined as follows:

4 Btu/sec = 4.21 KW @ 94 lb/KW = 396 lbs × 1/8*	= 49.5 lbs
18,000 Btu = 5.27 KW-hr @ 2.9 lb/KW-hr	= 15.3 lbs
	64.8 lbs

*This assumes that one receiver is filled at a time.

Based on the above discussion and weight summary, this system is considered feasible and should be given further consideration.

4.1.3 TRANSFER BY PUMPING. The system is shown schematically by Figure 4-5. In this system fluid is first transferred from the supply to the receiver by simple blowdown. When pressures are equalized the by-pass valve is closed and the compressor started. Transfer is terminated when the state of the supply fluid becomes two phase or when the receiver is fully loaded.

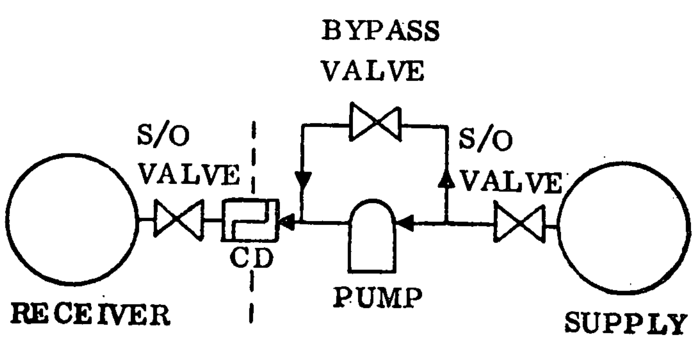


Figure 4-5. Schematic - Pumping System

The feasibility analysis for this method of transfer was accomplished under the following conditions.

Supply Bottle Volume	$= 100\ ft^3$	= constant
Initial Supply Density	$= 1.7\ lb/ft^3$	= constant
Equivalent Orifice Dia of Transfer Line	= 0.25 inch	= constant
Heat Through Transfer Line	= 0	= constant
Heat to Fluid	= 0	= constant
Initial Supply Pressure	= 1500 psia	= constant
Receiver Volume	$= 42.5$ to $100\ ft^3$	= variable
Initial Receiver Density	$= 0.2\ lb/ft^3$	= constant

The subroutine MACHINE was added to the Plumber code to compute work output required to compress the fluid from the supply bottle state to the receiver state.

Initial conditions of the supply bottle (volume, pressure, and density) were selected for an isentropic discharge of 140 lbs of hydrogen with the fluid state remaining gaseous. This would be a prime requirement for an actual system. The compressor was idealized as an electrically driven constant displacement device (constant speed piston type) operating at a rate of 1 ft^3/sec with a thermal efficiency of 85%.

The best system analyzed requires the compressor to put out approximately 600 Btu at a peak rate of 10.5 Btu/sec. At this rate the elapsed time to transfer was 5 minutes and the peak receiver bottle pressure was 1550 psia.

To reduce the size of the required electrical power supply, a smaller compressor and longer transfer time would be used in actual practice. In the weight summary presented below the electrical power supply is assumed to be the same as that used in Paragraph 4.1.2.

Weight summary:

Single Receiver Bottle (bare), 100 ft^3 @ 1500 psia	=	1479 lbs
Single Supply Bottle (bare), 100 ft^3 @ 1500 psia	=	1479 lbs
Compressor (Est. by Ref. 4-3 × 1/8)	=	30 lbs
Power Supply (Est. as in Para. 4.1.2)	=	50 lbs
Weight of System per 137 lbs Hydrogen Transferred (Excluding Insulation, Instrumentation, etc.)	=	3038 lbs

Based on a comparison of the above weight with the equivalent weight from Paragraph 4.1.2 this system is considered unfeasible.

4.1.4 TRANSFER BY HEATING AND PUMPING. The system schematic is presented in Figure 4-6. In this system, compression would be accomplished following the normal heated transfer in order to increase the final receiver pressure and reduce the receiver size. The system weight summary for the transfer of 137 lb of hydrogen is presented as a function of receiver volume in Figure 4-7. The weight breakdown is identical to that used for the supply heating system in Paragraph 4.1.2 except for the addition of 1/8th (pump shared by all 8 bottles) of the estimated pump weight. The summary, however, does not include the added weight of the electrical power supply for driving the pump.

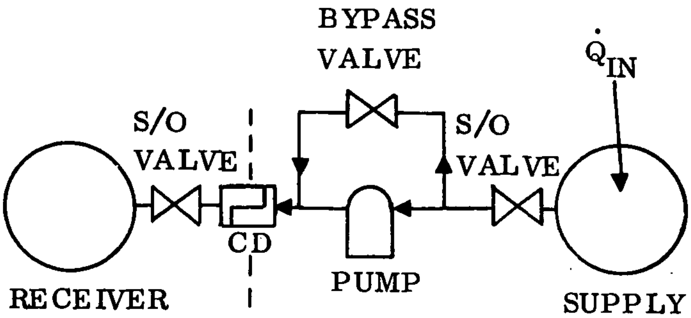


Figure 4-6. Schematic-Combination Pump and Supply Heating

Examination of Figure 4-7 shows that the minimum weight system of this type might be found with receiver volumes of approximately 104 to 105 cubic feet. However, by going to slightly larger volumes, the need for pumping is eliminated and the system becomes identical to that of Paragraph 4.1.2 which is about 30 lbs lighter and does not involve the complexity of the pumping system. The pumping system is therefore not worthy of further consideration in comparison to the heater-only system.

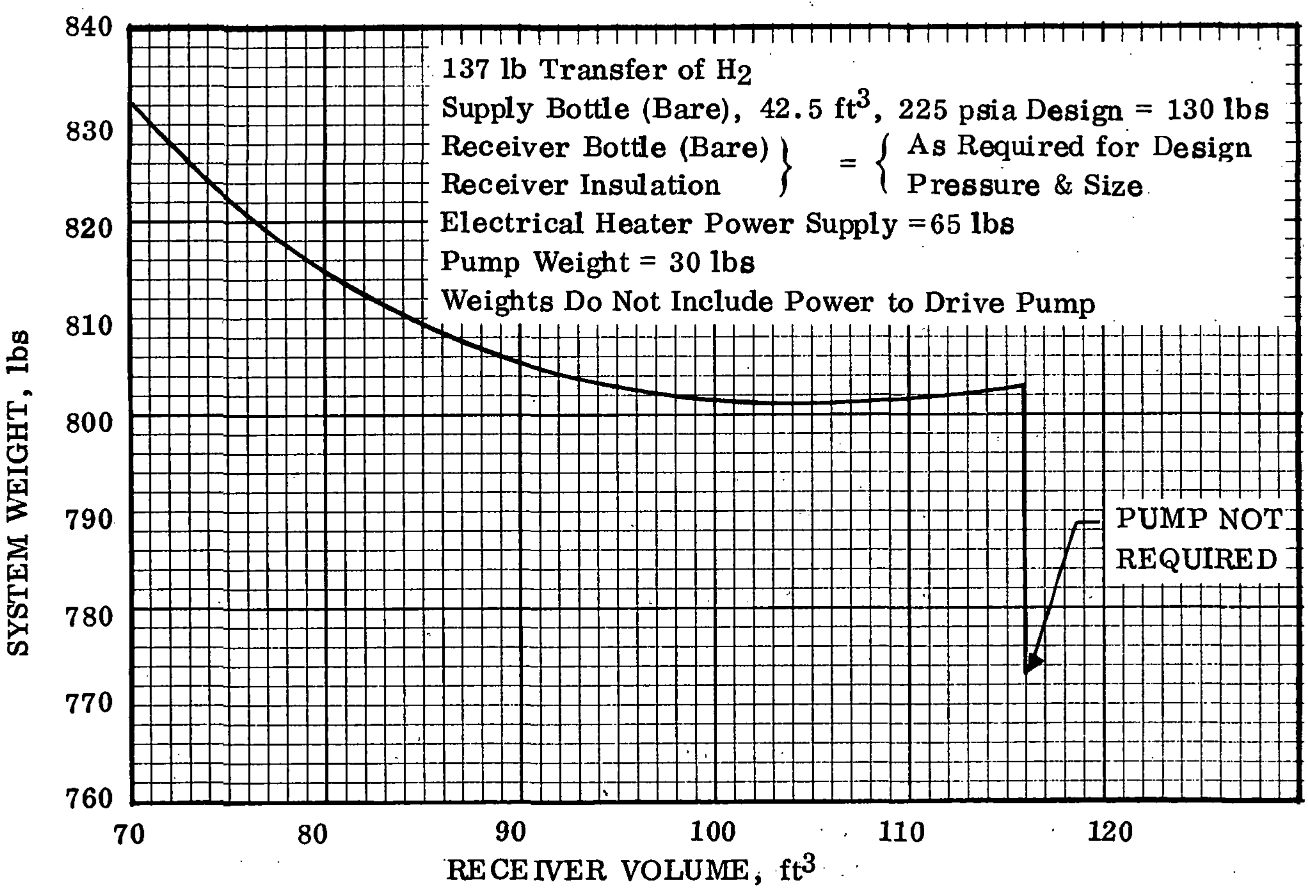


Figure 4-7. System Weight Summary for Combination Heated Supply and Pump System

4.1.5 TRANSFER BY HEATING SUPPLY AND COOLING RECEIVER. The system is shown schematically in Figure 4-8.

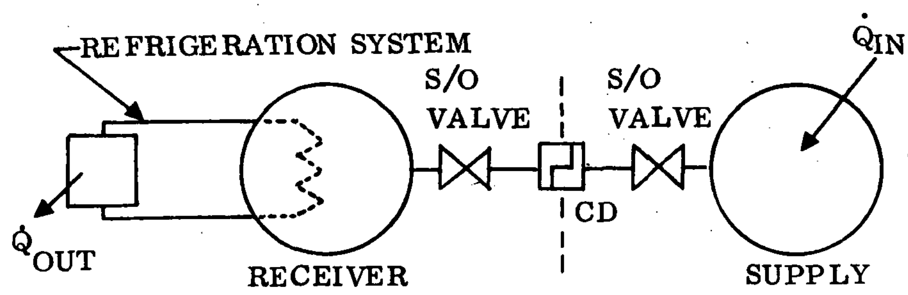


Figure 4-8. Schematic - Supply Heating/Receiver Cooling System

In this system fluid is transferred from the supply to the receiver by pressure differential. The absolute pressure levels in each bottle are maintained constant by adding heat energy to the supply and removing it from the receiver. Fluid transfer is terminated when the required mass transfer is achieved.

The feasibility analysis for this system was conducted under the following conditions:

Supply bottle volume =	42.5 ft^3 =	constant
Receiver bottle volume =	42.5 ft^3 =	constant
Supply bottle pressure =	250 to 400 psia =	variable
Receiver bottle pressure =	200 to 300 psia =	variable

Analysis indicates that a minimum of approximately 26,000 Btu addition to the supply bottle and 36,000 Btu removal from the receiver is required to operate this system. This occurs at the minimum pressure conditions (250 psia supply and 200 psia receiver). In order to accomplish the transfer within the allotted time (assumed to be 3 hours / bottle) the average rates of heat transfer would be 2.4 Btu/sec added and 3.4 Btu/sec removed. Peak rates are estimated from the computer outputs to be approximately twice the average or 4.8 Btu/sec and 6.8 Btu/sec respectively for addition and removal. According to the data presented in Reference 4-4, the weight of a space-borne closed cycle refrigeration system with a capacity to remove 6.8 Btu/sec at a low temperature of 38°R is prohibitively high. As an example, 1000 lbs of equipment are required for only 0.019 Btu/sec. The analysis indicates that it would be possible to increase the receiver bottle size and provide only a small amount of refrigeration. However, analysis indicated that the optimum condition would occur near the receiver size and weight of a system without refrigeration and the small weight reduction, if any, would definitely not offset the added complexity of the refrigeration system. This system is therefore considered undesirable for the present transfer application.

4.1.6 TRANSFER BY HEATING SUPPLY AND COOLING RECEIVER PLUS REGENERATION. The system is shown schematically in Figure 4-9.

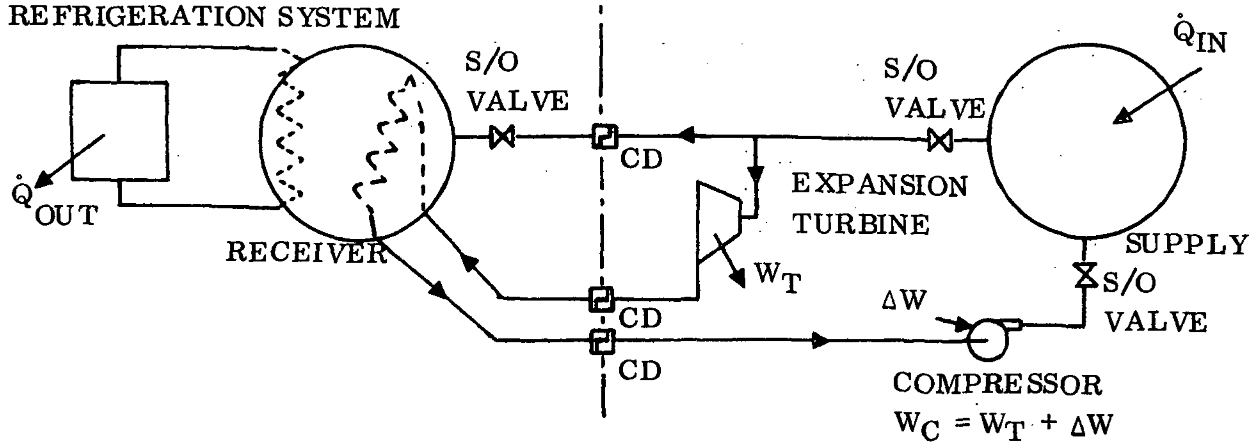


Figure 4-9. Schematic - Supply Heating/Receiver Cooling Plus Regeneration

In this system fluid is transferred from supply to receiver by a pressure differential. Pressure levels in each bottle are maintained constant by the addition and removal of heat energy as required. Fluid transfer is terminated when the required mass transfer is achieved.

This system is basically the same as the one discussed in Paragraph 4.1.5 with the addition of an expansion turbine, return compressor, and extra valves and disconnects. Even assuming the use of 100% efficient turbine and compressor, energy requirements for driving the system were reduced only approximately 10 percent by the regenerative setup. This makes no appreciable difference in the refrigeration requirements and therefore the refrigeration package weights remain prohibitively high. It is concluded that this is due to the enthalpy-temperature relationships associated with the hydrogen at near critical conditions; i.e., the expansion process (near isentropic pressure drop) relied on in the regenerative cycle to produce a temperature drop for subsequent heat transfer did not produce a large enough temperature change to significantly improve the overall system efficiency.

Based on the above discussion this system was not considered feasible and no further consideration was given to it.

4.1.7 TRANSFER BY HEATING SUPPLY WITH VORTEX TUBE ASSIST. The system is shown schematically in Figure 4-10.

In this system fluid is transferred by pressure differential. The supply bottle is maintained at constant pressure by adding heat from an external source and from the vortex tube's hot gas discharge. The hot gas discharge, in turn, is cooled as it is passed through the supply fluid and it is then mixed with the cold gas discharge of the vortex tube. This reduced temperature mixture is then transferred to the receiver. Transfer is terminated when the receiver pressure increases to that of the supply.

The vortex tube itself is a device having no moving parts, which is capable of separating one stream of gas into two streams, one at a higher and the other at a

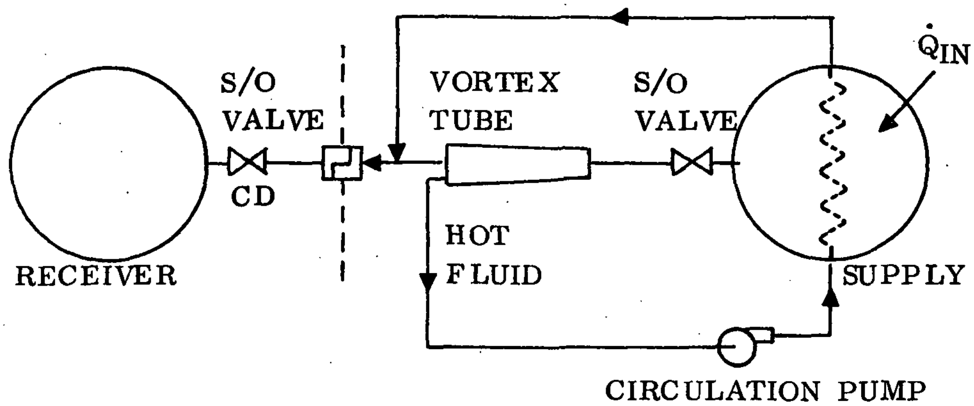


Figure 4-10. Schematic - Supply Heating Plus Vortex Tube Assist

lower temperature than the original stream. It was thought that a device with these characteristics, employed as shown in Figure 4-10, might allow a significant reduction in weight of the basic heated supply transfer system. Temperature differences on the order of 100°F between hot and cold streams have been obtained with air systems (Ref. 4-5).

A feasibility analysis of the Figure 4-10 system was made with emphasis on vortex tube performance. The analysis was based on information from References 4-5, 4-6, and 4-7. H. H. Bruun (Reference 4-5) concludes from experimental data that energy separation in the vortex tube is "mainly caused by adiabatic contraction and expansion of turbulent eddies in a centrifugal field." V. I. Metenin (Reference 4-6) corroborated with experimental data that the total heat removed from the cold gas stream equalled the heat added to the hot stream. Further, L. R. Inglis (Reference 4-7) indicates that the maximum cold side temperature drop is that obtained by isentropic expansion between the system inlet and outlet pressures. Applying the above information to supercritical hydrogen resulted in the following conclusions.

Expansion of hydrogen from a supply bottle at 225 psia thru a vortex tube to 150 psia, when the supply bottle is full, will result in a temperature drop of only 1°R for the cold stream discharge. When the supply bottle is nearly empty, the same expansion results (for the most idealized case) in a 7°R temperature drop for the vortex tube's cold stream, but because of the steep slope of the enthalpy lines the hot stream discharge is actually colder than the original inlet and thus could not be used to heat the supply bottle. These results indicate that use of a vortex tube will actually be a detriment to the performance of a basic heated supply transfer system operating near the critical pressure region. Analysis indicates that this will also be true for supercritical oxygen and nitrogen transfer.

4.1.8 OVERALL CONCLUSIONS. In summary the only system considered to be worthy of further analysis and data development was the simple supply heating system described in Paragraph 4.1.2.

4.2 DETAIL ANALYSIS AND DEVELOPMENT OF PARAMETRIC DATA

This section presents the analysis and optimizations performed on the heated supply transfer system chosen as the most promising under the Convair Aerospace IRAD program described in the previous section. The basic configuration analyzed is presented in Paragraph 4.1.2.

The following basic assumptions were made with regard to the initial parametric studies. The fluid to be transferred would be hydrogen and the minimum required transfer would be 137 lb per bottle. The heated supply bottle would be 42.5 cubic feet in volume. Its operating pressure would be as low as feasible in order to minimize bottle weight and yet provide sufficient margin to assure that the fluid state remained above saturation during the transfer process. The pressure selected was 225 psia. The supply bottle was assumed to be initially charged with LH_2 at one atmosphere to a 10% ullage and then allowed to self pressurize to 225 psia. The initial charge would then be 165 lbm at a density of 3.9 lbm/ft^3. One hundred psia was assumed to be a reasonable minimum pressure to supply a space station with hydrogen service, therefore, the pressure in the receiver bottle was assumed to be at 100 psia prior to the start of transfer. The volume and initial density of the receiver were not fixed at this point. Results of various trade-offs and analyses performed are presented in the following paragraphs.

4.2.1 EFFECTS OF RECEIVER VOLUME AND INITIAL DENSITY. The Plumber Computer code described in Appendix B was used to generate date for this analysis. A variety of initial densities were used with three different receiver volumes; 75, 100 and 125 ft^3. By interpolating between the data a minimum receiver size of 119 ft^3 was found to which 137 lbm of H_2 could be transferred. The data also indicated that transfer is maximized when receiver volume and initial density are maximized. The net mass transferred appeared to approach an asymptotic maximum at an initial density of approximately 0.2 lb/ft^3 for each receiver volume. Although it was not verified it is reasonable to assume that the net mass transferred with respect to increasing receiver volumes would approach an asymptotic maximum for the supply bottle conditions assumed.

Figure 4-11 shows the effect of initial receiver density on the transfer for a constant receiver volume of 125 ft^3. The figure is self-explanatory except for the mass vented for chilldown. Calculations showed that at low initial receiver densities the net (total minus initial) mass transferred could be increased (for fixed supply conditions) by venting some of the receiver fluid during the transfer. Near maximum net mass transfer was found to occur for the present case when the receiver tank pressure was limited to a maximum of 100 psia (by venting) until the

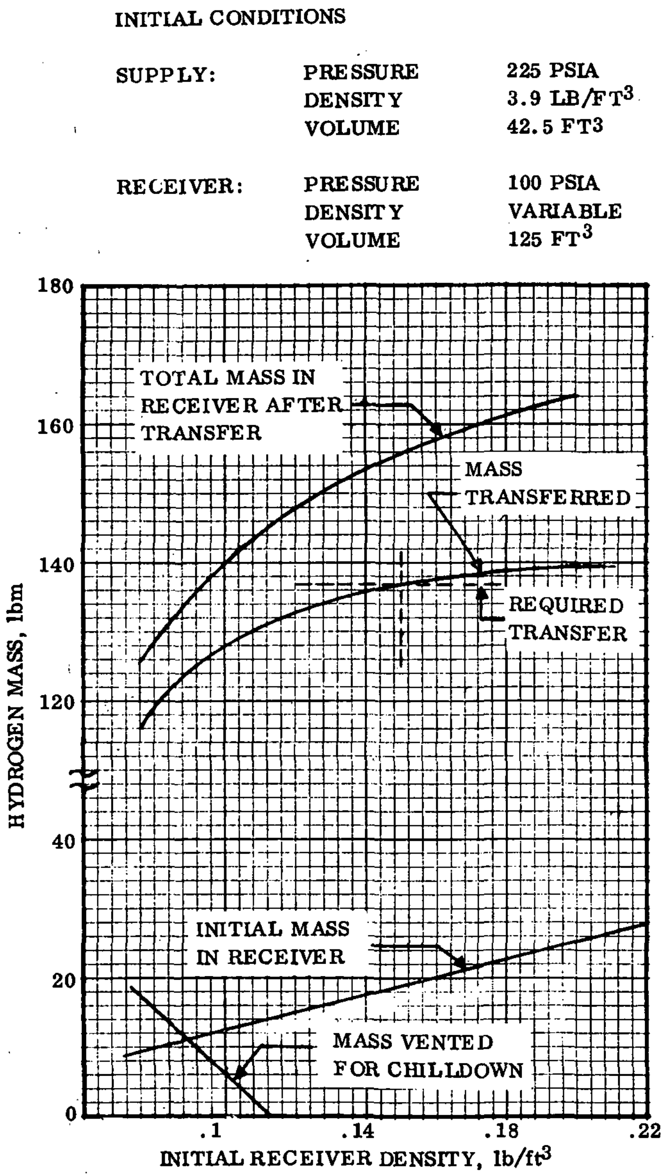


Figure 4-11. Supercritical Transfer as Function of Initial Receiver Density

point when the receiver fluid temperature approached to within a few degrees of the inflow fluid. The Figure 4-11 data were developed on this basis and the vent masses required are illustrated in Figure 4-11. It is noted that in all cases the pressure in the receiver at the beginning of the transfer process momentarily decreases below the initial pressure (100 psia in this case) before rising. This is due to the chilling of the receiver fluid by the initially cold supply fluid.

The venting process described is effective for maximizing the transfer because high energy residual fluid is removed and replaced by the denser, low energy fluid from the supply bottle. Figure 4-11 shows that the requirement for receiver venting goes to zero at an initial receiver density of 0.115 lb/ft^3 for the 125 ft^3 bottle. For higher initial densities, the receiver contents are chilled to the temperature of the incoming fluid before the pressure reaches 100 psia and therefore no venting is required for chilldown.

From Figure 4-11 it can be seen that the required transfer of 137 lbs of hydrogen can be met with an initial receiver density of 0.15 lb/ft^3 for the 125 ft^3 bottle. It is noted that if the bottle is used to a lower density, it cannot be fully recharged under the same supply conditions. The maximum recharge capability is approximately 139.5 lbs if the bottle is used to a density of 0.2 lb/ft^3 minimum.

For the purpose of further development of parametric data the standard receiver size was selected as 125 cubic feet and the minimum initial density as 0.15 lb/ft^3.

4.2.2 <u>EFFECTS OF VARIATIONS IN INITIAL RECEIVER TANK PRESSURE.</u> A second trade-off was performed with respect to determining the effect of variations in initial receiver tank pressure on the transfer process. Data from the Plumber Computer program were used to generate a curve of the estimated receiver volume required to transfer 137 lbs of hydrogen as a function of initial receiver pressure (Figure 4-12).

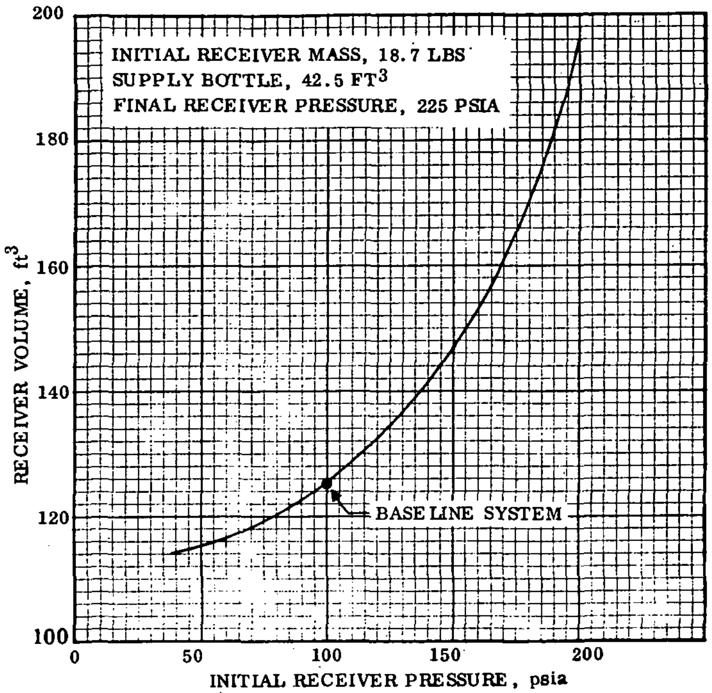


Figure 4-12. Required Receiver Volume as a Function of Initial Receiver Pressure for Transferring of 137 Lbs H_2

This data assumed the initial mass in the receiver at the start of transfer was constant for all receiver sizes. The supply bottle was constant at 42.5 ft^3 and 225 psia. Receiver system weights, bottle weight, insulation weight and total weight are given in Figure 4-13 as a function of bottle volume. Tank weights were obtained from the data in Appendix A for aluminum tanks. Receiver tank insulation weights were estimated on the basis of the ground rule requirement which says that the heat leak to a 42.5 ft^3 hydrogen bottle be limited to a value which would boil-off 50% of the contents in 180 days. As the receiver tank size is changed this limited amount of mass vented is maintained constant and thus the insulation thickness and weight increases with increasing tank size as the surface area increases. This is only true for the case where different receiver sizes are being considered for the same fluid transfer quantity. Insulation performance and weight data for Superfloc as described in Section 4.1 were used.

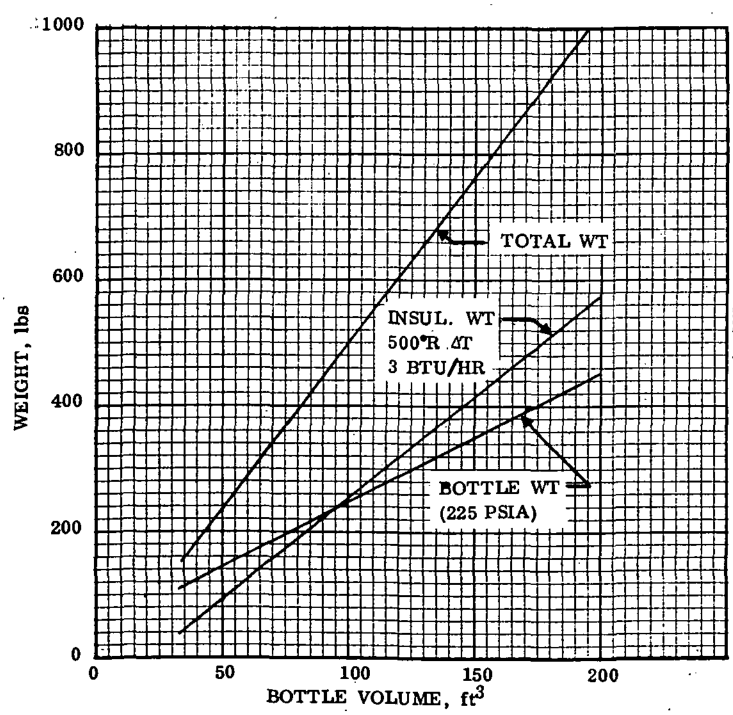


Figure 4-13. Estimated Receiver Weights Vs Receiver Volume

Figure 4-14 showing the effect of initial receiver pressure on receiver system weight was generated from the information on Figures 4-12 and 4-13. This weight is for one receiver bottle with insulation. Figure 4-14 illustrates the importance of initial receiver bottle pressure and that the minimum pressure possible is desirable.

This pressure could be minimized prior to refill by rapidly venting the remaining fluid. A previous analysis associated with a pure blowdown transfer system indicated, however, that it does not pay to throw away receiver fluid to obtain the low pressure. It would be better to use the receiver fluid to a low pressure as part of the normal station usage requirements. From the baseline system

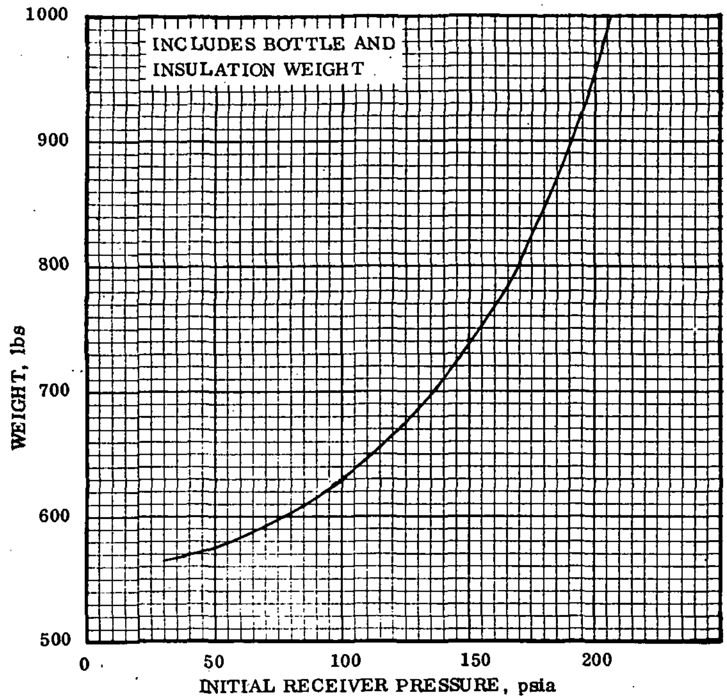


Figure 4-14. Estimated Receiver Weight as Function of Initial Receiver Pressure for Transfer of 137 lb of H_2

data presented in Section 2 it is assumed that 100 psia is a reasonable minimum pressure required to supply a space station with hydrogen service. Therefore for the purpose of further study, the minimum pressure at the initiation of refill was taken to be 100 psia.

4.2.3 EFFECTS OF TRANSFER RATE. The third trade-off performed was designed to show the effect of the rate of transfer on the refill process. The rate of transfer is controlled by a fixed diameter orifice located in the transfer line at the inlet to the receiver bottle. The supply and receiver bottles were assumed to be as defined by the first two analyses previously discussed (supply 42.5 ft^3, receiver 125 ft^3). The results of the present tradeoff are shown in Figure 4-15. The data was based on filling 8 hydrogen bottles in sequence. The data shows that the rate of transfer has a significant effect on the peak energy addition rate required to maintain the supply bottle at constant pressure. Rate of energy addition in turn has a significant effect on the weight of the electrical power system which must supply the energy. Weight of the electrical supply is also shown in Figure 4-15. This data indicates that the rate of transfer should be minimized (transfer time maximized) in order to minimize total system weight.

The data also indicates that the rate of transfer has no effect on the total mass transferred nor the total energy required to heat the supply bottles. This neglects the effect of additional external heat leak through the insulation (2 to 3 Btu/hr), which compared to the total energy added to the system from electrical sources (18,000 Btu/bottle) is considered negligible.

The longest transfer time allowable was then chosen as optimum, and for a 24 hour total transfer time, assuming an overall total of 12 bottles (8-H_2, 2-O_2, 2-N_2), the maximum allowable time per bottle would be 2 hours (7200 sec). The suggested transfer time presented in Figure 4-15 (6500 sec) allows 10% (12 minutes per bottle) for various hook up and handling operations associated with the transfer.

4.2.4 VARIATIONS IN SUPPLY TANK VOLUME. The fourth trade-off study was directed toward defining the effect of variations in supply bottle size on the transfer system. Figures 4-16 and 4-17 were generated with a constant volume receiver (125 ft^3) depleted to a density of 0.15 lb/ft^3. Supply bottles were assumed charged

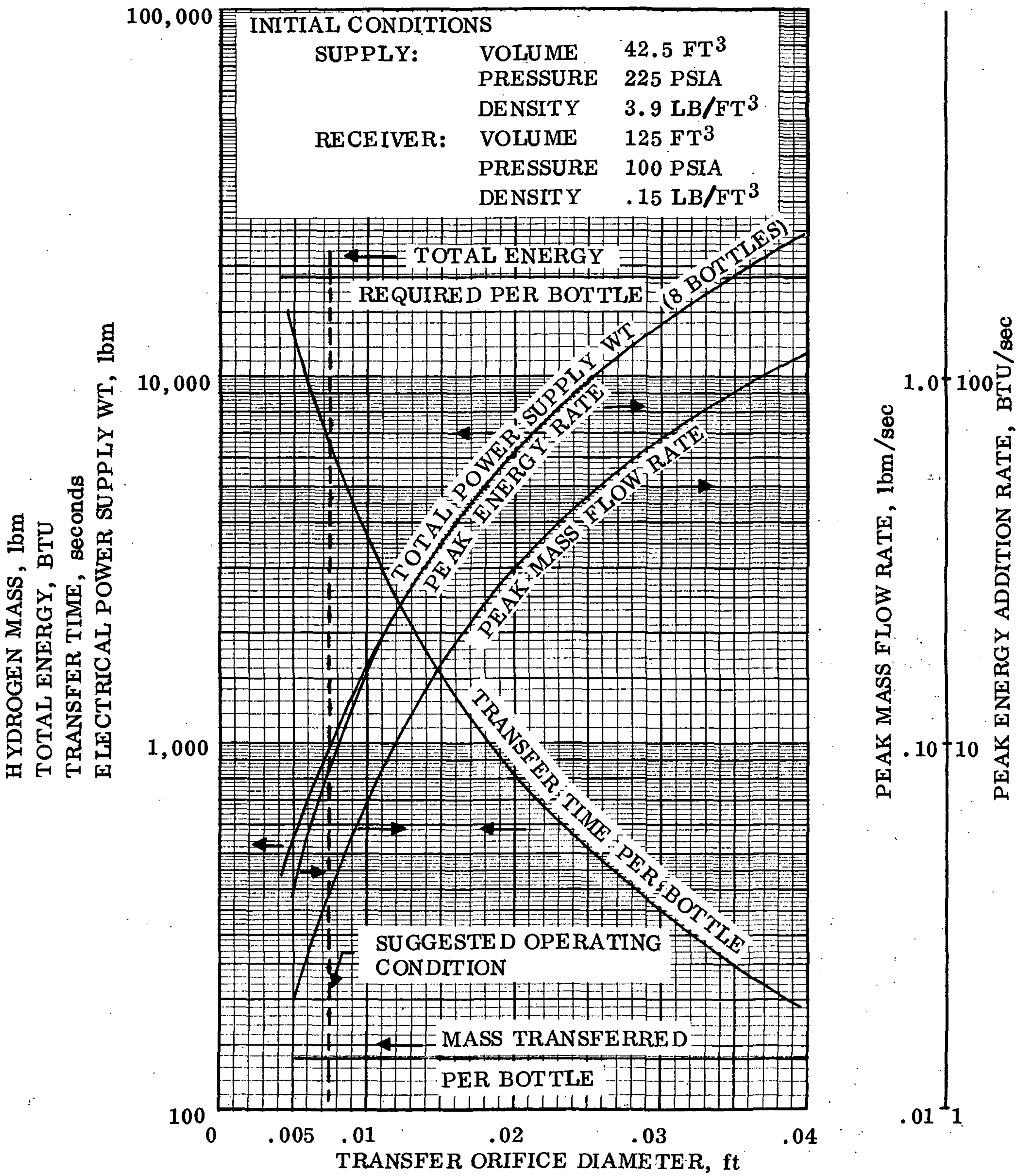


Figure 4-15. Effect of Flow Orifice Size on the Supercritical Transfer Process

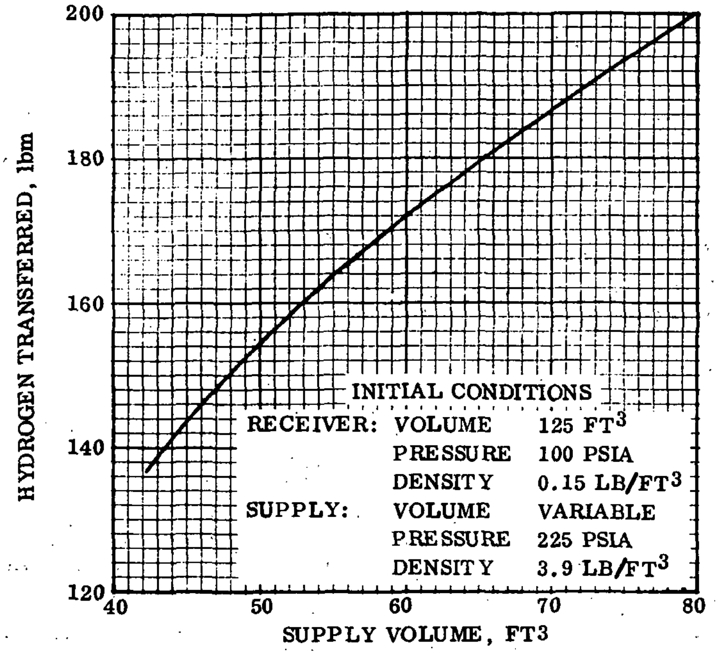


Figure 4-16. Effect of Supply Volume on Transferred Quantity of H_2 Mass

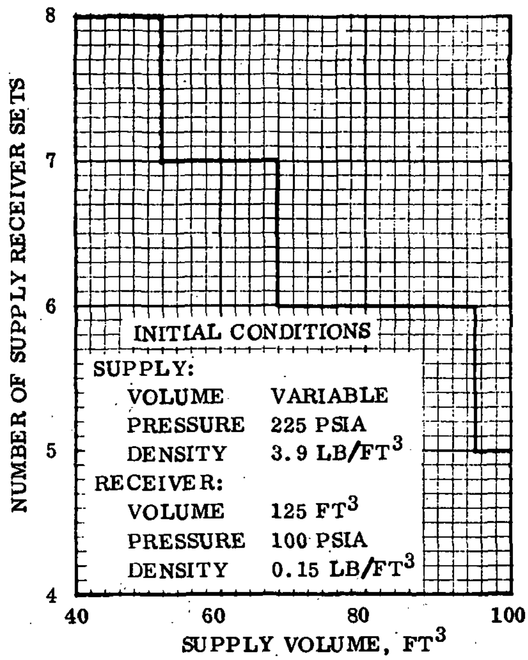


Figure 4-17. Effect of Supply Bottle Size on No. of Sets of Supply Receivers Required for Transfer of 1096 Lb of H_2

to 225 psia and 3.9 lb/ft^3 density regardless of size. Figure 4-16 shows how net mass transferred to the 125 ft^3 receiver is affected by increasing supply bottle sizes. Figure 4-17 shows the total number of supply and 125 ft^3 receiver bottle sets required to enable resupply of the required 1096 lb (baseline case, Section 2) of hydrogen as a function of supply bottle volume. Figure 4-18 was generated to show the permissible decrease in receiver volume as supply bottle size increases for transfer of 137 lb of hydrogen per bottle. This type of transfer requires 8 supply and 8 receiver bottles to transfer the 1096 lb of hydrogen.

Supply system weights, bottle weight, insulation weight and total weight are given in Figure 4-19 as a function of bottle volume. The tank weights were obtained from the data in Appendix A for aluminum tanks. Supply tank insulation weights were estimated on the basis of permitting the pressure of the supply bottle, filled to 10 percent ullage at 15 psia, to rise to 225 psia in 7 days. The use of "Superfloc" HPI was assumed.

4.2.5 OPTIMUM TANK VOLUMES. This trade-off was performed to assess the relative importance of supply weights versus receiver weights and thus define reasonably optimum tank volumes. It was assumed that costs are directly proportional to launch weight and that full resupply operations for a space station are required every six months over a 10 year period (20 times). This makes the supply weights 20 times more important than the receiver weights, assuming the receiver system is not required to be replaced over its 10 yr. life. Figure 4-20 was generated using the information from Figure 4-18, 4-19 and 4-13. Each launch is assumed to

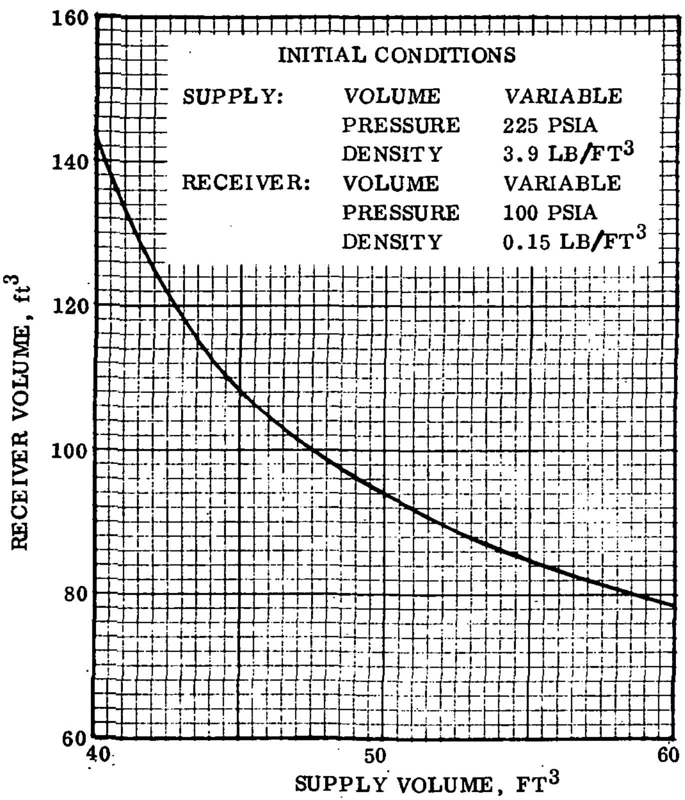


Figure 4-18. Receiver Volume Vs Supply Volume for Transfer of 137 Lbs of H_2

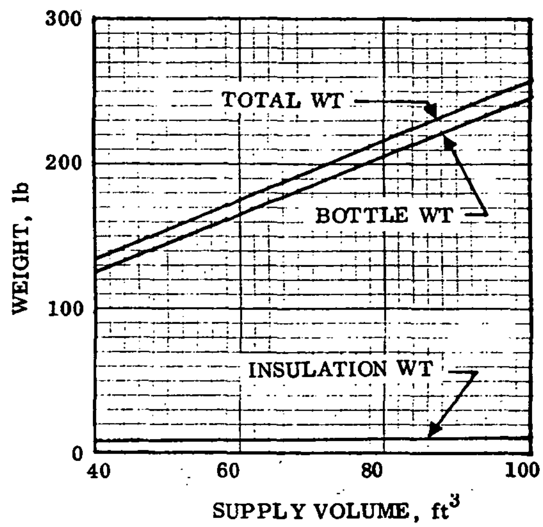


Figure 4-19. Estimated Supply Weights Versus Supply Volume (Hydrogen)

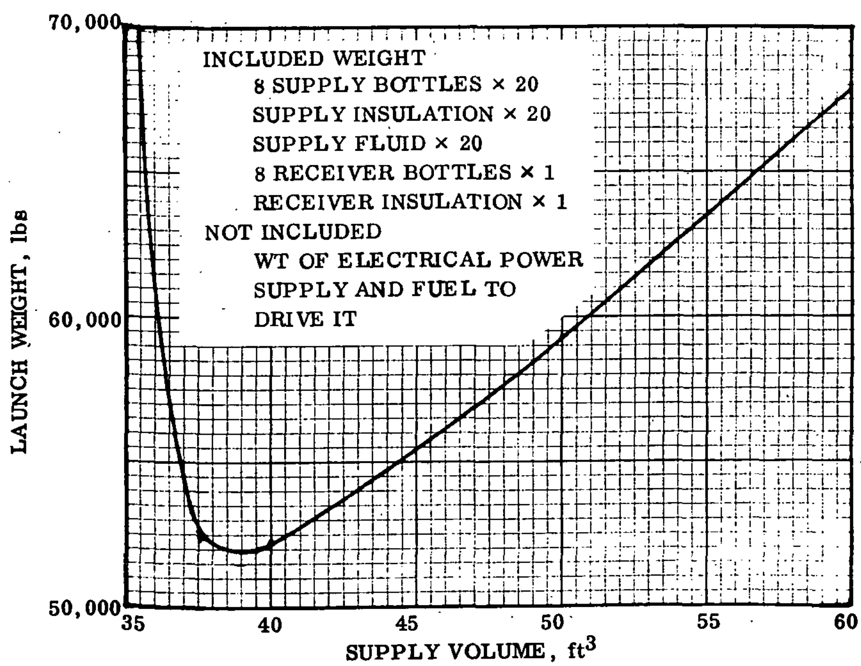


Figure 4-20. Supercritical Transfer Launch Weight Penalty for 20 Resupply Missions, Each of 1096 Lbs of Hydrogen

carry 8 full bottles (charge density 3.9 lb/ft^3) to a space station with 8 receivers depleted to a density of 0.15 lb/ft^3. The space station receiver bottles were assumed to be empty when initially launched. Figure 4-20 shows that for the assumed conditions an individual supply bottle volume of 39 ft^3 would be optimum to transfer 137 lb of hydrogen. From Figure 4-18 the corresponding supply volume would be between 150 and 160 ft^3.

The electrical power supply for heating the supply bottles, or the fuel to drive it, was not considered in Figure 4-20 since the weight is constant for a given transfer mass and transfer time. It is noted, however, that cost would obviously be minimized if the fixed weight items (hardware) associated with the electrical supply were a permanent part of the space station. Thus the equipment need only be launched once and can be used for other space station service when propellant is not being transferred. The fuel consumed for driving the electrical supply during transfer operations, regardless of its source (space station or launch vehicle supplied) must be launched from the ground. In actual practice the supply bottles would probably be oversized to handle the fuel consumption requirements during transfer.

In order to illustrate the effect of different assessments between the importance of supply and receiver weights, data are presented in Figure 4-21 for the case where equal importance is given to both supply and receiver weights. In this case the optimum supply bottle would be 60 ft^3 with a corresponding receiver of 78 ft^3.

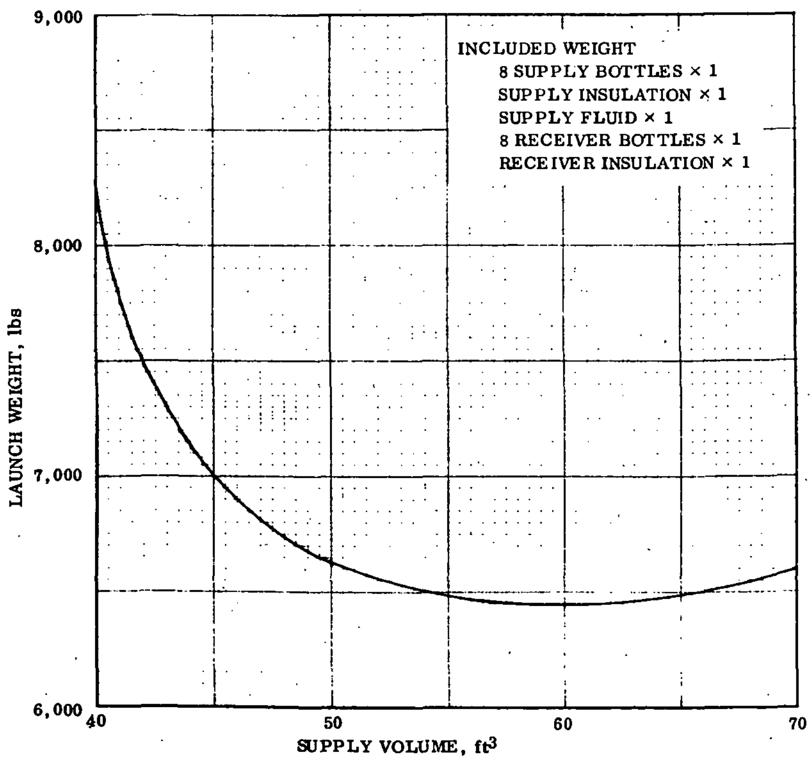


Figure 4-21. Supercritical Transfer System Weight for Single Supply of 1096 Lb of Hydrogen

4.2.6 <u>FILLING OF PARTIALLY FULL RECEIVER.</u>

This analysis was performed to determine if the standard system, as defined in the first two paragraphs of this section, was capable of refilling a half full receiver. Before this could be accomplished, a definition of the state of the fluid when the tank is half full was needed. Therefore, a pressure/density schedule for fluid expulsion from the receiver was developed. Items considered in the development of the schedule were:

a. The space station is assumed to operate with supercritical fluid.

b. Bottle pressure must be minimized so that refill at any density can be maximized. However, the pressure must be maintained at levels sufficient to avoid formation of two fluid phases (liquid and vapor).

c. Minimum bottle pressure must be of sufficient magnitude to permit adequate flow service to space station equipment. This is assumed to be 100 psia as in previous analyses of this section.

d. Fluid usage should be to minimum reasonable residuals. This has been defined in previous analysis as 0.15 lb/ft^3 density at 100 psia.

e. Ample margin must be left between the operating schedule and the saturated vapor line for control system operation such that the fluid state does not become two-phase.

Figure 4-22 presents a representative schedule of usage for the hydrogen receivers which meets the above criteria. The charging data are based on the use of 125 ft^3 receiver bottles. From Figure 4-22 the worst case (highest initial pressure) condition for refill of a half full bottle is determined to be: pressure, 170 psia and density, 0.7 lb/ft^3. It was further assumed that, regardless of the actual fluid quantities remaining aboard the space station, the supply bottles would be launched with a full charge. This

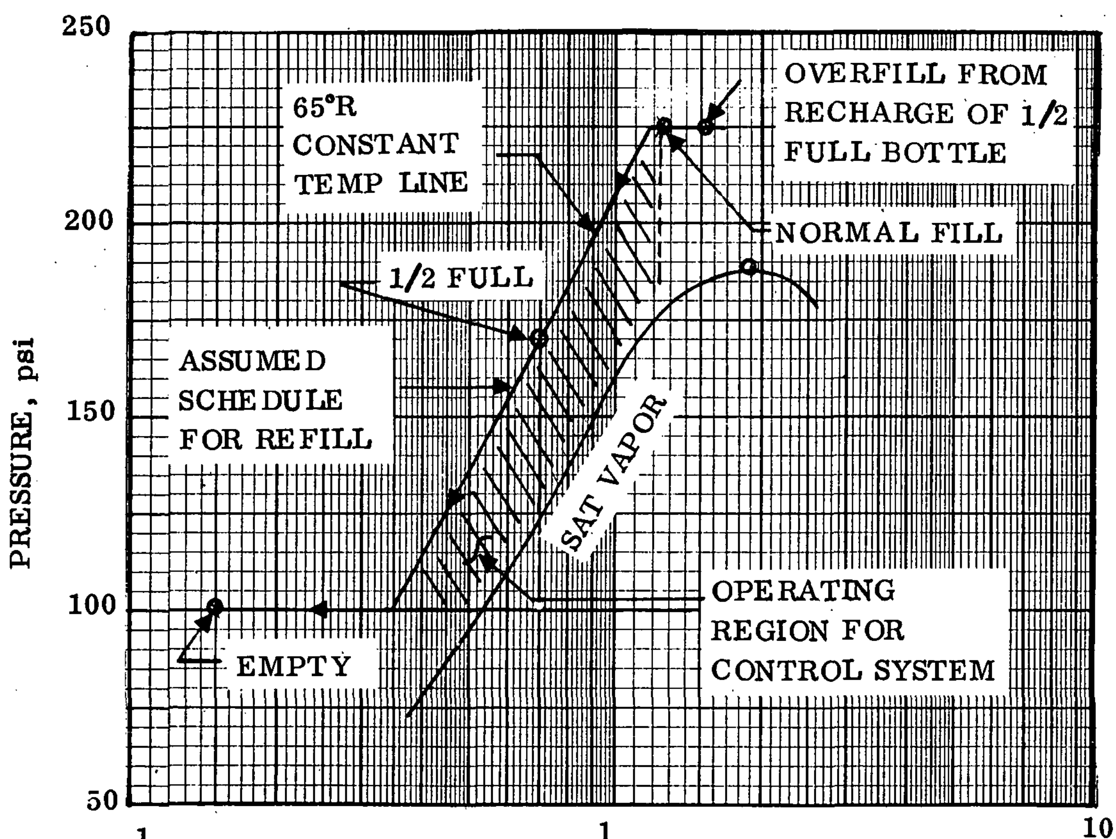


Figure 4-22. Usage Schedule for Station H_2 Storage Bottles

information was input to the Plumber Computer program which analytically performed the fluid transfer. The results indicated that 106.5 lbs of hydrogen were transferred to the receiver. This shows that, for the conditions assumed, a half full receiver can be refilled and in fact will be over-filled by approximately 38 lb of hydrogen.

4.2.7 HIGH PRESSURE O_2 TRANSFER. In addition to the previously discussed work on supercritical hydrogen transfer, a feasibility analysis was performed for the transfer of supercritical oxygen. Assumptions used in this work were based on reasoning identical to that presented in the hydrogen transfer analyses. The supply system was assumed to consist of 2 bottles, 25 ft^3 in volume, charged to 775 psia with a density of 63 lb/ft^3. The receiver selected for the initial analysis was 75 ft^3, initially at 100 psia. Supply and receiver volumes were chosen to provide the same ratio of filled to transferred mass as used for the H_2. The initial receiver density was permitted to vary. The transfer orifice was assumed to be 0.0075 ft in diameter. Total mass transferred for the 2 bottle system was required to be 2480 lbs of oxygen.

Figure 4-23 was generated, for a single set of bottles, from data obtained with the Plumber Computer program as input with the above assumptions. Energy consumption and time information are also listed on the figure. The data indicate that the maximum transfer will occur with initial receiver densities in the range of 1.4 to 1.6 lb/ft^3. This maximum is about 250 lb more than the required transfer which indicates that the receiver and supply are slightly oversized. No attempt was made during this analysis to optimize the size of the bottles. The total energy and rate of energy consumption required to accomplish the transfer of 1240 lbs of oxygen are of interest because of their magnitude. From Equation 3-5, the weight of an electrical power supply sized to provide 182 Btu (53.2 Kw-hrs) of energy for oxygen transfer to both receivers at a peak rate of 35.6 Btu/sec (37.5 Kw) would weigh approximately 3,674 lbm. Furthermore, this estimate is considered low because the time to transfer was considerably longer than the allotted 6500 seconds as discussed in Paragraph 4.2.3. In order to reduce transfer time, the rate of transfer must be increased. This in turn will require a higher rate of energy addition which will cause the power supply weight to increase. Energy addition rate for a 6500 second transfer is estimated to be 49 Btu/sec (51.8 Kw). This results in a power supply weight of 5,004 lbs. A comparison of the electrical supply requirements for hydrogen and oxygen transfer is given below:

	Space Station Fixed Weight (Based on Rate Capacity)	Launch Vehicle Added Weight for Total Energy Capacity (Fuel Consumed)	Total
Oxygen	4850 lbs	154 lbs	5004 lbs
Hydrogen	790 lbs	122 lbs	912 lbs

It is obvious from this analysis that oxygen transfer would size the electrical system requirements for the total transfer (assuming N_2 transfer does not require a peak energy rate greater than 49 Btu/sec). In order to effect a complete resupply of H_2, O_2

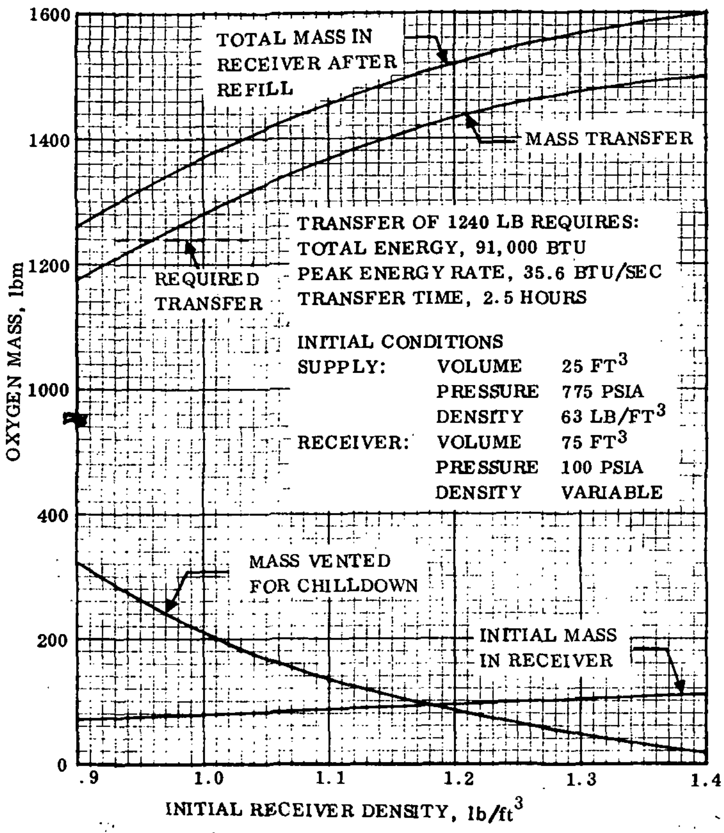


Figure 4-23. Supercritical Oxygen Transfer Process as Function of Initial Receiver Density

and N_2 such an electrical system would weigh 4850 lb with approximately 425 lb additional fuel required for each resupply mission (assuming N_2 transfer requires about 149 lb of fuel).

Figures 4-24 and 4-25 give representative usage schedules for supercritical space station O_2 and N_2 bottles which are to be resupplied from a high pressure source. These schedules are based on the same assumptions and criteria as used for the hydrogen system in developing the data for Figure 4-22.

It is noted that in the analyses discussed in this and preceeding paragraphs the emphasis was on demonstrating basic high pressure system feasibility in comparison with other systems, and pressure control details as to instabilities. temperature stratification and fluid mixing were not studied. Such details would be part of a final system development if such a high pressure system were to show sufficient promise to proceed to this point. The present energy analyses were all based on the assumption of a homogeneous fluid mixture in both supply and receiver tanks.

Also, all work was done with respect to filling bottles which are used for supercritical supplies to the space station functions. A gross energy analysis was performed to determine the feasibility of filling or resupplying station bottles where the fluid must be stored as a liquid for use in the station. This analysis showed that in this case a refrigeration system would need to be used to condense the transferred fluid and the energy and hardware requirements were excessive and the system considered impractical. This was found to be true for all the fluids studied (H_2, O_2 and N_2).

4.2.8 HYDROGEN SYSTEM SCALING EQUATIONS. In order to allow a determination of system weights for a large range of mass transfer, the scaling equations presented below were developed.

$$\text{Receiver Volume, ft}^3 = \frac{H_2 \text{ mass transferred, lb}}{1.1} \qquad (4\text{-}1)$$

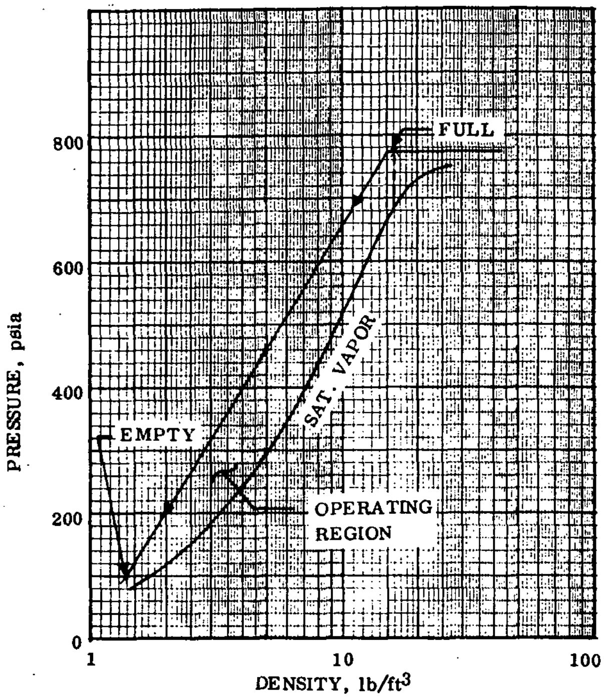


Figure 4-24. Usage Schedule for Station O_2 Storage Bottles

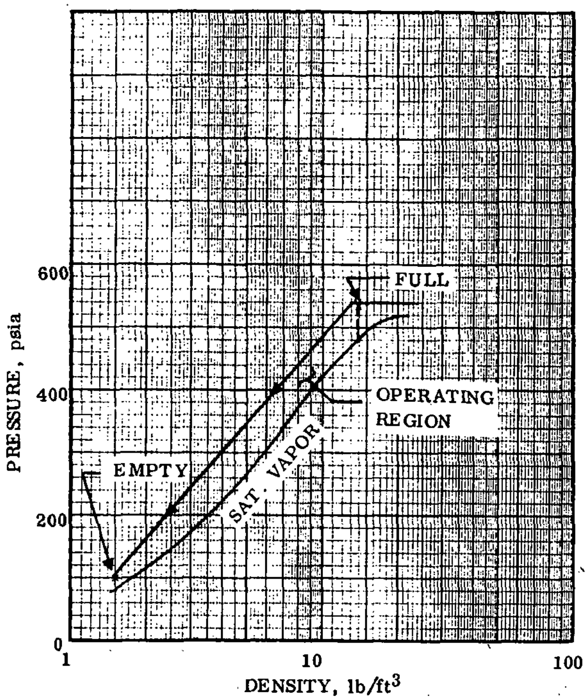


Figure 4-25. Usage Schedule for Station N_2 Storage Bottles

$$\text{Supply Volume, ft}^3 = \frac{\text{Receiver Volume, ft}^3}{2.94} \tag{4-2}$$

$$\text{Power Supply Weight, lb} = 10.4\ (\text{H}_2 \text{ Transfer Rate, lb/hr}) + 0.115\ (\text{H}_2 \text{ Transferred, lb}) \tag{4-3}$$

The above equations are for hydrogen only and are based on scaling the information contained in the previous paragraphs to larger sizes and mass transfers. Use of a 42.5 ft^3 supply and a 125 ft^3 receiver to transfer 137 lb of hydrogen was taken as the base. In developing Equations 4-1 and 4-2, initial and final fluid densities in both supply and receiver were assumed constant at the values presented below.

	Receiver	Supply
Initial Density, lb/ft^3	0.15	3.90
Final Density, lb/ft^3	1.25	0.67

Fixing of these densities automatically fixes the size of receiver required for a given mass transferred and also the relation of supply to receiver volume.

Equation 4-3 is based on the use of fuel cells per Equation 3-5, and base power requirements from Figure 4-15, with the assumption that the total energy required is directly proportional to the mass transferred and the peak rate of energy required is directly proportional to the average mass transfer rate.

5

SUBCRITICAL SYSTEM DEFINITION AND ANALYSIS

The work presented in this section was performed under the Convair Aerospace 1971 Independent Research and Development (IRAD) program (Reference 4-1) and is reported herein for reference as it relates to the present propellant transfer study.

Detailed system definition, analysis and development of parametric weight data were accomplished with respect to the most promising subcritical systems as determined by the screening study presented in Section 3. In addition to the basic supply or liquid orientation and collection schemes presented in Section 3, the complete transfer system is taken to include requirements for thermal control, pressurization and transfer line and receiver tank chilldown. Analyses associated with these requirements or auxiliary systems are presented in Sections 5.1, 5.2 and 5.3 respectively. Detailed system definitions and analyses for surface tension, bellows, metallic diaphragm and paddle vortex liquid orientation and collection systems are presented in Sections 5.4, 5.5, 5.6 and 5.7.

5.1 GENERAL THERMAL ANALYSIS

The primary requirement for thermal control of the subcritical supply systems occurs between the time of cryogenic loading and the beginning of actual transfer in space. Based on the data contained in Section 2,a reasonable maximum for this time is taken to be seven days.

It is noted that the purpose of the analysis presented in this section is not to define optimum thermal protection systems but only to determine reasonable and/or likely storage bottle heating, evaluate various modes of operation such as use of a locked-up versus a vented tank and determine approximate insulation weight penalties. Detailed thermal protection systems design as to optimum insulation materials, fabrication, installation, tank supports and fluid penetrations and the trade off between use of vacuum jacketed dewars versus a purge bag type insulation system is not within the scope of the present study. It is assumed that other programs such as that being conducted under NAS8-27419 (development of a purge type insulation system) will provide the data to optimize any final operational transfer system design.

Also, the present analysis is designed to apply generally to the various subcritical systems analyzed in Sections 5.4 through 5.7 and any thermal considerations special to these systems will be covered in those sections.

An important tradeoff when considering the requirement for a seven day storage is between use of a vent system to control tank pressure and the allowance of a locked-up tank without venting. In the case of the locked-up tank, a weight penalty is paid for the requirement to design the tank to a higher pressure, but the complexity of a low-g vent system is eliminated. In this comparison, the final tank pressure at which transfer occurs is an important parameter, since the tank must be designed for this pressure and allowance of a pressure rise to this value during the seven day period would not result in additional tank weight penalty.

Pressure rise rates for a locked-up tank without mixing are determined from the equations of Reference 5-1 as presented below for hydrogen and oxygen. Data are not available for nitrogen and it is assumed that this fluid will behave similar to oxygen.

$$\left[\frac{\Delta P}{\Delta \theta}\right] \frac{\text{psi}}{\text{hr}} = 86 \left[\frac{\dot{Q}_{IN} \ \text{Btu/hr}}{M_T \ \text{lb} \ S \ \%}\right]^{0.975} \qquad H_2 \qquad (5\text{-}1)$$

$$\left[\frac{\Delta P}{\Delta \theta}\right] \frac{\text{psi}}{\text{hr}} = 1450 \left[\frac{\dot{Q}_{IN} \ \text{Btu/hr}}{M_T \ \text{lb} \ S\%}\right]^{1.14} \qquad O_2 \qquad (5\text{-}2)$$

Pressure rise rates for a mixed tank from the same Reference are;

$$\left[\frac{\Delta P}{\Delta \theta}\right] \frac{\text{psi}}{\text{hr}} = 1.15 \left[\frac{\dot{Q}_{IN} \ \text{Btu/hr}}{M_T \ \text{lb}}\right] \qquad H_2 \qquad (5\text{-}3)$$

$$\left[\frac{\Delta P}{\Delta \theta}\right] \frac{\text{psi}}{\text{hr}} = 3.58 \left[\frac{\dot{Q}_{IN} \ \text{Btu/hr}}{M_T \ \text{lb}}\right] \qquad O_2 \qquad (5\text{-}4)$$

Initial calculations were made for a 42.5 ft^3 tank using hydrogen with a design pressure of 100 psia. Assuming an initial tank pressure of 20 psia, following ground loading, and a final allowable pressure of 100 psia, at the start of transfer following a seven day storage, from Equations 5-1 and 5-3 the allowable heat loads for a locked-up tank with unmixed and mixed fluid are respectively;

$$\dot{Q}_a \ \text{Btu/hr} = .00485 \ (M_T \ \text{lb}) \ (S \ \%) \qquad \text{No Mixing} \qquad (5\text{-}5)$$

$$\dot{Q}_a \ \text{Btu/hr} = 0.414 \ (M_T \ \text{lb}) \qquad \text{Mixing} \qquad (5\text{-}6)$$

The total mass of fluid in the tank (M) and percent ullage (S) can be put in terms of initial fluid densities and ullage fraction (F_V) as follows.

$$M_T = V_t \left[\rho_L (1-F_V) + \rho_V F_V \right] \qquad (5\text{-}7)$$

$$S = 100\ F_V \qquad (5\text{-}8)$$

Substitutions into Equations 5-5 and 5-6 result in;

$$\dot{Q}_a = .485\ V_t\ F_V\ [\rho_L(1-F_V) + \rho_V\ F_V\ ] \qquad \text{No Mixing} \qquad (5\text{-}9)$$

$$\dot{Q}_a = 0.414\ V_t\ [\rho_L\ (1-F_V) + \rho_V\ F_V\ ] \qquad \text{Mixing} \qquad (5\text{-}10)$$

Taking ρ_V = 0.11 and ρ_L = 4.34 lb/ft^3 for H_2 at a saturation pressure of 20 psia with F_V = 0.05 and V_t = 42.5 ft^3 results in the following values for allowable heat leak;

$$\dot{Q}_a = 4.25\ \text{Btu/hr} \qquad \text{No Mixing}$$

$$\dot{Q}_a = 72.6\ \text{Btu/hr} \qquad \text{Mixing}$$

An investigation was made into the possibility of using a simple foam insulation. Such an insulation would have typical values of thermal conductivity of 0.012 Btu/hr-ft-°F. Required thickness of such an insulation is estimated from the following equation;

$$\dot{Q}_i = \frac{k_i\ A_s\ \Delta T_i}{t_i} \qquad (5\text{-}11)$$

For a ΔT_i of 500°F and for the 42.5 ft^3 tank with A_s = 59 ft^2 and taking the mixed case with $\dot{Q}$ = 72.6 Btu/hr then t_i = 58.5 in. which is of course excessive. Considering a case where venting is accomplished and a 12 inch thick insulation is used, then from Equation 5-11;

$$\dot{Q}_i = 354\ \text{Btu/hr}$$

and for λ = 190 Btu/lb, the required boiloff in 7 days (168 hours) would be 313 lb which is over 100% of the tank contents. Therefore, the use of foam type insulations without high performance insulation (HPI) is not practical for the present application.

Thus for subsequent analyses performed with respect to the subcritical transfer systems, the use of "Superfloc" HPI with an effective conductivity of 6.5×10^{-5} Btu/hr-ft-°F and an overall weight of 1.29 lb/ft^3 was assumed. The effective conductivity and weight values, assumed for "Superfloc," come from a test program performed on a complete 87-inch tank system as reported in Reference 5-2. This is an overall nominal value and includes mounting pins, tank support struts and line penetrations. Detailed calculations and analysis of the test data indicated that the support struts and line penetrations were a small amount of the total. The six support struts were made of unidirectional fiberglass. A low heat contribution from supports was also assumed to be true for the present analysis. Designs using CRES lines coming into the tank, with lengths necessary to maintain low heat leak, are also assumed.

For comparison with other methods of tank pressure control, weight data were obtained for the locked up tank without mixing as a function of initial ullage fraction. The weights considered to be variable between the systems and which were included in the analysis are the thermal control elements and the storage tank. Data were generated for tanks both with and without a vacuum jacket. The following assumptions were used.

a. H_2 fluid.

b. Initial tank pressure is 20 psia with a liquid density of 4.34 lb/ft^3 and a vapor density of 0.11 lb/ft^3.

c. Final allowable tank pressure is 100 psia.

d. Tank and jacket weight data are as presented in Appendix A.

e. The basic or reference tank load is 95% liquid and deviations from this are taken account of by an adjustment of tank size and corresponding weight.

f. Temperature differential across the insulation (ΔT_i) is 500°F, $k_i = 6.5 \times 10^{-5}$ Btu/hr ft °F and $\rho_i = 1.29$ lb/ft^3.

The following steps were performed in the calculations.

a. Assume a base tank size (D_{tb} and V_{tb}).

b. Assume a value for ullage fraction (F_V).

c. Calculate the actual tank size required for the above F_V from $V_t = \dfrac{.95\ (V_{tb})}{(1 - F_V)}$

d. Calculate $\dot{Q}_a$ from Equation 5-9.

e. Determine D_t and A_s from $D_t = [V_t\ (6/\pi)]^{1/3}$ and $A_s = \pi\ D_t^{\ 2}$.

f. Calculate the required insulation thickness from Equation 5-11. For the present conditions,

$$t_i,\ \text{ft} = \frac{.0325\ (A_s\ \text{ft}^2)}{\dot{Q}_a\ \text{Btu/hr}} = \frac{0.102\ (D_t\ \text{ft})^2}{\dot{Q}_a\ \text{Btu/hr}}$$

g. Calculate insulation weight from

$$W_i = \rho_i\ (\frac{\pi}{6})\ [\ (D_t + 2\ t_i)^3 - D_t^{\ 3}\ ]$$

h. Determine tank weight from Figure A-1 at D_t and 100 psi.

i. Determine required gap between inner tank and vacuum jacket. Minimum required spacings for design are presented in Figure 5-1 for two cases; (1) assuming the use of strut type inner bottle mounting and (2) assuming the use of minimum length support pads such as used with the Apollo cryogenic storage tanks. The data presented are engineering design estimates only and are used to establish a trend rather than absolute numbers. A check was made between the spacing required for the insulation and that required for fabrication from Figure 5-1 and the larger of the two used to determine the vacuum jacket diameter. Vacuum jacket weight was then determined from Figure A-5.

j. Determine the sum of tank and insulation weights.

k. Steps b. through j. are then repeated to find the minimum weight systems.

The above calculations were made for a basic bottle of 42.5 ft^3 for three different assumptions of vacuum jacketing (1) maximum gap per Figure 5-1, (2) minimum gap and (3) no vacuum jacket. Total weight data for the three cases are plotted in Figure 5-2.

It is noted that in all cases the optimum ullage is between 12 and 16%. Comparative data were also generated for basic tank sizes of 25 in. and 150 in. diameter and are presented in Figures 5-3 and 5-4. These data show that the optimum ullage is shifted toward higher values for the smaller tank and lower values for the larger tank. Resulting insulation thicknesses are presented in Figure 5-5 for reference. Optimum initial ullages are presented in Figure 5-6 as a function of basic or reference tank diameter and are based on averages from Figures 5-2, 5-3 and 5-4.

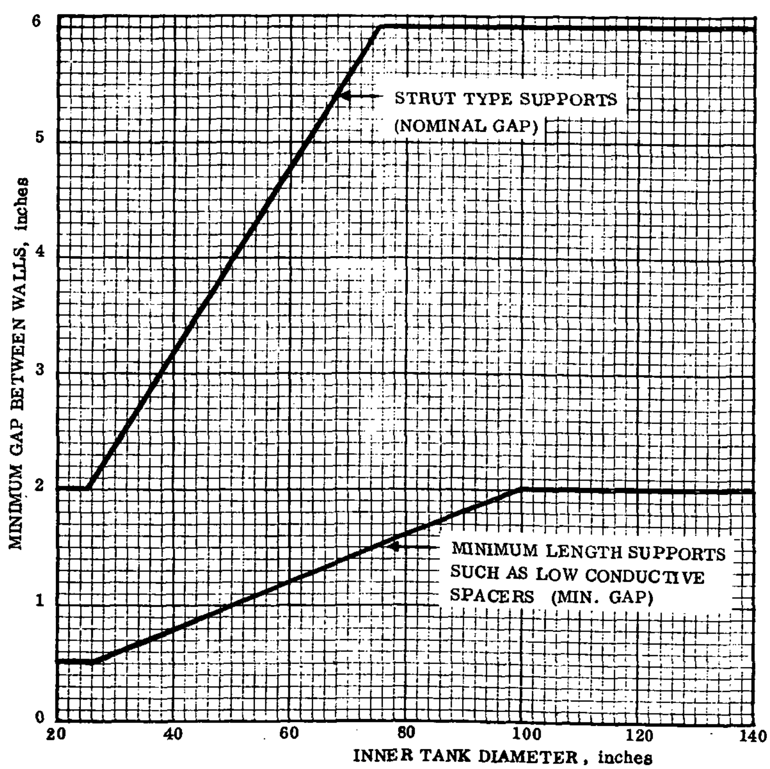


Figure 5-1. Estimated Spacing Requirements Between Inner and Outer Walls of Vacuum Jacketed Tanks

The next case considered assumed propellant mixing to allow the use of less insulation. Calculations were made in a manner similar to that for the case without mixing except that Equation 5-10 was used in place of 5-9 for determining the allowable heat leak. It is noted that for the mixed case the pressure rise is not a function of the ullage volume. However, the analysis is

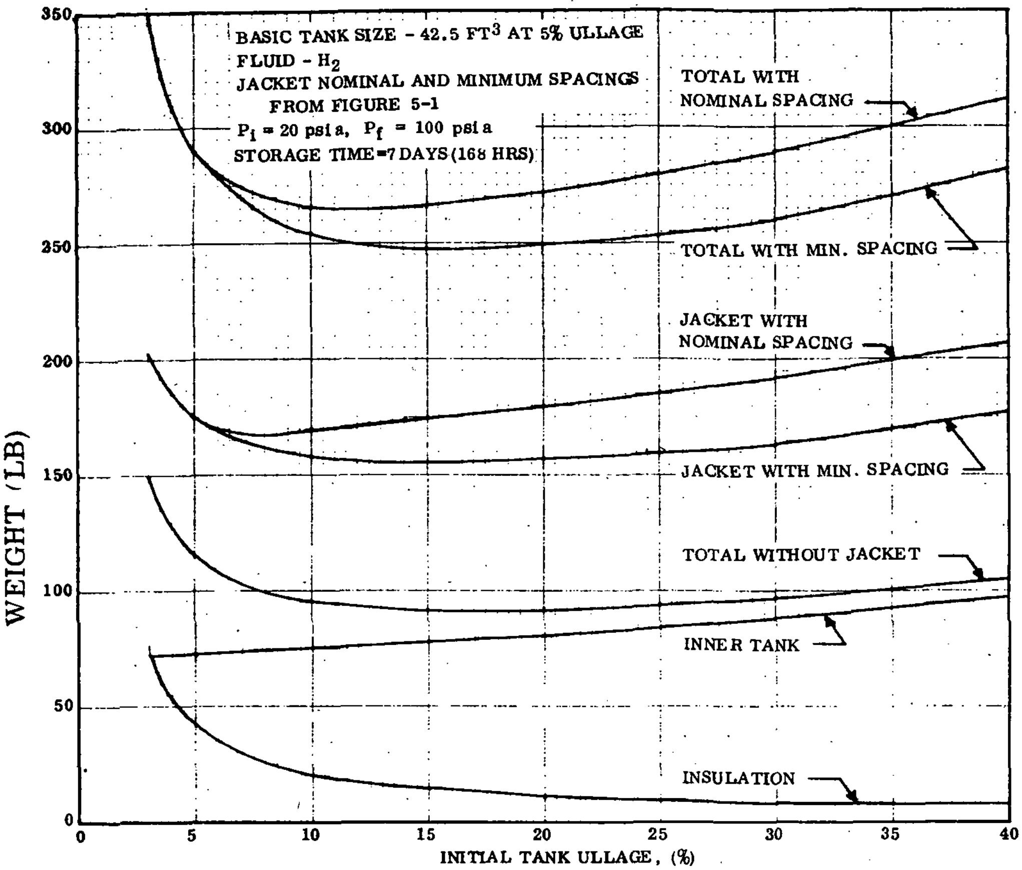


Figure 5-2. Locked-Up System Weights Without Mixing (52 Inch Tank)

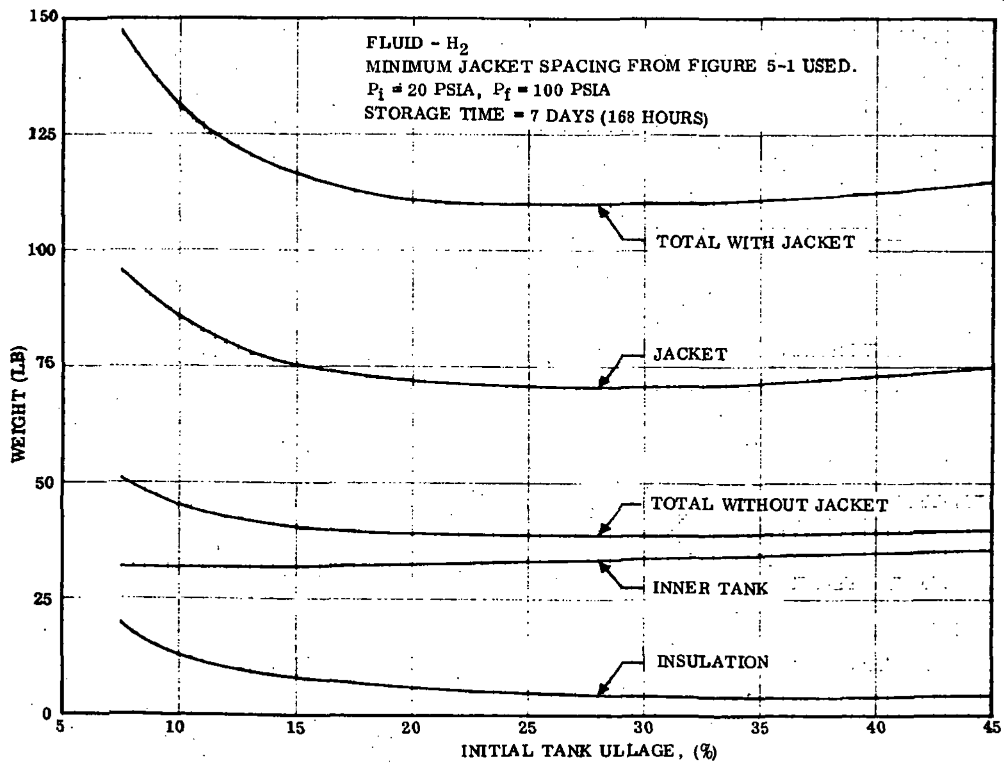


Figure 5-3. Locked-Up System Weights Without Mixing (25 In. Tank)

JACKET SPACINGS ASSUMED USE
OF STRUT FROM FIGURE 5-1
FLUID - H_2
P_i = 20 PSIA, P_f = 100 PSIA
STORAGE TIME = 7 DAYS (168 HOURS)

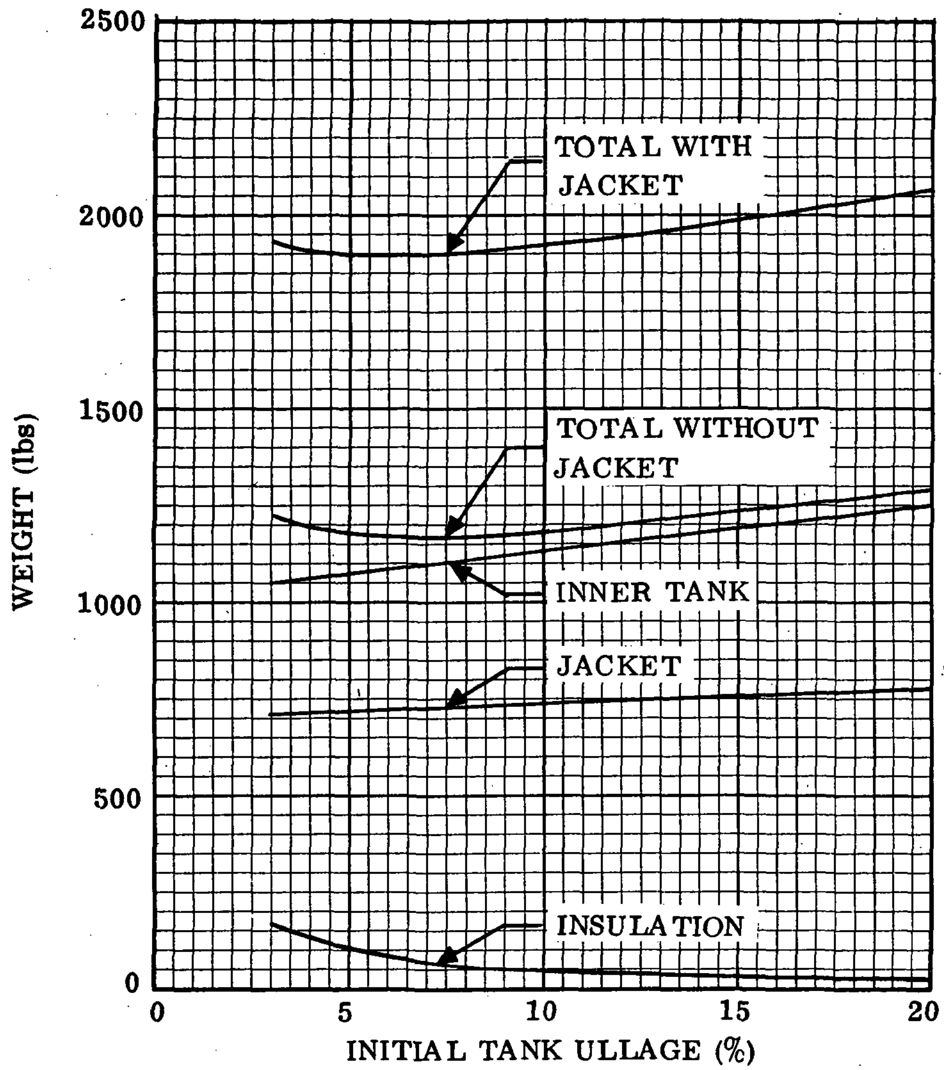


Figure 5-4. Locked Up System Weights Without Mixing (150 In. Dia. Tank)

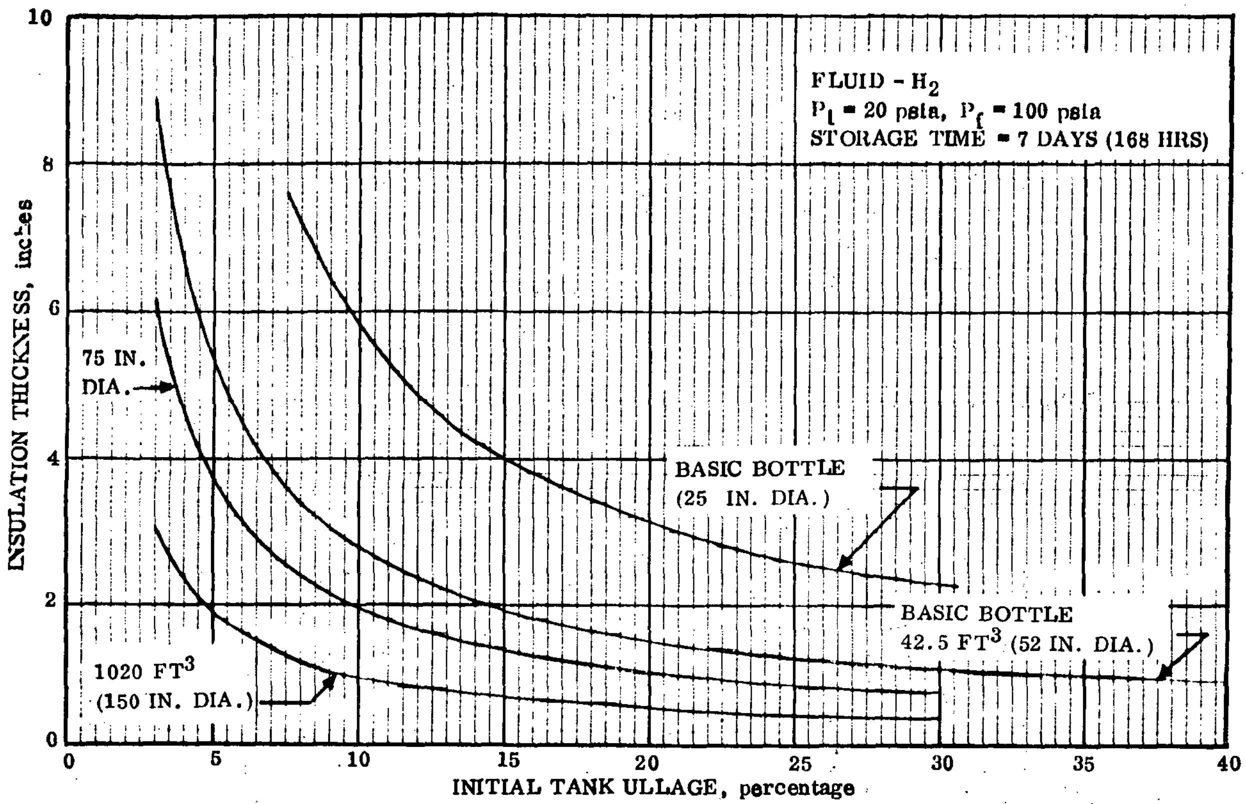


Figure 5-5. Required Insulation Thickness for Locked Up Tank Without Mixing

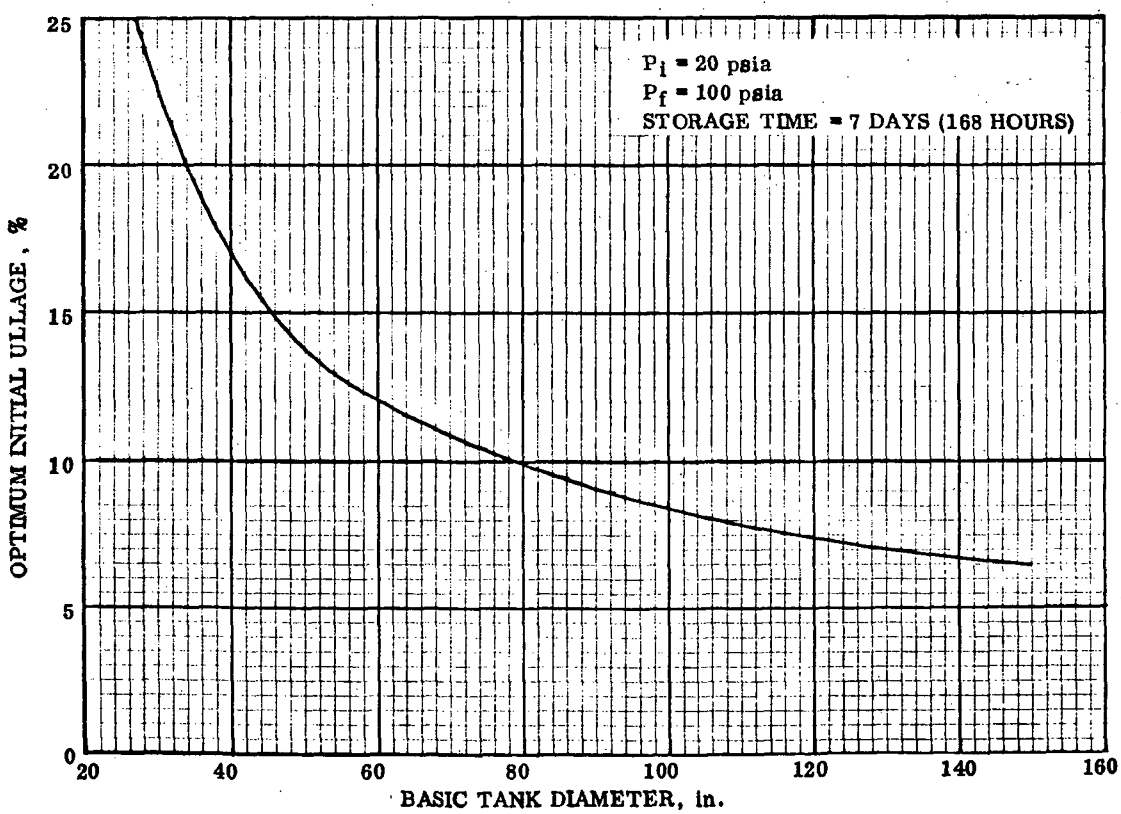


Figure 5-6. Optimum Initial Ullage for Locked Up H_2 Tank

only applicable to conditions where some ullage exists, since a two-phase mixture of gas and liquid is assumed. For this to be true throughout the full pressure rise, the initial loaded fluid mass must not be greater than that corresponding to a full tank of saturated liquid at the final tank pressure or total mass, $M_T = \rho_{Lf} V_t$. Also, $M_T = [\rho_{Li} F_{Li} + (1-F_{Li}) \rho_{Vi}] V_t$ where F_{Li} is the fraction of liquid initially loaded. Combining these two equations the maximum liquid fraction at loading is

$$F_{Li} = \frac{\rho_{Lf} - \rho_{Vi}}{\rho_{Li} - \rho_{Vi}} \tag{5-12}$$

For a final pressure of 100 psia with $\rho_{Lf} = 3.54$ lb/ft^3 and an initial pressure of 20 psia or $\rho_{Li} = 4.34$ lb/ft^3 and $\rho_{Vi} = 0.11$ lb/ft^3;

$$F_{Li} = 0.811$$

and

$$F_{Vi} = 0.189 = 18.9\% \text{ ullage}$$

Applying this ullage requirement to a basic 42.5 ft^3 (52 inch dia) tank (initial liquid load = 4.34 × 42.5 × 0.95 = 175 lb) results in an actual tank diameter requirement of 54.9 in. This increase in tank diameter compensates for the required increase in initial ullage from 5% to 18.9%. For this mixed case, the required insulation thickness is calculated to be only 0.35 in. The mixed tank weight was determined to be 81 pounds and the insulation weight 2.5 pounds. Using the minimum spacing requirement from Figure 5-1, the weight of a vacuum jacket would be 154 lb. The total system weight without and with a vacuum jacket is then 83.5 and 237.5 pounds, respectively. This compares to 90 and 247 pounds for the unmixed system from Figure 5-2. This small reduction in system weight between using a mixer versus not using a mixer is in general considered not worth the added complexity of the mixer.

Calculations were also accomplished for the non-mixed storage of LO_2 using Equation 5-2. For the same pressure rise (80 psi) and storage time (168 hr) as for the hydrogen case, then for O_2

$$\dot{Q}_a = 0.00089\ M_T S \tag{5-13}$$

where $\dot{Q}_a$, M and S are in Btu/hr, lb and percentage respectively.

Equation 5-9 used for H_2 now becomes for O_2

$$\dot{Q}_a = 0.089\ V_t\ F_V\ [\rho_L\ (1-F_V) + \rho_V\ F_V] \tag{5-14}$$

For saturated O_2 conditions at 20 psia of $\rho_L = 70.4$ lb/ft^3 and $\rho_V = 0.37$ lb/ft^3, and assuming that the product $k_i \Delta T_i$ is the same for both hydrogen and O_2 then the step by step procedure as used for the hydrogen case resulted in development of the weight curves presented in Figures 5-7 and 5-8. In the present case the gap between the vacuum jacket and inner tank was taken as the larger between insulation thickness (Figure 5-9) and minimum length support requirements from Figure 5-1.

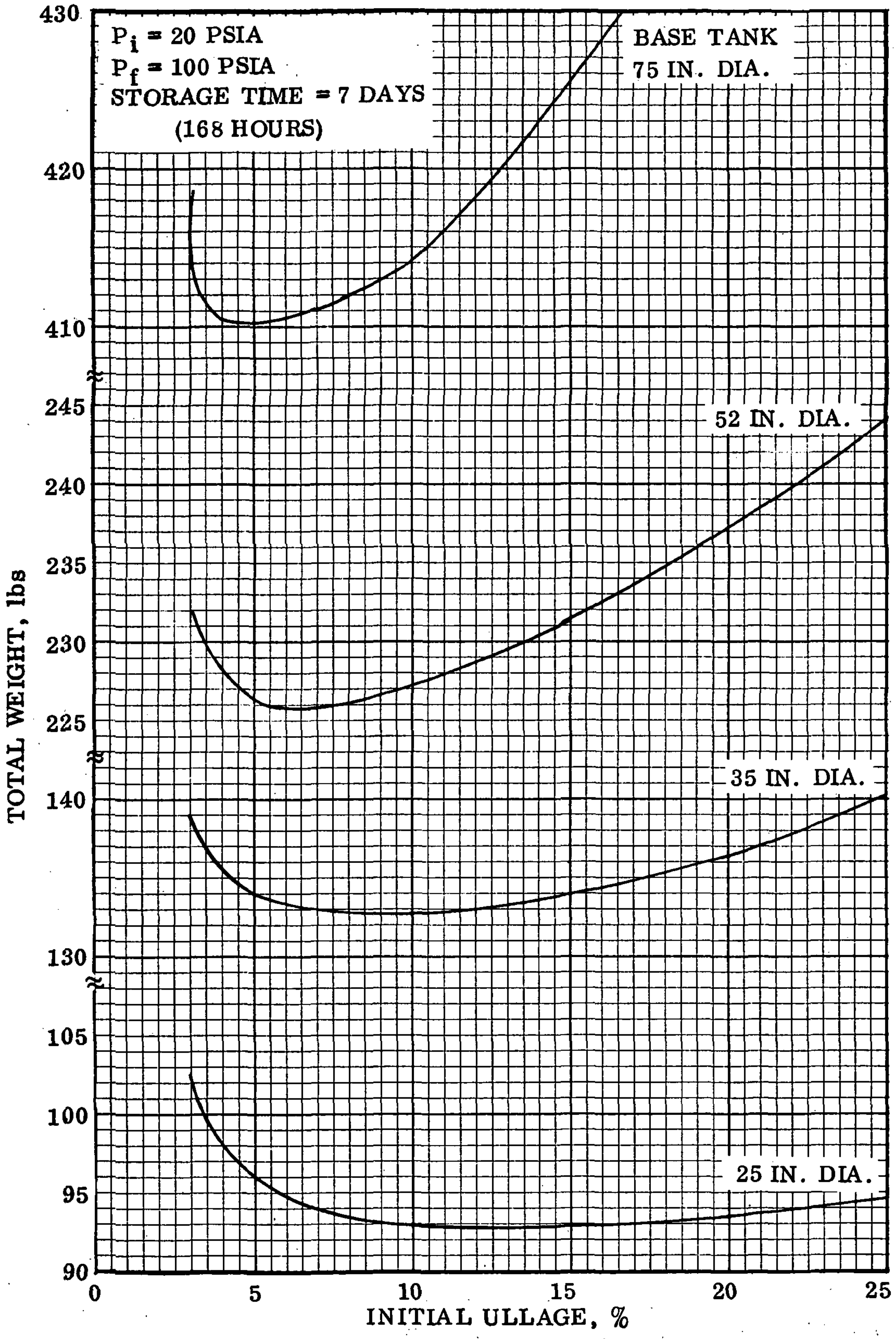


Figure 5-7. Total Dry Weight Including Vacuum Jacket, Unmixed LO_2 Storage

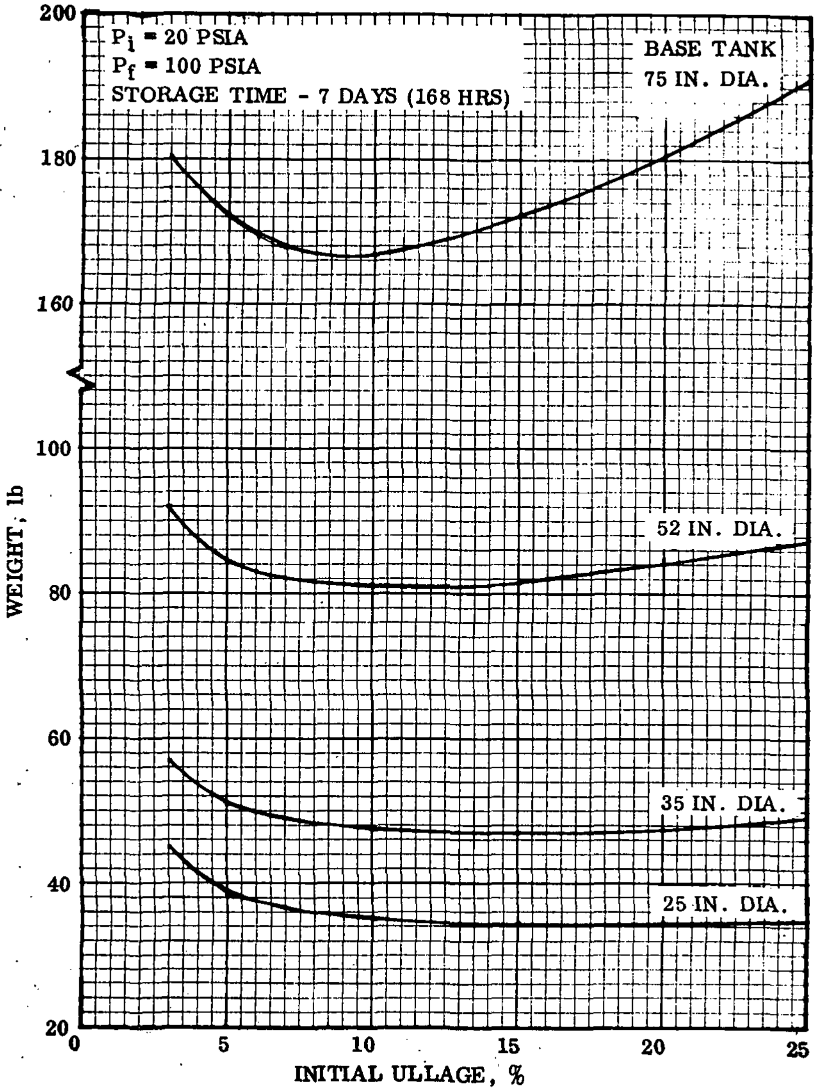


Figure 5-8. Total Dry Weight Without Vacuum Jacket, Unmixed LO_2 Storage

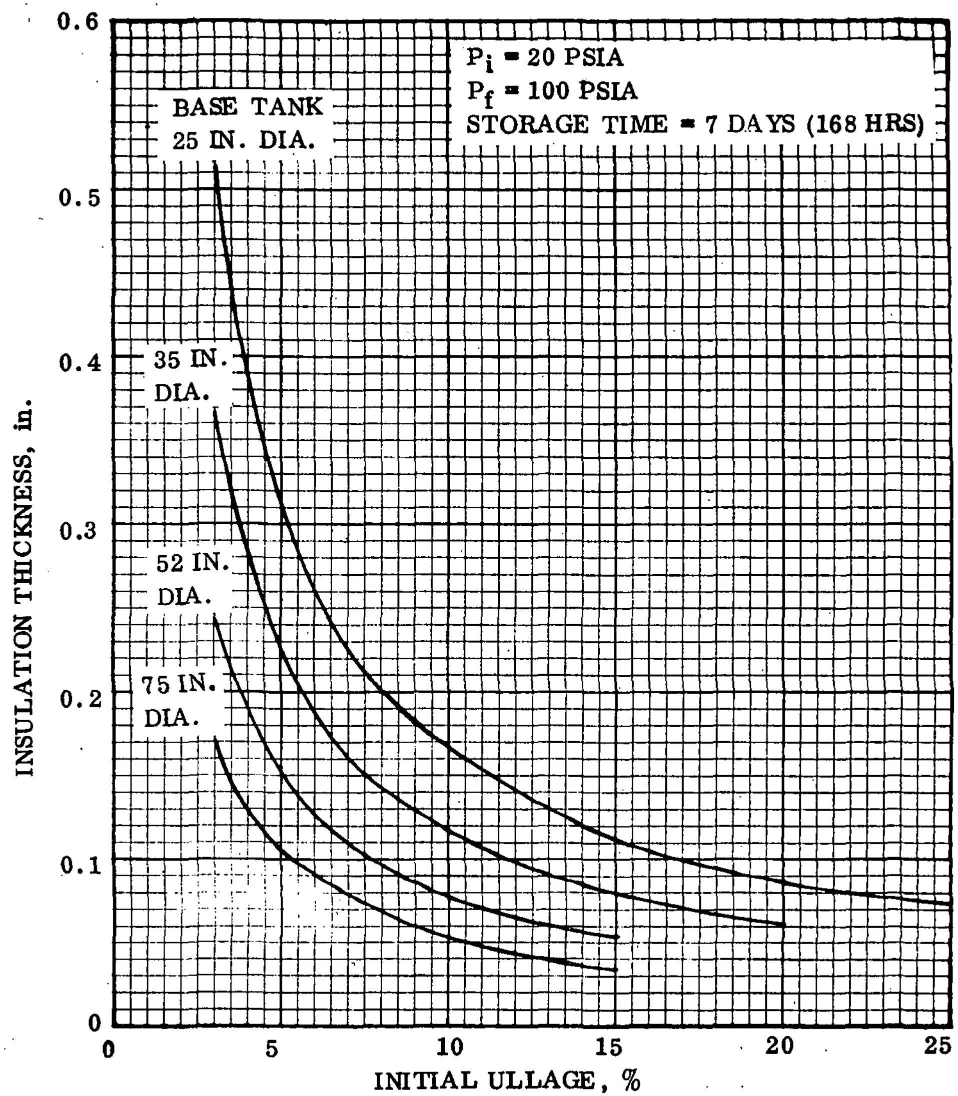


Figure 5-9. Required Insulation Thickness for Locked Up O_2 Tank Without Mixing

5.2 GENERAL PRESSURIZATION ANALYSIS

The use of helium pressurant is the basic method of liquid expulsion assumed for the subcritical supply systems. Previous work (Ref. 3-30) applied to the general problem of orbital refueling recommended this method for the expulsion of H_2 and O_2 cryogenics. In developing pressurant requirements the Epstein correlations presented in Ref. 5-3 were used. The basic equations are presented below.

$$\text{Weight of Helium Pressurant Required } (W_{He_R}) = (CF) \frac{P_t V_t}{R_G T^o} \tag{5-15}$$

$$CF = \left[\left(\frac{T^o}{T_s} - 1 \right) \left(1 - e^{-P_1 C^{P_2}} \right) \left(1 - e^{-P_3 S^{P_4}} \right) + 1 \right] \tag{5-16}$$

where

$$C = \frac{\rho_w \, C^o_{p_w} \, t_w \, T_s \, R_G}{P^o_G \, C^o_{p_G} \, D_t} \tag{5-17}$$

$$S = \frac{h_f \, \theta_T \, T_s \, R_G}{P^o_G \, C^o_{p_G} \, D_t} \tag{5-18}$$

P_t = absolute tank pressure during expulsion

θ_T = total propellant outflow time

P_1 = correlation constant; use 0.33 for H_2 and 0.775 for O_2

P_2 = correlation constant; use 0.281 for H_2 and 0.209 for O_2

P_3 = correlation constant; use 4.26 for H_2 and 3.57 for O_2

P_4 = correlation constant; use 0.857 for H_2 and 0.790 for O_2

It is noted that in the complete equation for the collapse factor (CF) given in Ref. 5-3, there is another term to account for the effect of external heat transfer. Calculations showed, however, that for the present conditions where superinsulation is used this term is negligible and was therefore not included here.

In the present analysis initial calculations were made to estimate the effects of locating the helium storage bottle inside or outside the cryogenic propellant tank. For storage outside the propellant tank, the following assumptions were made.

$T^{\circ} = 500°R$ | $C_{p_w} = 0.211$ Btu/lb-°F

$T_s = 40°R$ (H_2) | $D_t = 52$ in.

$C^{o}_{p_G} = 1.25$ Btu/lb-°F | $h_f = 23.3$ Btu/hr-ft^2-°F (from Ref. 5-4)

$t_w = 0.2$ in. | $P_t = 200$ psia

$\rho_w = 0.1$ lb/in^3 (aluminum tank) | $R_G = 386$ ft-lb/°R

$\theta_T = 1,000$ sec | $V_t = 42.5$ ft^3

Based on the above conditions the collapse factor (CF) was calculated from Equation 5-16 to be 2.5 and from Equation 5-15 the required pressurant is 15.8 lb. Then, assuming 3300 psia helium storage at 500°R ($\rho_i = 2.15$ lb/ft^3) and a useful storage bottle pressure decay to 350 psia, a helium bottle volume requirement of 8.8 ft^3 was determined. The following equation was used.

$$(V_{He})_t = \frac{(W_{He})_R}{\rho_i - \rho_f} \qquad (5\text{-}19)$$

The final temperature of the helium in the storage bottle was determined assuming a polytropic expansion such that

$$T_f = T_i \left[\frac{P_f}{P_i} \right]^{(n-1)/n} \qquad (5\text{-}20)$$

The actual value of the polytropic expansion coefficient (n) depends on the heat transfer into the bottle during discharge which is primarily a function of the discharge time. For a rapid discharge the expansion process will approach isentropic and n will equal k, the ratio of specific heats. For a slow expulsion the isothermal process is approached with $n = 1$ and $T_f = T_i$. Although the isentropic coefficient for He is 1.66, for the present case, it is assumed that the isentropic process is only partially approached with $n = 1.4$. Then from Equation 5-20, $T_f = 263°R$ and $\rho_f = 0.36$ lb/ft^3.

Storage requirements were next estimated for the case where the bottle would be located inside the propellant tank and maintained at LH_2 temperatures. For this case the collapse factor (CF) is assumed to be 1.0 and from Equation 5-15 with $P_t = 200$ psia, $T^{O} = 40°R$ and $V_t = 42.5$ ft^3, $(W_{He})_R = 7.93$ lb.

For storage at 3300 psia and 40°R, the initial helium density is 12.5 lb/ft^3. Assuming use down to 350 psia and 40°R, the final density $\rho_f = 3.2$ lb/ft^3 or $\rho_i - \rho_f = 9.3$ lb/ft^3 and the helium bottle volume from Equation 5-19 is then 8.53 ft^3. It is noted that this value is quite close to that required for the ambient storage case. Therefore, except for the helium weight itself the storage system weight would be similar for either case.

Further refinements and standardization of the methods used to calculate helium requirements and helium system weights were made to provide a basis for subsequent analyses of overall subcritical transfer systems. The following step by step procedure was developed for calculating pressurant system weights.

a. Determine if helium to be stored internal or external to the propellant tank. If stored internal and at propellant temperatures the collapse factor is assumed to be 1.0 and Equation 5-15 is used to calculate the helium pressurant requirement. In the case of storage at higher temperatures then Equation 5-16 is used to determine a value for the collapse factor through the following steps.

b. If a spherical tank is used an equivalent cylindrical tank diameter is determined since the basic collapse factor equations were developed for a simple cylinder. This is accomplished by defining a volumetrically equivalent cylinder with a height equal to the spherical tank diameter such that

$$\frac{\pi D_{eq}^2}{4} D_t = \frac{\pi D_t^3}{6}$$

Solving for the equivalent cylinder diameter

$$D_{eq} = 0.816\ D_t$$

This value for D_{eq} is then used in Equations 5-17 and 5-18 in place of D_t to determine the constants C and S. For a cylinder such as used for the bellows system the actual inner tank diameter is used.

c. Determine an equivalent tank wall thickness for use in Equation 5-17. This is accomplished by taking the total weight of all hardware in contact with the pressurant and dividing by the tank surface area and material density, or

$$(t_w)_{eq} = \frac{W_{TOTAL}}{A_s \rho_w}$$

This allows an accounting of the heat loss to hardware other than just the wall. Substitution into Equation 5-17 results in

$$C = \frac{C^o_{pw} T_s R_G W_{TOTAL}}{P^o_G C^o_{PG} A_s D_t} \qquad (5\text{-}17a)$$

d. Determine a value for the heat transfer coefficient, h_f. From Reference 5-4 the following equations are used.

Helium on H_2 $\quad h_f = 1.275\ P_t^{0.666}/(T^o)^{0.1}$

Helium on O_2 $\quad h_f = 0.492\ P_t^{0.666}$

where P_t, psia, T^o, °R and h_f, Btu/ft^2 hr °F.

e. Determine the collapse factor (CF) from Equation 5-16. The following property values are assumed to be used for helium pressurization.

$C^o_{P_{He}} = 1.245$ Btu/lb-°F

for $T^o = 250$ to 600°R and $P_t = 50$ to 500 psia.

C_{p_w} Al Aly = 0.21 Btu/lb-°F at 500°R

C_{p_w} CRES = 0.11 Btu/lb-°F at 500°R

f. Determine the quantity of helium pressurant required from Equation 5-15 or from a knowledge of the density of the helium at the tank pressure and pressurant inlet temperature; i.e.

$$(W_{He})_R = (CF)\ V_t\ \rho^o_G$$

g. The volume of the required helium storage bottle is then determined from Equation 5-19. For determining the final bottle density Equation 5-20 is used to calculate the final temperature. The total mass of helium stored in the bottle is determined from

$$W_{He} = (V_{He})_t\ (\rho_{He})_i$$

h. The weight of the helium bottle is determined from Figure A-6 and the helium system weight is taken as the sum of the bottle plus stored helium weight.

Using the above procedure an investigation was made to determine the effect on the collapse factor of variations in transfer time and propellant tank size. The following basic conditions were assumed; $P_t = 100$ psia, $T_s = 40$°R, $T^o = 500$°R, spherical aluminum propellant tanks, H_2 propellant, internal hardware weight negligible.

Applying the above data and the calculation methods in b. through e. for spherical tanks the following equations were developed for the constants C and S.

$$C = 122 \frac{(W_t \text{ lb})}{(D_t \text{ in.})^3} \tag{5-21}$$

$$S = 3.1 \frac{(\theta_T \text{ min.})}{(D_t \text{ in.})} \tag{5-22}$$

Using these equations in conjunction with Equation 5-16 the data curves presented in Figures 5-10 and 5-11 were developed.

Propellant tank weight data was obtained from Figure A-1. It is seen from Figures 5-10 and 5-11 that the transfer time and tank size have a fairly significant effect on the collapse factor and thus the helium requirements. Also, the larger the tank the lower the collapse factor.

5.3 LINE AND RECEIVER TANK CHILLDOWN

An important consideration with respect to subcritical transfer is the potential requirements for chilldown of initially warm lines and receiver tanks. The energy to be absorbed from cooling a tank and/or transfer line from ambient temperatures on the order of 500°R to cryogenic temperatures can be quite high. As an example, data from Reference 5-5 showed that for constant pressure chilldown of a hydrogen

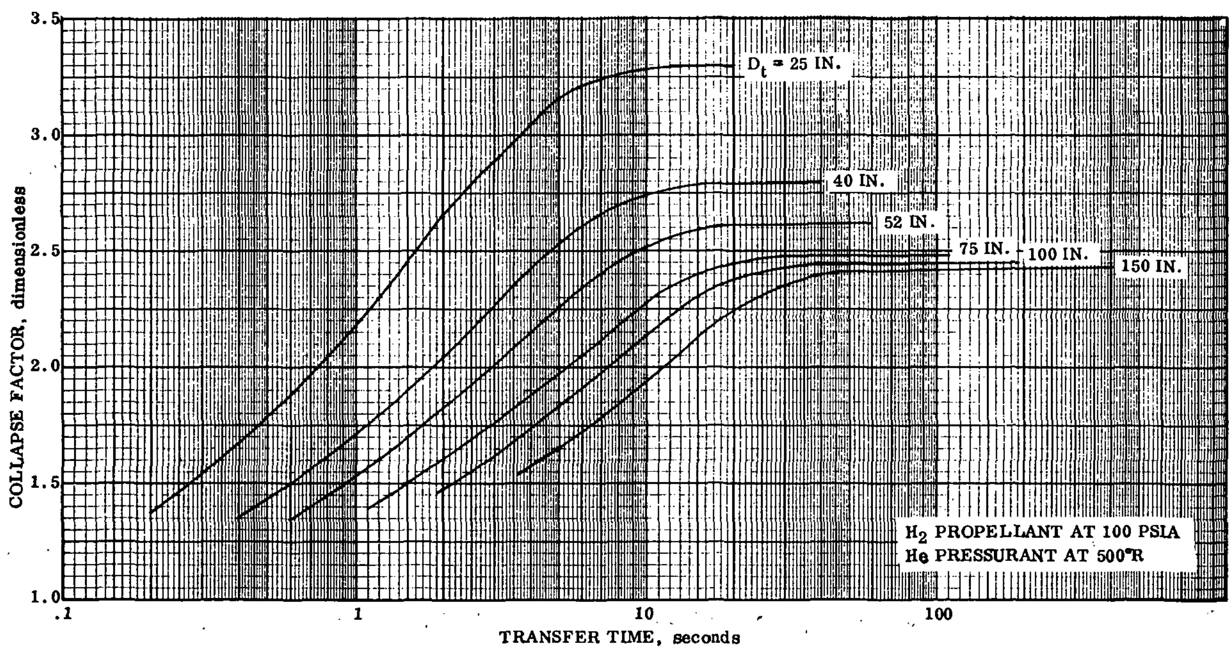


Figure 5-10. Pressurant Collapse Factor as Function of Transfer Time

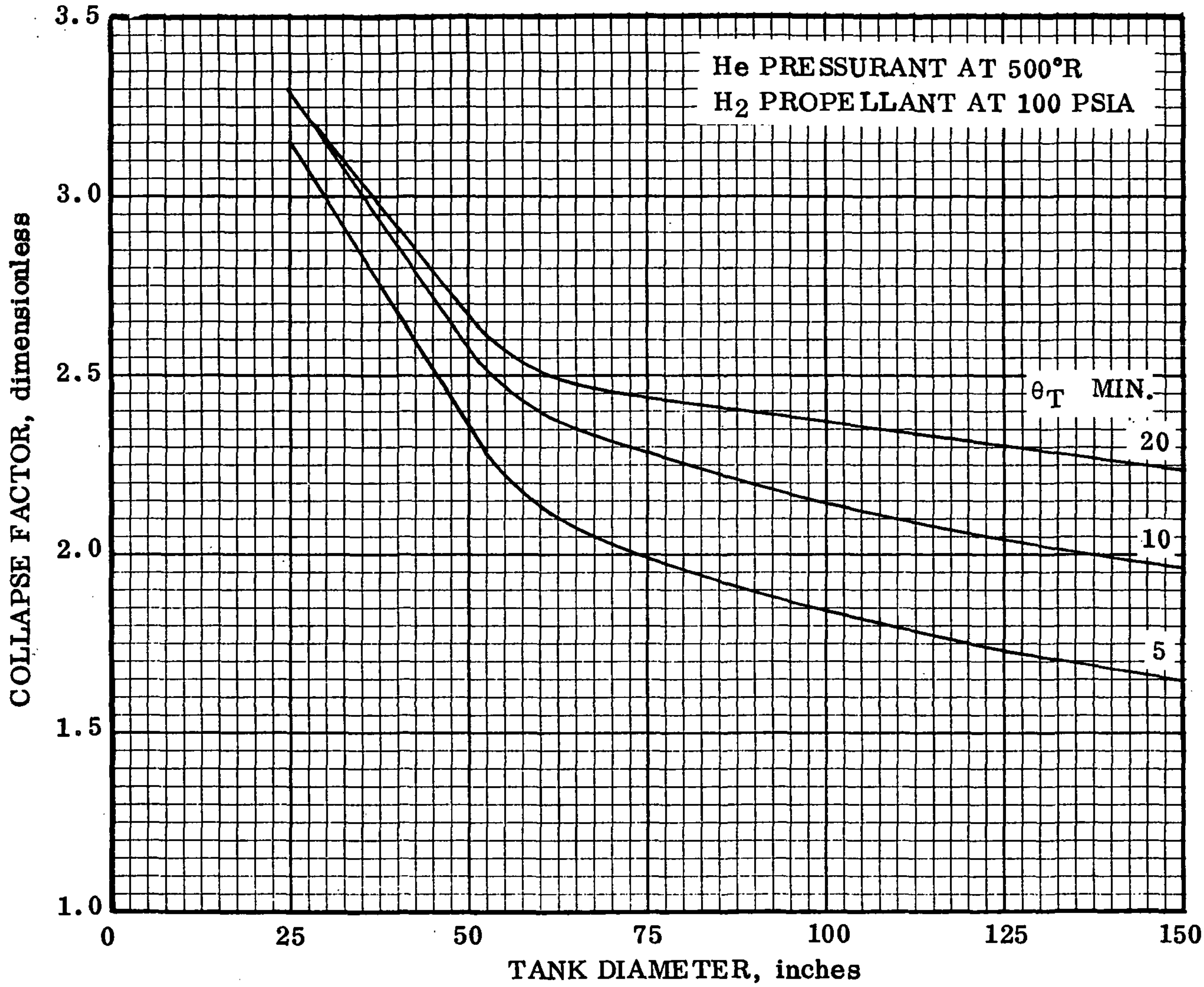


Figure 5-11. Pressurant Collapse Factor as Function of Tank Diameter

tank from 450°R to 40°R that slightly over 20 percent of the total fill mass would need to be vented. This assumes the vent fluid is 100 percent saturated vapor at the tank pressure condition of 20 psia and that only the tank itself is involved.

A significant amount of pertinent work was accomplished under the Reference 5-5 program with respect to defining optimum vent methods to be employed during chilldown at low-gravity and for purposes of the present study further work in this area is not considered to be necessary.

It is noted, however, that an alternative to the venting of a tank during chilldown is to maintain the tank in a locked-up condition and to design the tank to withstand the rise in pressure occurring during the chilldown and subsequent fill. Thus the bulk of the work in this section is devoted to the investigation of the limitations and/or supply system requirements associated with such a locked-up tank fill. In order to perform the various analyses the mixed model (fluid within receiver assumed mixed) line and tank chilldown and fill computer programs described in Ref. 1-1 were used.

The work performed is presented in the following paragraphs. Paragraph 5.3.1 presents initial exploratory runs using the line and tank combination computer program. Paragraph 5.3.2 presents the work performed with respect to the chill-down and fill of the receiver tank only and Paragraph 5.3.3 presents results of filling an initially cold tank when the line is initially warm. For comparison, Paragraph 5.3.4 presents results of computer runs made where constant pressure venting is accomplished at various vent fluid conditions. Overall conclusions are presented in Paragraph 5.3.5.

5.3.1 EXPLORATORY LINE AND TANK CHILLDOWN ANALYSES. Initial computer runs were made with the receiver tank and line chilldown combination program described in Ref. 1-1. Tank pressure and pertinent temperatures associated with the chilldown process for LH_2 flowing to the inlet of a one-half inch diameter by 100 ft long aluminum line and a 42.5 ft^3 aluminum receiver tank are presented in Figure 5-12 as a function of chilldown time. Both the line and tank are assumed to initially contain gaseous H_2 at 15 psia and 540°R. It is seen from the Figure 5-12 data that the line chills in a relatively short time. The flow rate into the tank and the accumulation of liquid in the receiver are presented in Figure 5-13. The data shows that initially, vapor existed in a significant portion of the line and choking flow occurred. As the line chills down, the flow increases to a relatively steady state value which is then affected only by changes in downstream or receiver tank pressure.

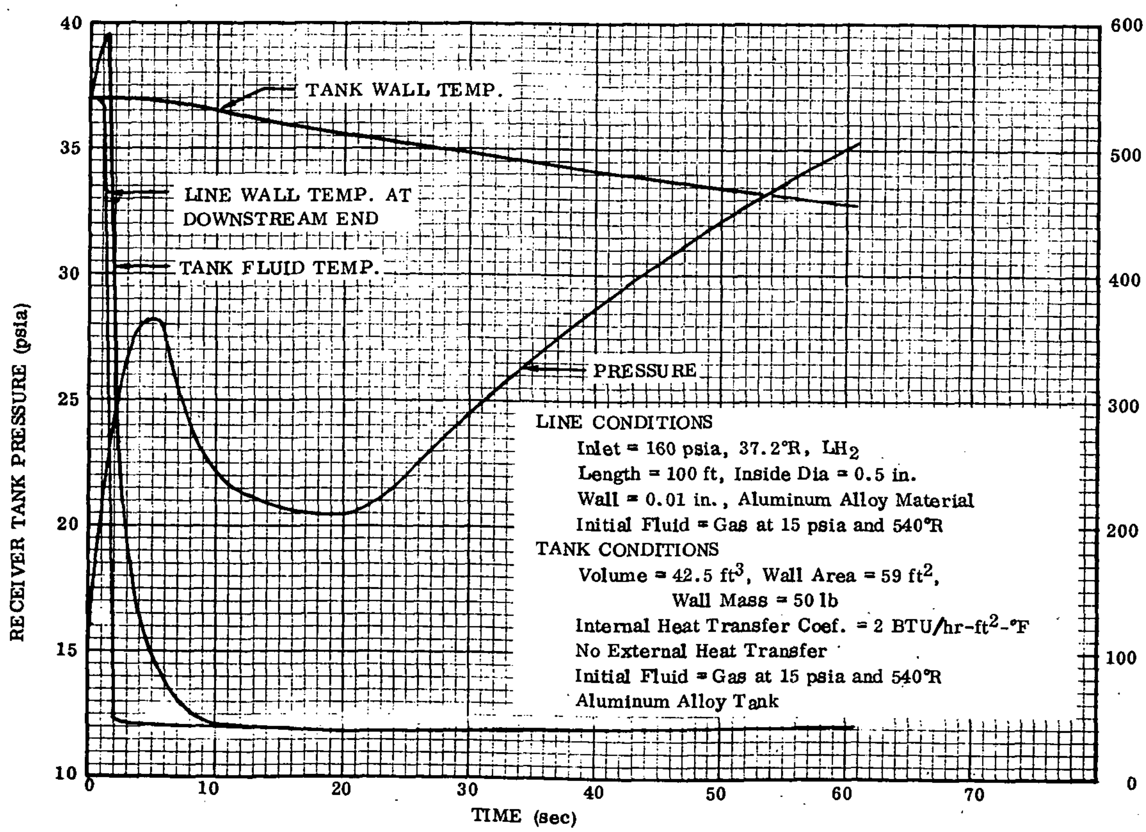


Figure 5-12. Line and Tank Conditions During Chilldown With LH_2

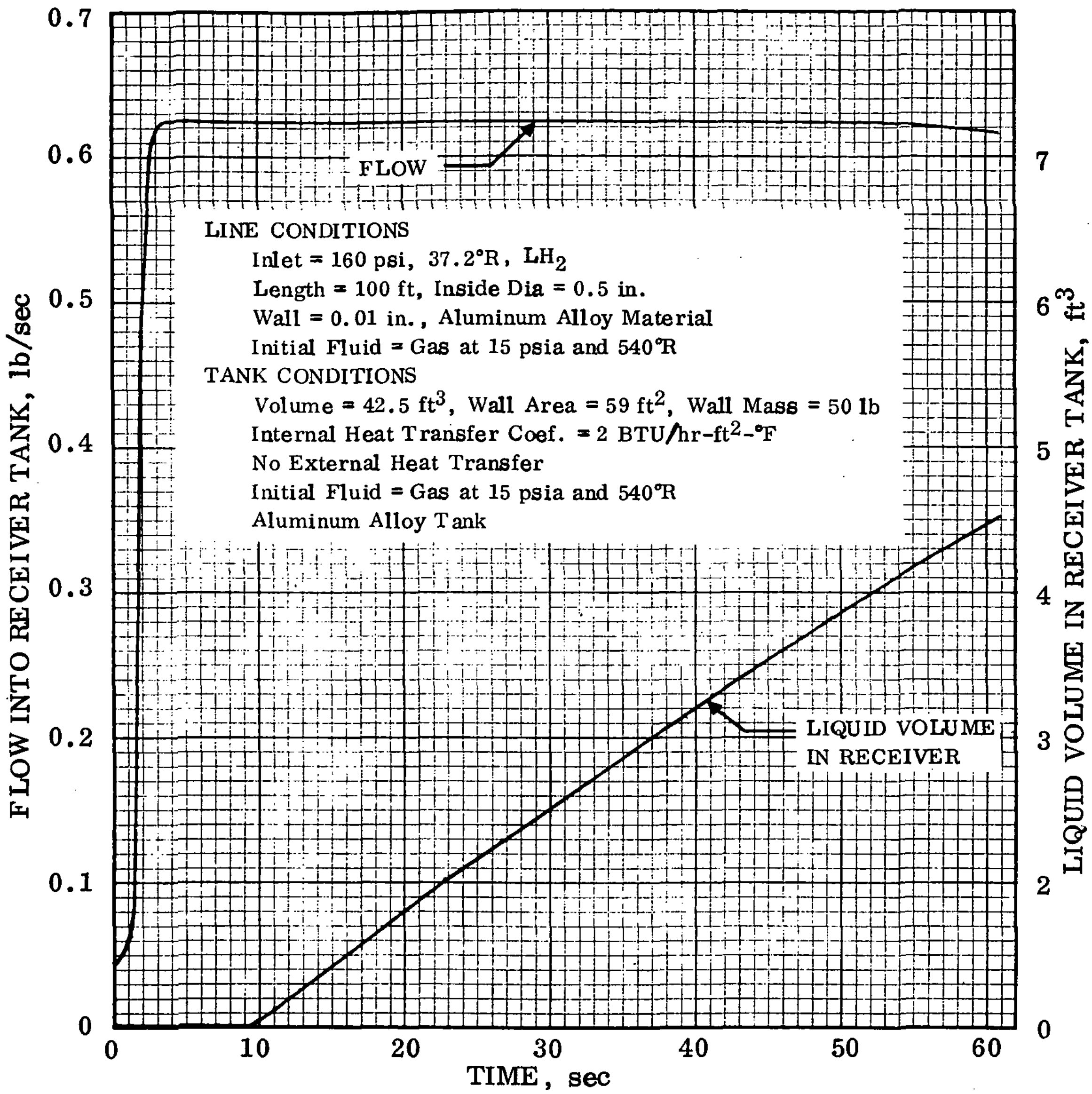


Figure 5-13. Flow and Receiver Liquid Accumulation During Chilldown With LH_2

It is noted that the data only show a portion of the chilldown and this run was designed primarily to look at the effect that the line would have on the overall chilldown process. Since the line chills in such a short time this effect is illustrated primarily by the initial pressure increase shown in Figure 5-12 and is due to the line being chilled with gas coming initially into the tank and compressing the fluid already there. This pressure rise is, however, not large when compared to the final pressure projected by Figure 5-12. The effect of the line on this final pressure would be a direct function of its mass (1.9 lb) in relation to that of the tank (50 lb) and thus would be small.

For comparison a run was made to chilldown the receiver tank without the line. Fluid input to the tank was taken at the same conditions as for input to the line and the flow was assumed to be constant at the steady state condition corresponding to that following the line chilldown from Figure 5-13. Data are presented in Figure 5-14. It is seen that the initial peak is significantly less than for the case with the line, however, the final tank pressure is still projected to be much higher than this peak. As an example the tank wall has only chilled to 450°R while the tank pressure has risen from almost 15 psia to 30 psia.

Based on the above analysis it was felt that most data of interest could be obtained by running the line and tank programs separately. In this manner a significantly greater

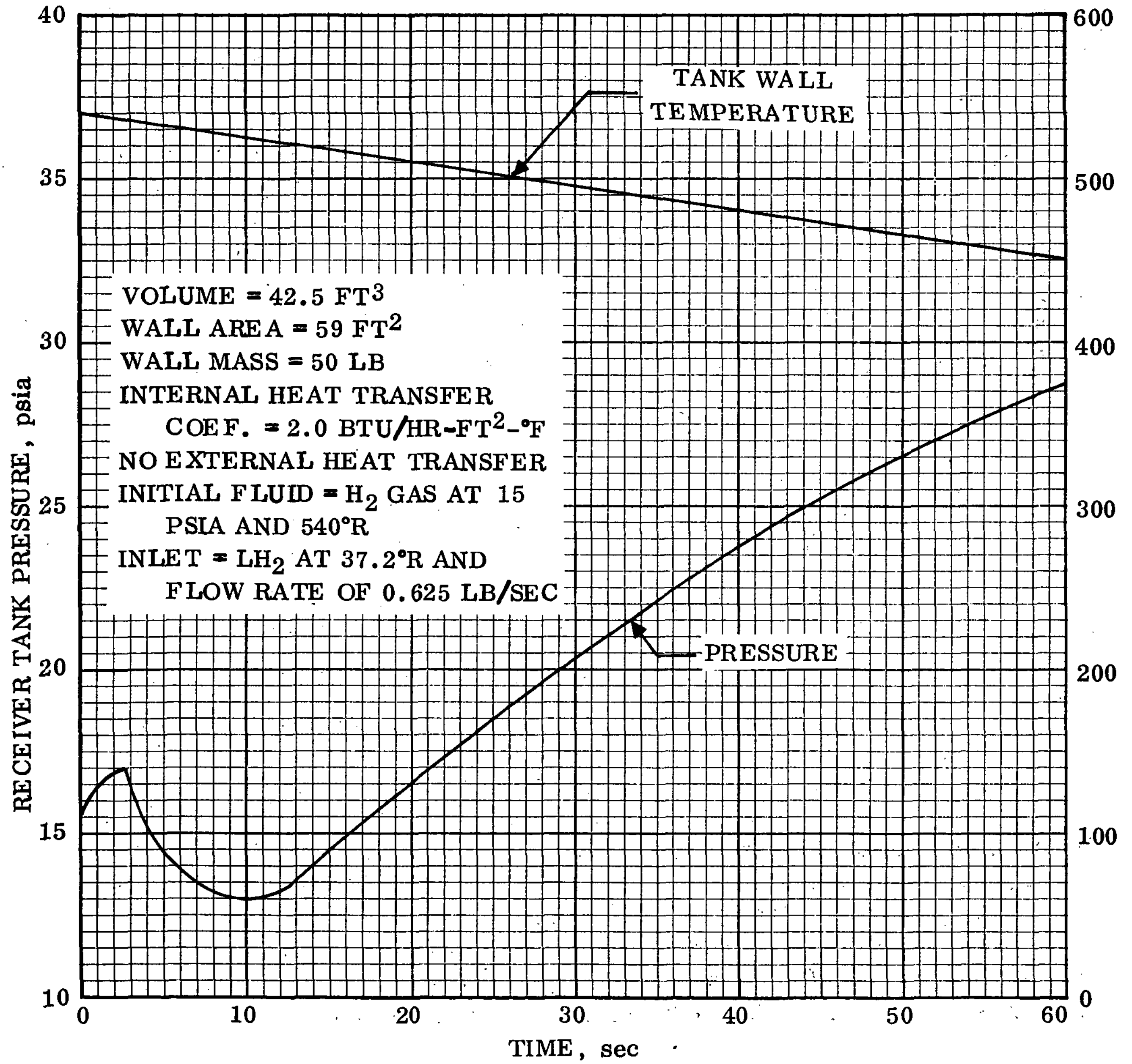


Figure 5-14. Conditions During LH_2 Chilldown of Tank Only

amount of parametric data could be obtained for the same computer time than could be obtained with the combination program. This is especially true if the number of line runs can be minimized and receiver tank data obtained independent of the line since the longest computer running time was associated with the line portion of the calculation.

The line chilldown data obtained over a range of line sizes and inlet conditions is discussed below. Line weights used in the analysis were taken from Appendix A and, including mounting and expansion sections, are presented in Table 5-1.

Table 5-1. Line Data for Chilldown Calculations

Material	Diameter, in.	Length, ft	Weight, lb
Al Aly	0.5	100	4.1
Al Aly	0.5	200	8.2
Al Aly	1.0	200	25.7
CRES	0.5	200	11.4

From the above total weights an equivalent wall thickness was then determined for use in the line chilldown computer program. In all cases run the outlet pressure was taken to be 15 psia and the line inlet fluid was LH_2 at 37.2°R.

Taking wall temperature at the end of the line as representative of chilldown a typical chilldown curve is presented in Figure 5-15. A chilldown of 3.8 seconds is determined

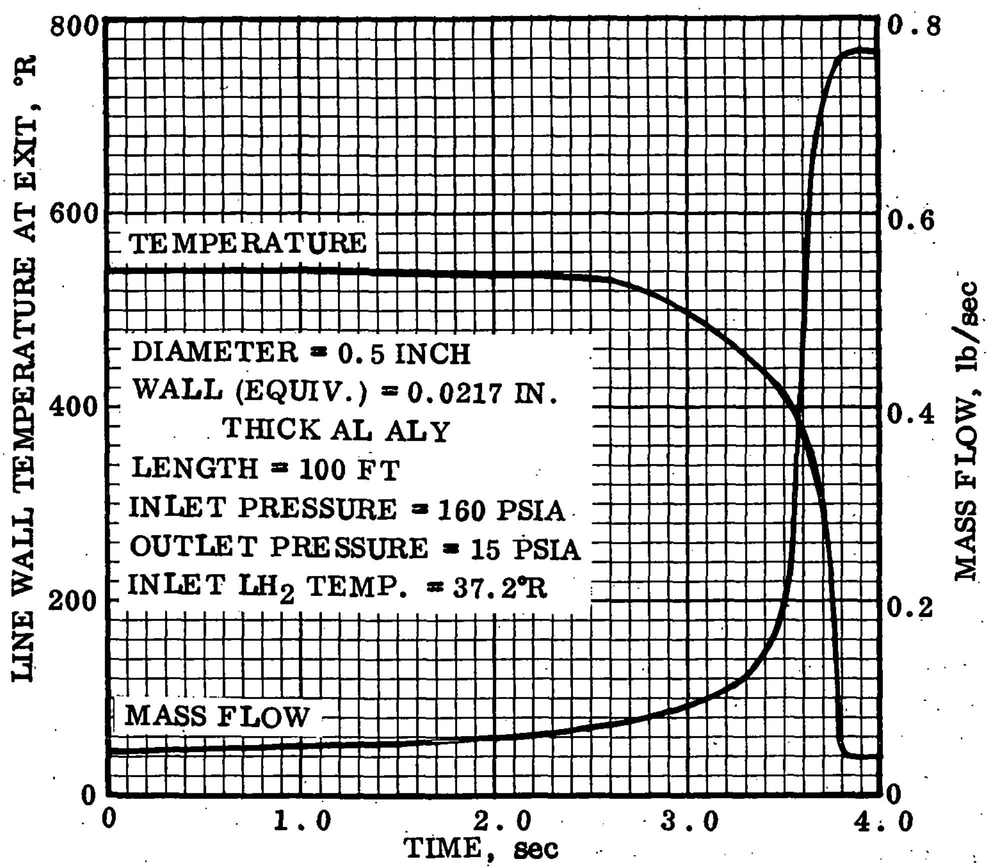


Figure 5-15. Typical Line Chilldown Data

from this data. The mass flow rate data are also presented in Figure 5-15. Mass flow following chilldown is seen to be 0.67 lb/sec. At this rate it would take approximately 250 sec to provide 167 lb of LH_2 to a receiver tank. This fluid quantity is required to fill a 42.5 ft^3 receiver tank to 90% capacity with liquid at a saturation pressure of 15 psia. Therefore, in this case, the line chilldown time represents only 1.52% of the nominal tank fill time.

Chilldown calculations were made over a range of line inlet pressures and for two different line lengths and line diameters in order to illustrate the potential effect of the line on the overall transfer process. Chilldown times are presented in Figure 5-16 and final rates and the percent of the nominal fill time required to chill the line are given in Figure 5-17. This percentage data gives an indication of the relative importance of the line in the overall tank fill process. It is seen that at low inlet pressures and long line lengths, with resulting low flow rates, line chilldown begins to become a significant factor. Data obtained for CRES lines indicate less of a contribution due to their lower heat capacity.

Data associated with chilldown and fill of the receiver-tank-only are presented in the following paragraph.

5.3.2 RECEIVER TANK CHILLDOWN. Preliminary analysis indicated that the maximum pressure which can occur in a locked-up (no venting) receiver tank during chilldown and fill is a strong function of the rate of heat transfer between the tank wall and fluid and also the inflow rate. This is illustrated by the data presented in Figure 5-18, where tank pressure histories during chilldown are given for several different wall to fluid heat transfer coefficients and inflow rates. It is seen that the minimum pressure rise occurs for the high inflow, low heat transfer case. The low heat transfer rate would be representative of a condition where cold liquid is not being injected on the hot tank wall and most of the wall cooling is by gaseous convection. The maximum pressure for this case was 66 psia versus 189 psia for the highest rate heat transfer case.

The various key points in the transfer process are illustrated on the curves shown in Figure 5-18. In all the cases presented, the first thing to occur is the start of liquid accumulation in the tank. Then, for the high heat transfer cases, the wall becomes essentially chilled and the filling proceeds at a fairly constant wall temperature. The pressure, however, decreases due to the continued addition of low energy liquid. For the low heat transfer high flow case, the tank becomes filled before the wall has had a chance to completely chilldown. The tank fluid thus absorbs additional heat following fill and the tank pressure rises by a small amount. It is noted that for the low inflow rate case the problem was terminated prior to tank filling or completion of wall chilldown. However, it appears that the peak pressure has been reached. For the transfer processes which were carried to completion, the final pressures are seen to be approximately equal. This is expected since essentially the same overall mass and energy have been transferred in each case.

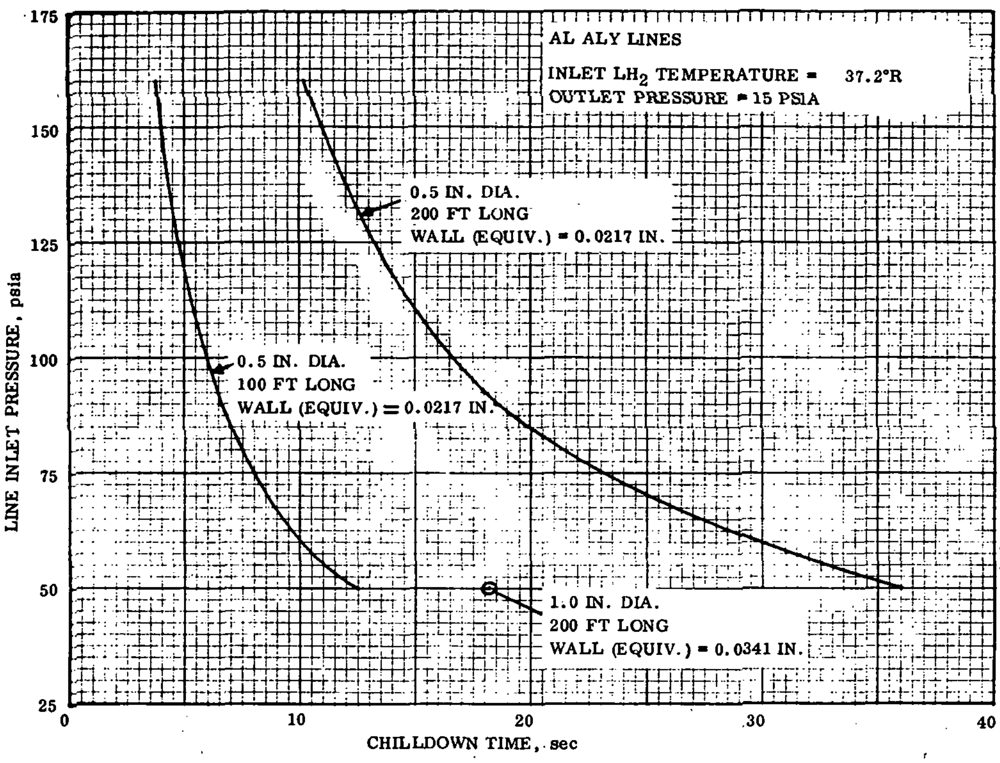


Figure 5-16. Chilldown Time Vs Line Inlet Pressure

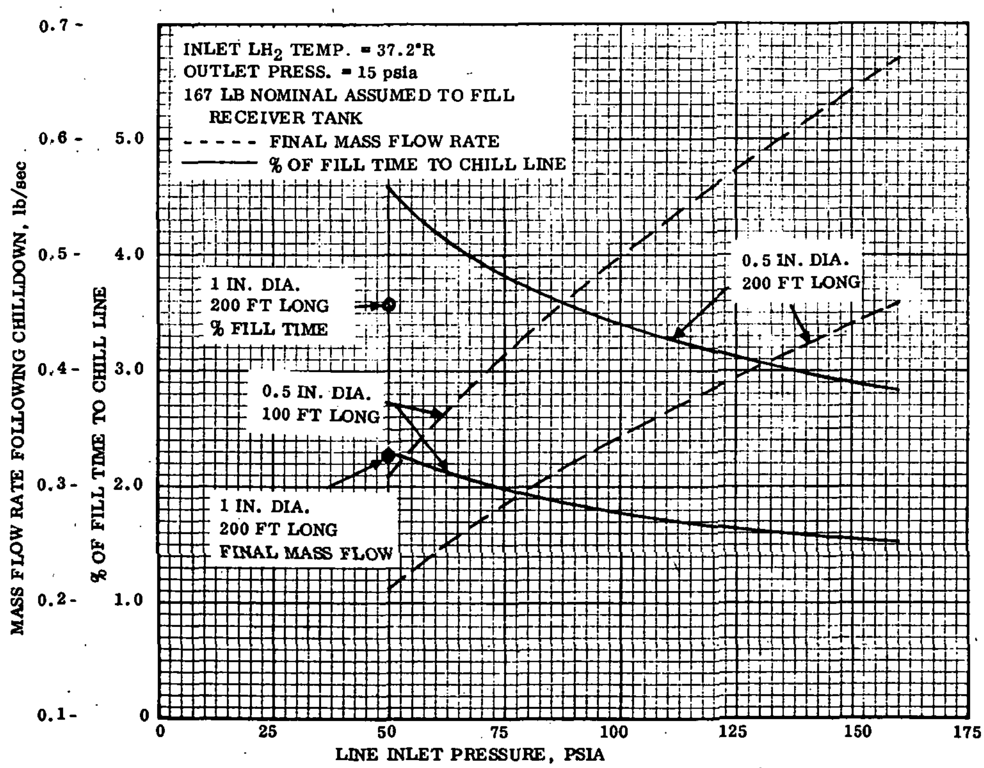


Figure 5-17. Effect of Line Length and Flow on Relative Chilldown Time

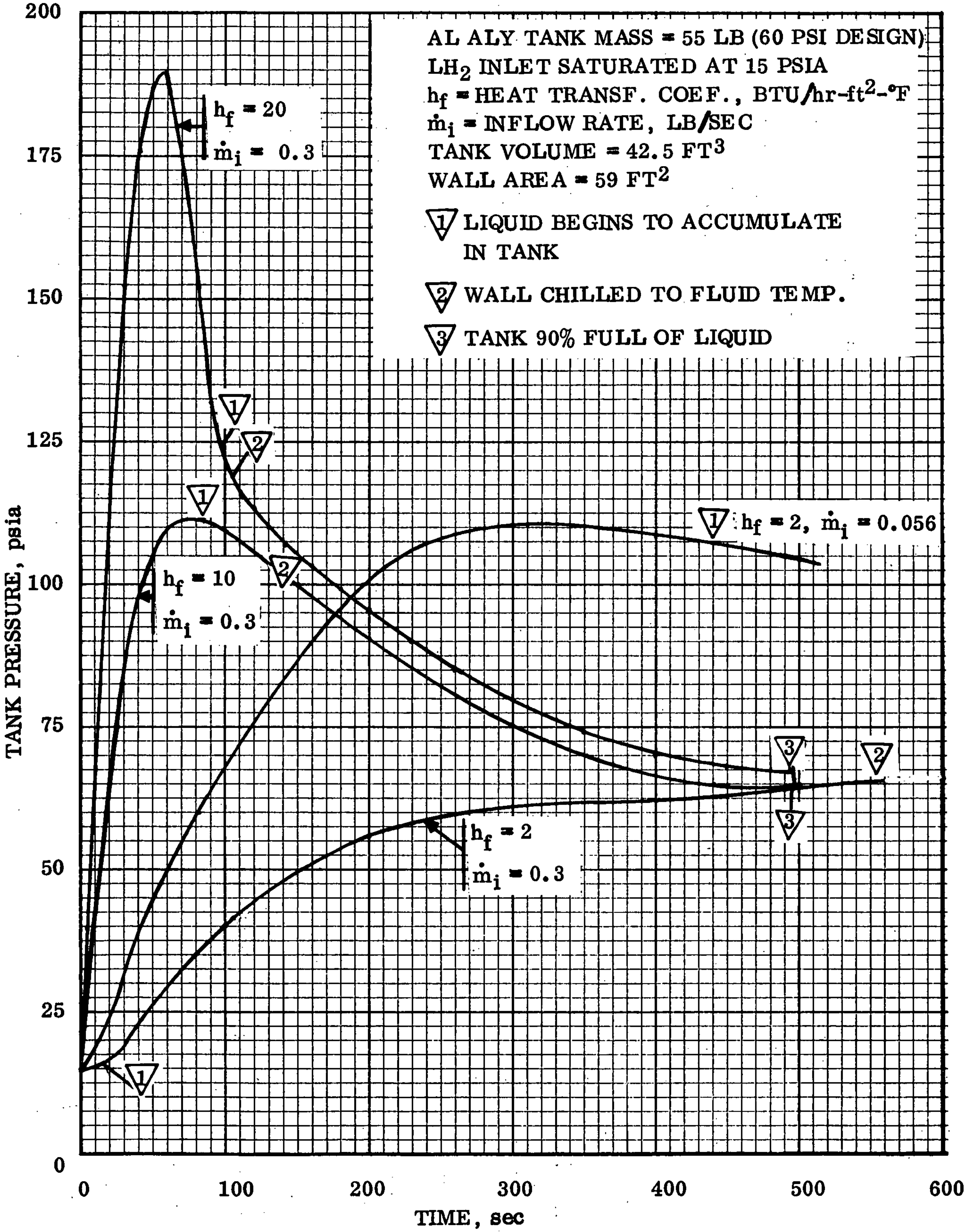


Figure 5-18. Receiver Tank Pressure During Childown and Fill

The above data serves to show that significant variations can occur in peak tank pressure, depending upon the conditions imposed. It is also noted that the peak tank pressures exceeded the tank design.

Since the assumed heat transfer coefficient has such a pronounced effect on the peak pressures reached, further investigation was made into what the actual values could be. Looking at the worst case condition (maximum heat transfer coefficient) it is assumed that liquid is present around the complete tank and boiling heat transfer would occur. Past studies for the LH_2 case (Ref. 5-5) showed that temperature differences between the wall and fluid where nucleate boiling would occur represented an insignificant part of the chilldown and therefore film boiling correlations should be used. Data from Reference 5-5 indicates that a significant variation can exist in the calculated film boiling coefficient, depending on which of the numerous correlations are used. A reasonable average value for this coefficient for the present case was determined from Ref. 5-5 to be h_f = 57 Btu/hr-ft^2-°F. This value was then used as a worst case condition in subsequent analyses.

It is also noted that the data shown in Figure 5-18 was based on a constant inflow rate which is independent of the supply and receiver tank pressures. Further investigations were made using the receiver tank computer program option where the tank inlet flow is controlled by the following equation.

$$\dot{m}_i = C_1 \sqrt{C_2 - P_t} \tag{5-23}$$

where for a constant diameter line of length L the value for C_1 would be

$$C_1 = A_{flow} \sqrt{(2g_c \rho_L)/(fL_\ell/D_\ell)} \tag{5-24}$$

and C_2 = the line inlet pressure.

Also, in the runs to follow the design pressure was increased to 100 psia. The maximum pressure data from Figure 5-18 showed the 55 lb bottle at 60 psi design to be marginal. By increasing the design pressure, a more favorable relation of maximum pressure to design pressure could be achieved.

For the first case run using the above flow rate relationship, a simulated transfer line size of 0.5 in. dia by 100 ft long was used. This results in a fL_ℓ/D_ℓ value of 24.5 for a smooth pipe with turbulent flow where the friction factor (f) is 0.0102. Additional allowances were made for miscellaneous bends and valving. Bend and valving K values were determined from the Convair Design Manual (Ref. 5-6) as K = 0.2 for 90° bends and 0.1 for each valve. Assuming, as a typical case, 10 - 90° bends and 5 valves gives a K_{TOTAL} of 2.5 which added to the line fL_ℓ/D_ℓ value resulted in a total effective fL_ℓ/D_ℓ value of 27. Applying this to the 0.5 inch dia line with g_c = 32.2 ft/sec^2 and ρ_L = 4.4 lb/ft^3 then from Equation 5-24, C_1 = 0.053.

Pressure versus time data for a line inlet pressure (C_2) of 165 psia are presented in Figure 5-19.

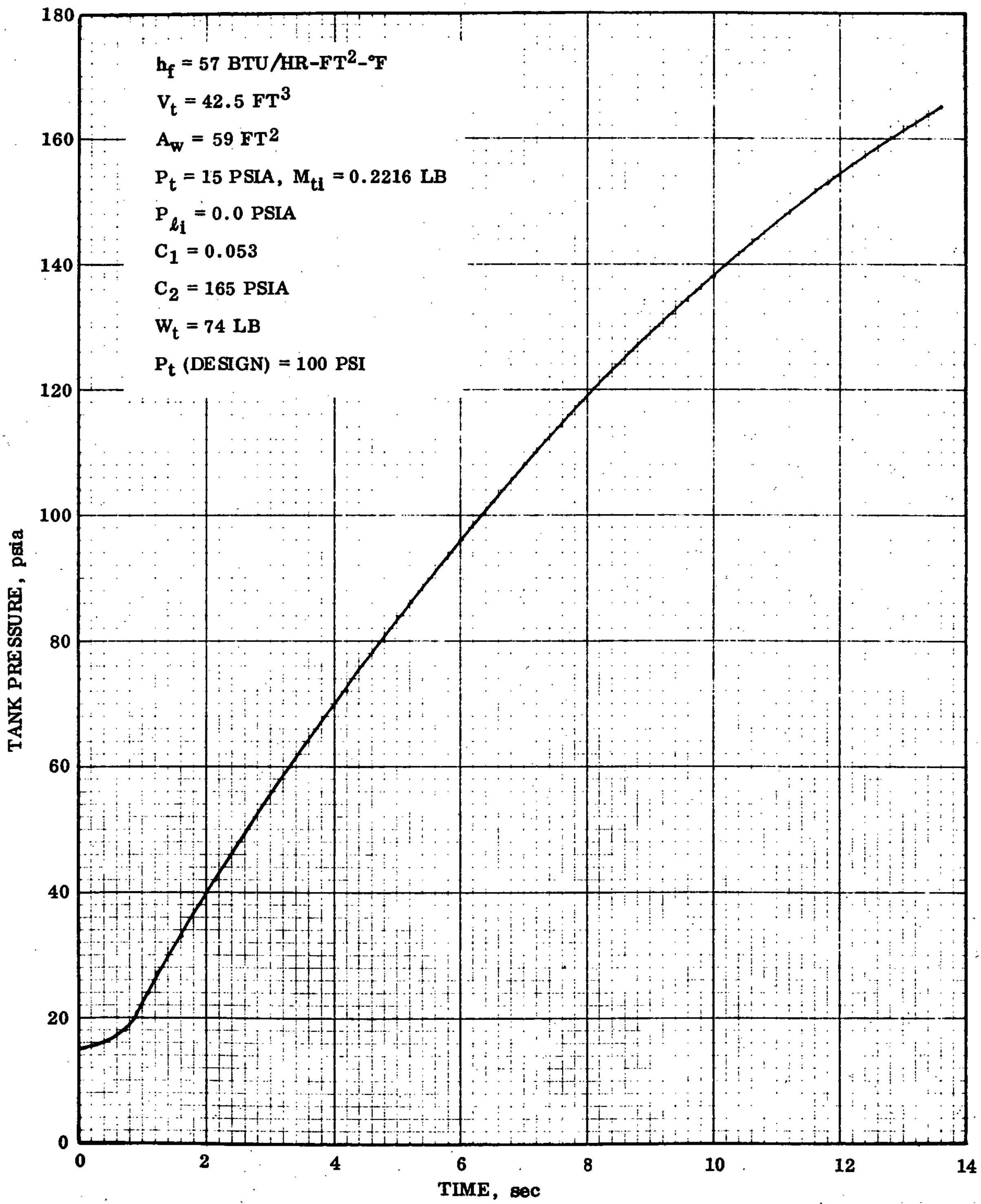


Figure 5-19. Tank Pressure During Fill With 165 PSIA Line Inlet

It is seen that the pressure immediately rises to 165 psia at which time the inlet flow stops due to the lack of a pressure differential between the supply and receiver. At this time the fluid in the receiver tank was still warm and no liquid had yet begun to accumulate in the tank. This pressure is also significantly above the 100 psia design of this tank. Indications were that the line flow was not large enough at the heat transfer rate assumed to prevent receiver overpressure.

In order to find optimum fill conditions for a locked-up tank, further analysis was accomplished with higher line inlet pressures and larger values of the constant C_1 corresponding to larger and/or shorter transfer lines. Data are plotted in Figure 5-20 for a number of cases.

It is seen that the minimum peak pressure case is for a value of C_1 of 1.0 and a line inlet pressure of 215 psia. This value of C_1 would correspond nominally to a 1.62 in. dia line by 100 ft long.

It is noted, however, that even though Case 4 results in the minimum peak pressure of the Figure 5-20 data, it was suspected that the potential minimum would be more on the

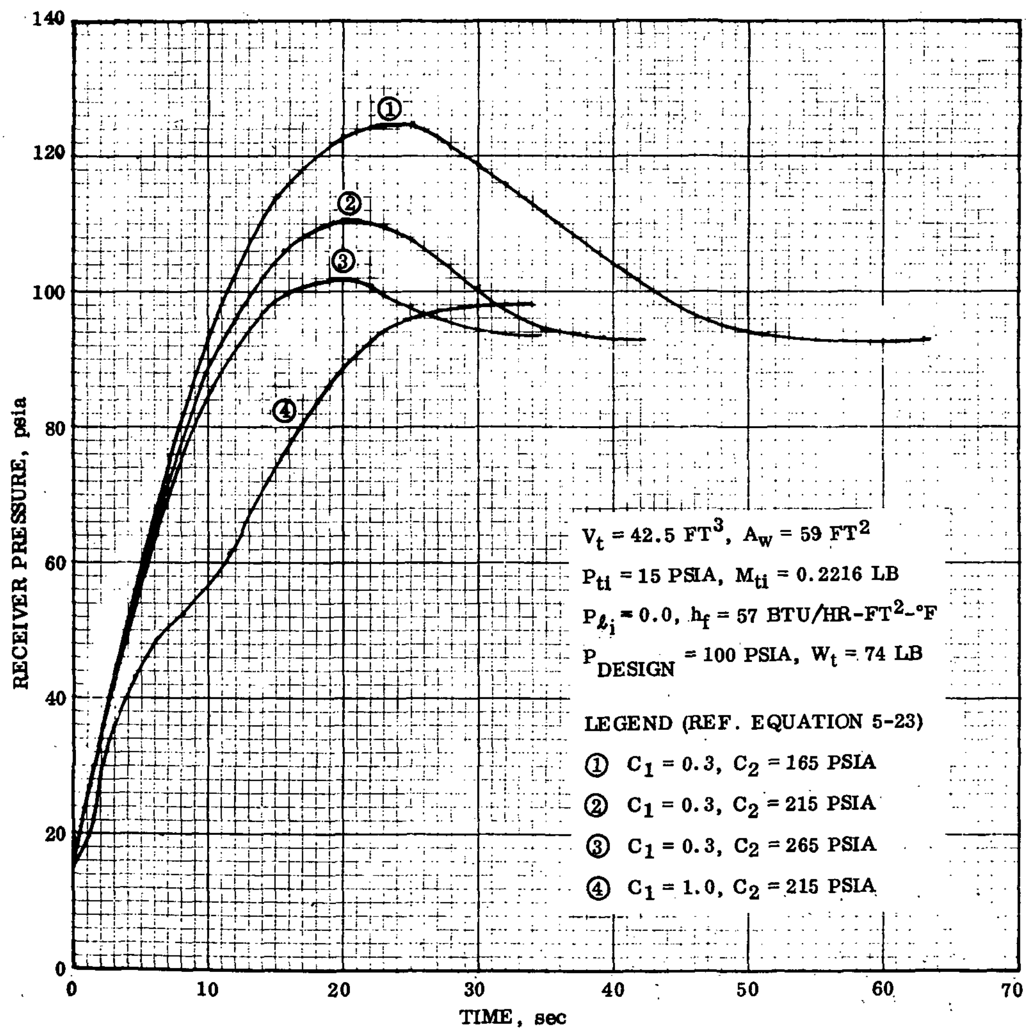


Figure 5-20. H_2 Receiver Pressure Schedule During Chilldown for Various Line Geometries and Inlet Pressures

order of 92 psia as indicated by the final pressures of Cases 1, 2 and 3. In Case 4 the inflow rate was too high for optimum fill. Further runs over a significant range of inlet conditions confirmed this. A summary of the various data obtained is presented in Table 5-2. It is noted that in Cases 4 through 13 and 15 through 17 that the maximum pressures occur at the end of chilldown. Peak pressures for Cases 1 through 3 are shown in Figure 5-20 and the peak pressure for Case 14 occurs at 20.2 sec. The ideal fill case would thus seem to be at a high rate but with a limit on the total mass allowed to enter the tank. It is noted that a mass limit rather than a volume limit would be more realistic due to the low-g nature of the filling where total mass gaging rather than volume gaging would likely be used.

Table 5-2. Chilldown Data for 42.5 Ft^3 Bottle With 100 psi Design (Weight = 74 lb)

Case	Line Inlet Pressure PSIA	C_1	Time to Fill to 90% Liquid, sec	Total Mass of Fluid at Fill, lb	Time at Completion of Wall Chilldown, sec	Max. Press. psia
1	165	0.3	61.0	153.4	63.4	124.5
2	215	0.3	42.3	141.2	36.2	110.3
3	265	0.3	35.1	141.4	35.1	101.9
4	215	1.0	11.5	151.3	33.7	98.0
5	265	1.0	10.3	152.1	33.1	100.6
6	165	1.0	14.4	148.4	33.8	94.0
7	185	1.0	12.9	149.7	33.7	95.3
8	145	1.0	15.5	146.9	33.7	92.7
9	130	1.0	17.1	145.4	33.7	92.0
10	115	1.0	19.9	143.5	33.8	92.2
11	100	1.0	28.7	140.4	33.9	94.0
12	265	0.6	17.0	146.0	33.8	92.4
13	165	0.6	24.9	141.7	33.9	93.5
14	130	0.6	34.8	142.9	34.8	97.7
15	215	0.6	19.8	144.2	33.8	91.9
16	240	0.6	18.1	144.6	33.9	91.5
17	190	0.6	21.5	142.8	33.9	92.2

The above data indicates that for minimum pressure rise, the actual fill time should occur in approximately half the total chill time. For the 42.5 ft^3 bottle this would require a fill in approximately 17 sec with a total fill mass of 146 lb. To stay below the design pressure, the fill time would need to be equal to or just slightly greater than the chill time.

Runs were also made to investigate the effects of other bottle sizes and design pressures and filling to a fixed fluid mass rather than volume. Typical data obtained are presented in Table 5-3.

Table 5-3. Effects of Tank Size, Design Pressure and Fill Mass on Receiver Childown

Tank Size, ft^3	Des. Press. psia	Tank Wt. lb	C_1	C_2	Max. Fill Criteria Mass or Vol.	Time to Fill sec	Mass at Fill lb	Time to Chill sec	Max. Press. psia
42.5	245	140	0.6	500	90% Vol.	11.9	147.5	65.9	704.9(1)
42.5	125	84	1.0(3)	130(3)	90% Vol.	17.6	143.6	38.6	106.6(1)
42.5	125	84	1.0(3)	165(3)	145 lb mass	13.7	145.3	38.5	105.9(1)
340.0	100	380	1.0	215	90% Vol.	99.0	1210.6	45.0	84.2(2)

(1) Maximum pressure occurs at end of childown.

(2) Maximum pressure occurs at 28.8 sec.

(3) Represents runs with C_1 and C_2 values corresponding to minimum pressure rise condition.

It is seen from the foregoing data that the feasibility of filling a receiver tank without venting is subject to a number of limiting conditions, such as rapid fill and/or low values of heat transfer between the tank walls and contained fluid. Also, tank design pressures and tank volumes have a significant effect on pressure conditions during fill. It is noted that the minimum pressure rise occurs when the peak pressure is at the end of wall childown with a full tank. Therefore, further investigations of the feasibility of non-vent childown were performed on a bulk energy basis. That is, knowing the tank mass, initial energy of tank fluid and specific energy of the incoming liquid then the final tank pressure can be determined as a function of the final contained fluid mass. The applicable energy balance is presented below.

$$M_f (h - Pv)_f - M_i (h - Pv)_i - (M_f - M_i)\, h_{in} = Q_w \tag{5-25}$$

where Q_w is the total energy removed from the wall between the initial and final conditions. A computer solution to the above equation was developed using the properties data of the basic equilibrium childown program. Derivations and a description of this program (CHLEND) are included in Reference 1-1.

The calculation steps used to determine the minimum peak pressure to be expected for a locked up childown and fill are outlined below.

a. Assume initial conditions of bottle temperature, fluid contained and specific enthalpy of liquid inflow to the tank.

b. Assume a final tank fluid fill mass or specific volume and calculate the change in energy of the tank contents as a function of final tank pressure using the CHLEND program.

c. Determine the bottle energy content as a function of the final tank pressure. Integrated specific heat data from Reference 5-7 is used. The assumption is made that at the final childown and fill condition the wall temperature is equal to the fluid temperature which is taken at saturation corresponding to the tank pressure.

d. By an examination of the data from steps b. and c. or by plotting on a common scale, the final pressure is determined as that where the change in energy of the tank contents minus the fluid energy added just equals the energy removed from the tank wall (Equation 5-25).

Representative data are presented in Figure 5-21 for bottle designs of 200, 150, 100 and 50 psi and final specific volumes of 0.27, 0.29 and 0.35 ft^3/lb for a 42.5 ft^3 tank. Bottle weights are obtained from Appendix A. A specific volume of 0.29 corresponds to a tank fill mass of 146 lb for the 42.5 ft^3 tank.

It is seen from this data that the final fill mass plays an important part in determining the feasibility of filling the various bottles. The Figure 5-21 data show that it would not be feasible to fill a 50 psi bottle under any of the conditions shown since the final tank pressure always exceeds the design pressure.

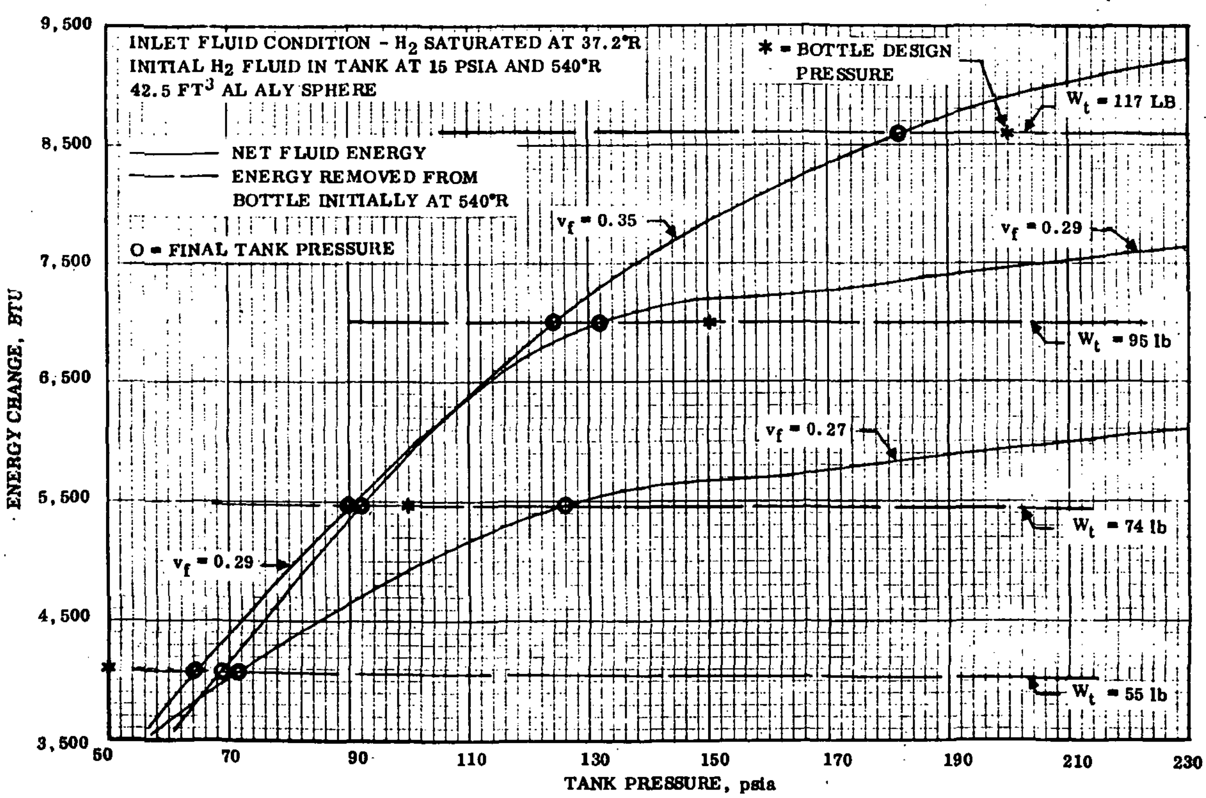


Figure 5-21. Final Tank Pressure as Function of Bottle Weight or Design Pressure

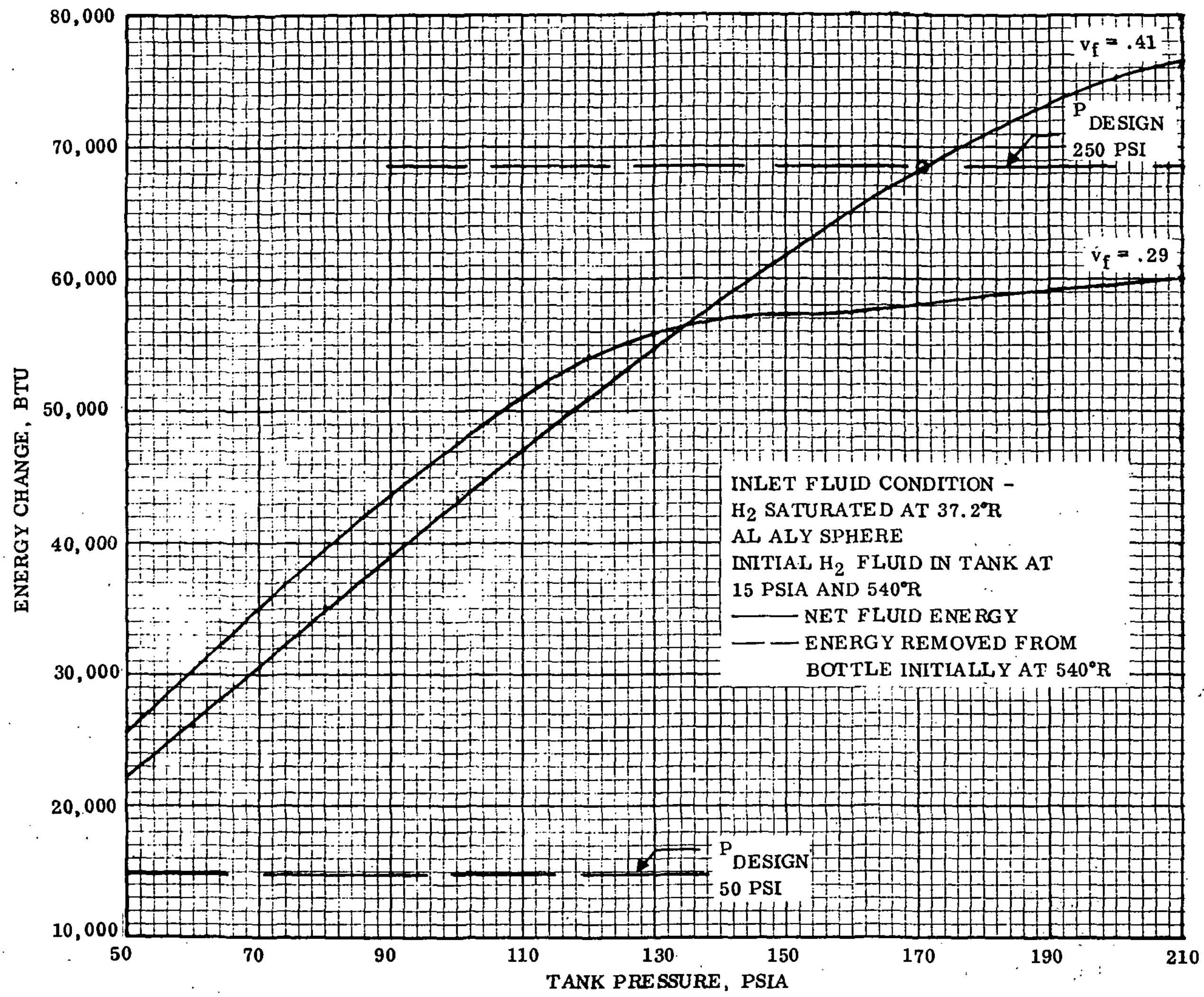


Figure 5-22. Final Tank Pressure as Function of Design for 340 Ft^3 Bottle

Data were generated for the case of filling a large 340 ft^3 bottle. This is shown in Figure 5-22. In this case it was found that the chilldown of a bottle designed for 50 psi was feasible and that the larger the size the more likely that a non-vent transfer can be accomplished.

From the foregoing data it is concluded that for a given bottle design pressure there would be a minimum size bottle, below which a non-vent chilldown and fill would not be feasible. Such minimum-size data were generated for several design pressures using the following procedure.

a. For a given design pressure, the optimum value of final specific volume is found. This is taken as the specific volume at which the net fluid energy change between the initial pressure and the tank design pressure is a maximum. The net energy (left side of Equation 5-25) as a function of specific volume for 50 psia, 100 psia and 250 psia hydrogen and 50 psia O_2 are presented in Figures 5-23 and 5-24. It is noted

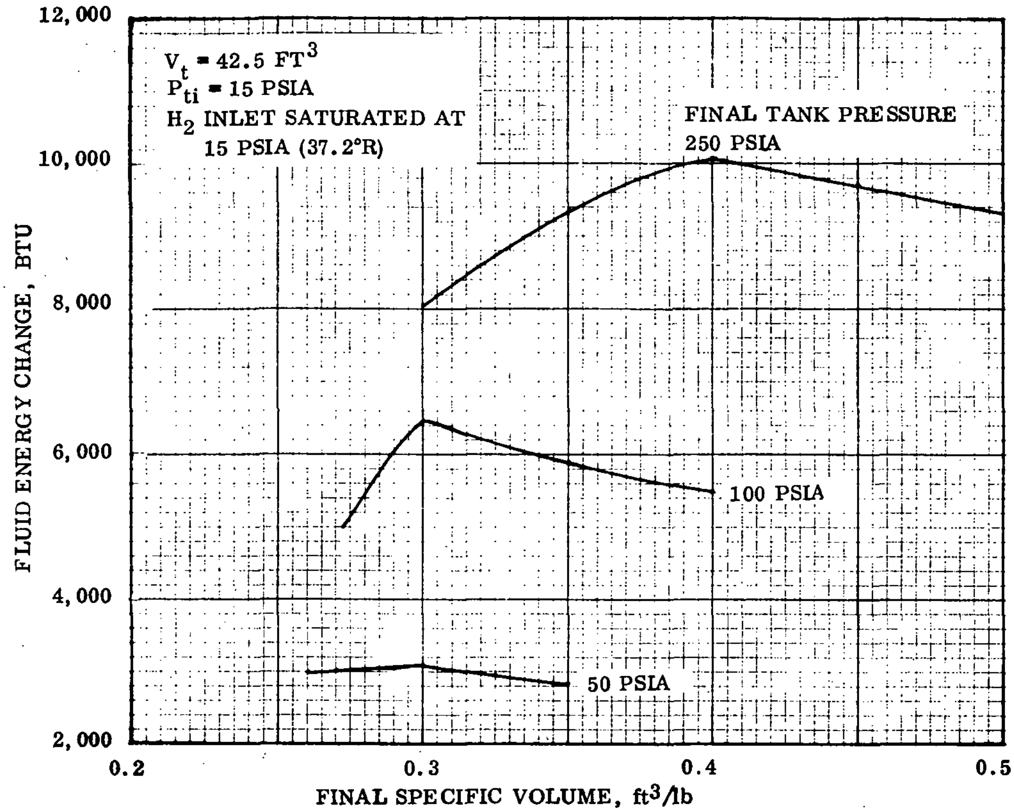


Figure 5-23. Optimum Fill Mass for H_2 Receiver Tank Childown

that the optimum values of fill mass or final specific volume, as presented by these curves, is a representation of the total energy content of the final fluid contained at a given pressure and would be independent of the initial tank fluid condition. Also the specific volume values found from Figures 5-23 and 5-24 at the peak energy conditions are applicable to any size tanks since the net fluid energy is

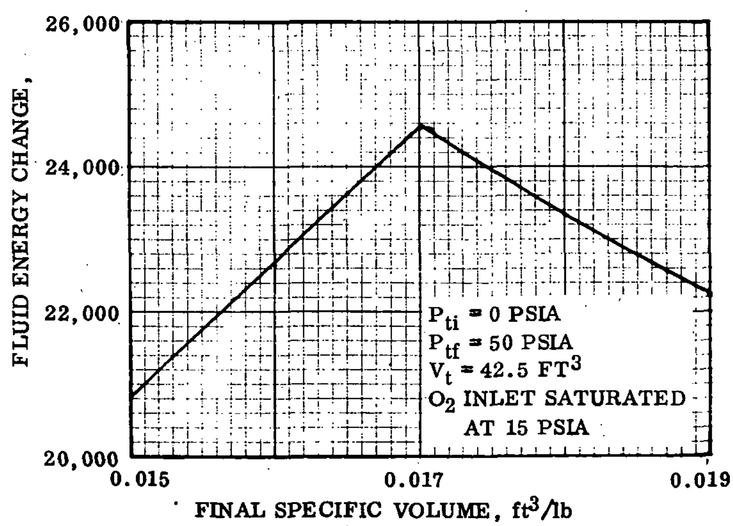


Figure 5-24. Optimum Fill Mass for O_2 Receiver Tank Childown

directly proportional to fluid mass or tank volume at any given specific volume. Optimum final specific volumes were found to be 0.3, 0.3, 0.4 and 0.017 ft^3/lb for 50, 100 and 250 psia hydrogen and 50 psia oxygen, respectively.

b. Using the above values of specific volume, the energy absorbed by the fluid is then plotted as a function of tank size along with the energy removed from the bottle. Bottle weight data as a function of size are obtained from Appendix A and corresponding specific heats are obtained from Ref. 5-7. The minimum size tank which can be filled without venting is determined by the point where the fluid energy curve crosses the bottle energy curve. That is, tank sizes below which locked-up chilldown and fill cannot be accomplished are illustrated by the conditions where the energy capacity of the fluid is less than the energy required to be absorbed from the bottle.

Typical data are presented in Figure 5-25. The effect of different initial tank pressures is illustrated in Figure 5-25 for the 50 and 100 psia design pressure cases, showing that allowable bottle sizes can be reduced by venting the receiver to 0 psia prior to the transfer. The data also shows that the use of CRES tanks allows a wider range of filling conditions (smaller sizes) than do the aluminum tanks. This is due to the lower heat capacities of the CRES tanks in relation to their weight.

Calculations with respect to oxygen transfer showed that a non-vent transfer with this fluid would be feasible under the assumed worst case conditions of a 25-inch diameter aluminum alloy bottle designed for 50 psia. For this case the energy absorbing capacity of the fluid, with a final specific volume of 0.017 ft^3/lb and an inflow energy associated with O_2 saturated at 15 psia, was 2,710 Btu while the energy required to be absorbed from the bottle, initially at 540°R, was only 1,820 Btu.

From the foregoing analyses it is concluded that chilldown of a locked-up H_2 tank is marginal in many cases and impossible in others. Also, even where feasible, the fill rate must be rapid and/or means devised to keep the heat transfer rate between the wall and fluid at a minimum.

Further, it appears that for the high flow rates involved, the required line sizes can be large and can thus significantly increase the pressures attained in a locked-up tank. Calculations were made on a bulk energy basis to illustrate this effect. For a 42.5 ft^3 aluminum sphere designed for 100 psi H_2 it was found that the maximum allowable length of a 1-inch diameter aluminum line would be 101 feet. Tank and line weights were based on the data from Appendix A. Any combination of line diameter and/or length increase above this would result in a final tank pressure of greater than 100 psia. Similar calculations for a 150-inch diameter tank and 3-inch line, both aluminum, however showed that line lengths up to 200 ft still did not increase the minimum final pressure to the 100 psi limit.

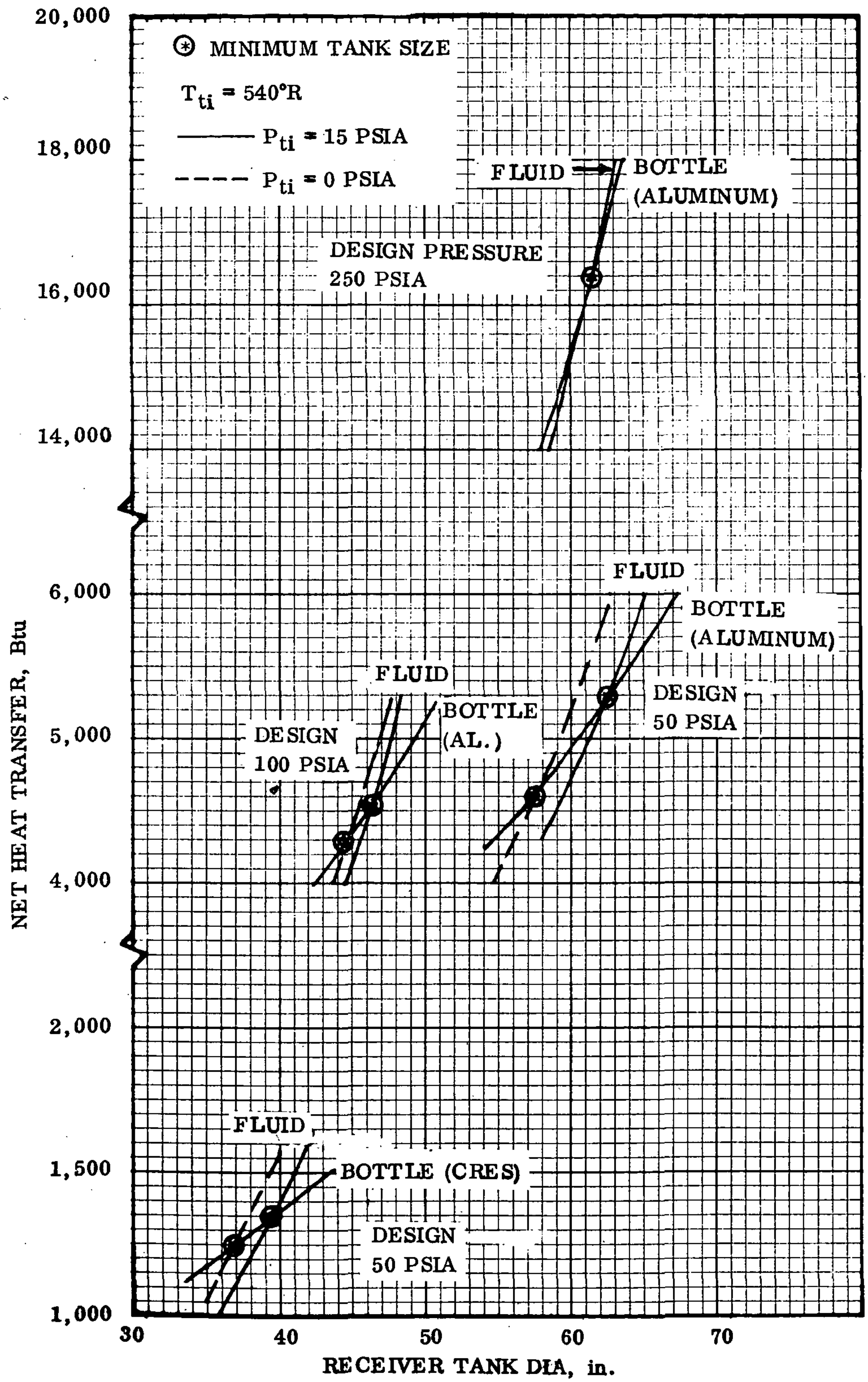


Figure 5-25. Minimum Size H_2 Bottles Which Can be Chilled Without Venting

The next step in the overall line and tank chilldown analysis was to investigate the effects of filling an initially cold tank through an initially warm line. The results of this investigation are presented in the following paragraph.

5.3.3 CHILLDOWN WITH WARM LINE AND COLD TANK. The investigation described in this paragraph covers the condition when fluid is to be resupplied to a tank which is not completely empty. The transfer line is assumed to be isolated from the receiver or orbital storage tank by a shutoff valve and has thus warmed to ambient temperature prior to the transfer. Data from a typical run are presented in Figure 5-26. The initial tank pressure is taken to be 100 psia of saturated liquid hydrogen occupying 10% of the tank volume. The inlet line is 1 inch in diameter by 100 ft long with an inlet or supply tank pressure of 120 psia. No heat transfer is allowed to occur between the bottle and the contained fluid. It is seen that the pressure rise is fairly significant for this size line and the line is less than half chilled at the termination point. A profile of the line temperature at this time is presented in Figure 5-27.

It is noted that runs without the line chilldown, such as would be the case if a bleed-off system were used to chill the line prior to tank fill, indicated that the tank pressure would actually decrease during the fill. In this case for the initially 10 percent full 42.5 ft^3 hydrogen tank at 100 psia the final pressure after filling to a total mass of 137 lb or 85% full was 70.5 psia.

This analysis shows the line to have a significant effect on filling of an initially cold tank and even in this case the use of a receiver tank vent may be desirable to minimize the required design pressures of both the supply and receiver.

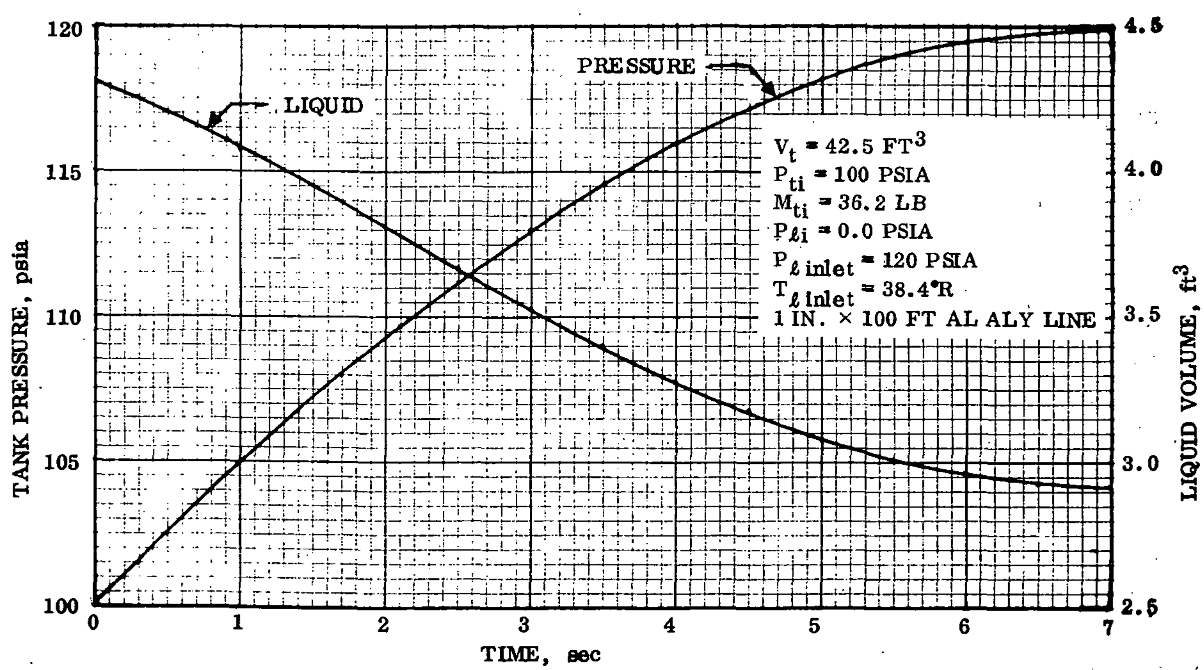


Figure 5-26. Hydrogen Tank Pressure During Line Chilldown With Cold Receiver

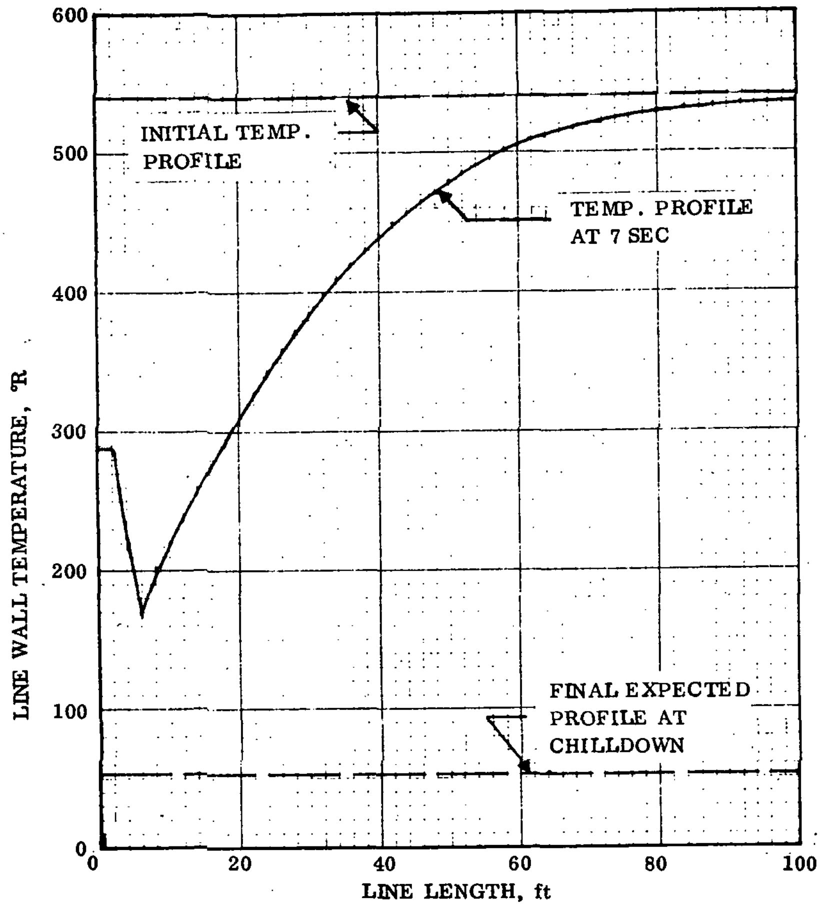


Figure 5-27. Line Wall Temperature Profile After 7 Seconds of Chilldown

A discussion of the use of venting to control receiver tank pressure is presented in the following paragraph.

5.3.4 <u>RECEIVER TANK CHILLDOWN WHILE VENTING.</u> For cases where use of a locked-up receiver tank is not feasible or advantageous from a weight and/or operational (fill rates may be very high) standpoint, venting must be accomplished. There are a number of assumptions and/or venting conditions which are possible. These are primarily with respect to the condition of the fluid when actually vented overboard. The vent conditions analyzed in the present study and for which the computer program described in Reference 1-1 can be set up are; (a) fluid vented at average mixed bulk fluid conditions, (b) fluid vented at saturated vapor conditions, and (c) vent heated to tank wall temperature.

In all cases venting was assumed to be required only during chilldown. However, for reference, data were also obtained during the fill portion following chilldown. A summary of the pertinent data obtained for H_2 venting is presented in Table 5-4. Venting is assumed to occur at a constant tank pressure of 15 psia from a 42.5 ft^3, 74 lb Al Aly bottle initially at 15 psia and a wall temperature of 540°R. The inflow rate is constant at 3.67 lb/sec [C_1 = 0.3 and C_2 = 165 psia (Equation 5-23)] at a condition corresponding to saturation at 15 psia. Other conditions are h_f = 57 Btu/hr-ft^2-°F, $A_w \approx 59$ ft^2, $P_{\ell i}$ = 0 psia, and final fill mass = 169.6 lb (90% full).

It is noted that all chilldown times were the same since the fluid temperature was essentially constant and the heat rate from the wall to the fluid was also constant. Vent flow rates corresponding to the Table 5-4 data are presented in Figure 5-28. A typical cooldown curve, for comparison with the vent rate curve, is shown in Figure 5-29.

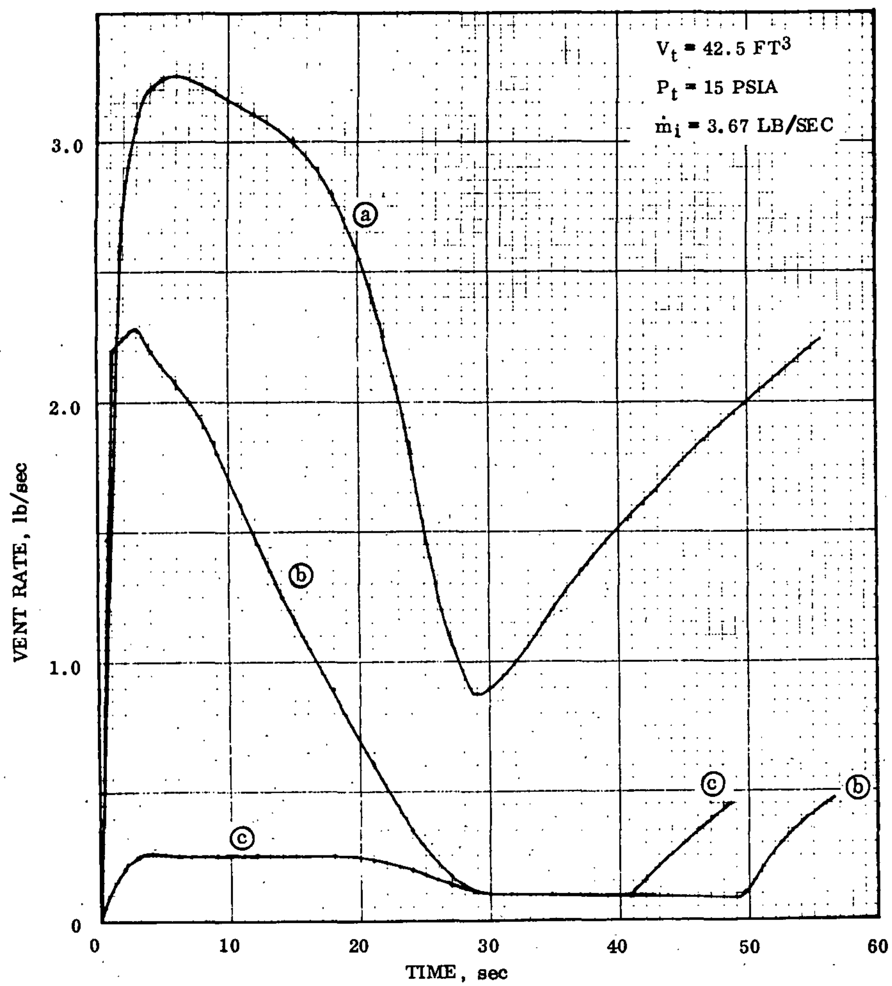


Figure 5-28. Vent Rates During H_2 Tank Childown at Constant Pressure

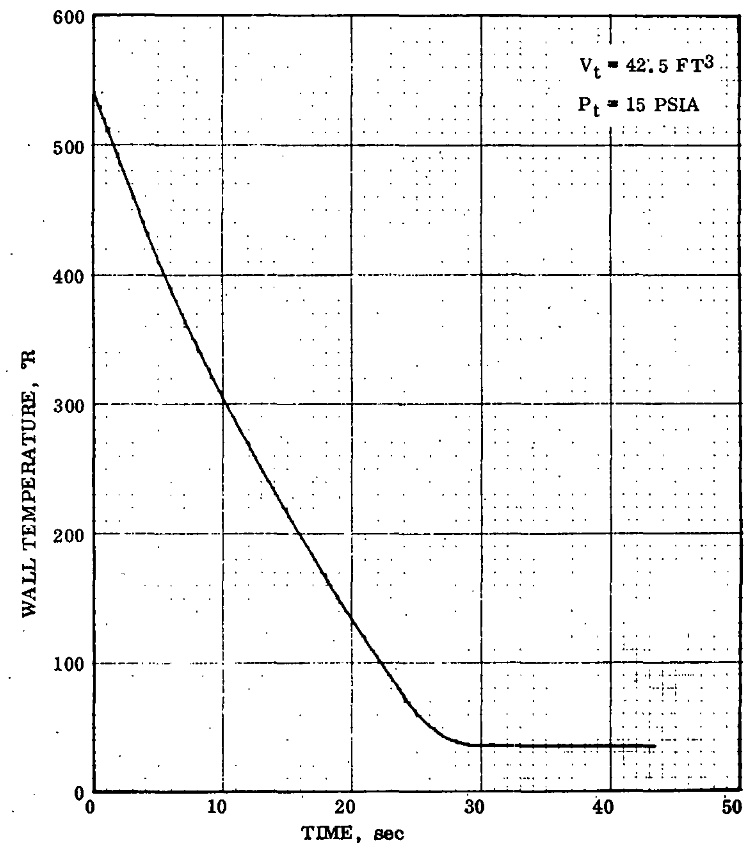


Figure 5-29. Tank Wall Temperature During H_2 Childown at Constant Pressure

Table 5-4. Vented H_2 Chilldown Data

Cond.	Mass Vented to Chill Tank lb	Time to Chill sec	Mass Vented at Fill lb	Time to Fill sec	Maximum Vent Rate lb/sec	Minimum Vent Rate lb/sec
a	72.9	28.8	∞	∞*	3.67	0.87
b	34.5	28.8	35.4	56.8	2.28	0.107
c	6.4	28.8	7.1	48.8	0.265	0.084

The above data show that a considerable variation can exist in the vent mass required and that it can be a significant proportion of the total loaded mass. That is, increasing the energy of the vent fluid can considerably reduce the required vent mass.

5.3.5 CONCLUSIONS. Based on the foregoing analyses associated with the chilldown of a receiver tank and/or transfer line, the following conclusions have been made.

a. When chilling down both the line and tank, it was found that the line chilled down rapidly in relation to the tank. The only effect of a nominal size line was to cause an initial receiver tank pressure rise which peaks after a few seconds of chilldown. This peak is caused by warm gas, vaporized in the line, initially coming into the tank. The maximum pressure reached in the tank normally occurs subsequent to this initial peak and is only affected on a bulk energy basis in proportion to the ratio of line mass to tank mass.

b. The maximum pressure reached in a locked-up (non-vented) receiver tank during chilldown and fill is a strong function of the relationship between inflow rate and the heat transfer rate between the fluid and the tank walls. In general, high inflow rates and/or correspondingly low heat transfer rates are desirable to minimize the receiver tank pressure. With respect to the receiver tank, a fill which occurs in approximately one-half the time it takes to chill the wall to fluid temperature appears to be optimum (minimum pressure rise). Fill times up to those corresponding to the chill time are, however, considered to be reasonable for producing pressure rises close to the minimum.

c. The theoretically lowest maximum tank pressure occurs at the end of the transfer or when chilldown is completed with a full tank of fluid. Bulk energy balances showed that under certain conditions of tank size and design pressures, chilldown of a locked-up tank is not feasible. That is, the maximum pressure reached cannot be maintained below the tank design pressure. It was found that the larger the tank the more likely a non-vent transfer would be feasible. Taking an aluminum H_2 tank designed for 100 psia and with an initial pressure of 15 psia the minimum size for a non-vent transfer would be 46.5 in. diameter (30 ft^3). Design pressures of 50 psi and 250 psi increase this theoretical minimum to respectively 62.5 and 61.5 inches. Starting with the tank at 0 psia also has the effect of slightly reducing the allowable size. As an example, the 100 psi H_2 bottle could go from 46.5 to 44.5 inches. Use of a 100 psi CRES bottle instead of aluminum allowed a further reduction to 37 inches diameter. Data for O_2 showed that locked-up transfer was feasible for all tank sizes (down to 25 inch diameter) and materials (aluminum and CRES) studied.

d. In order to meet the high inflow rates required to limit actual receiver pressures to theoretical minimums, the line sizes can become large. These large line sizes will then increase the maximum tank pressure. As an example, the limit on line diameter for a 100 ft line supplying a 42.5 ft^3, 100 psi design, aluminum sphere with hydrogen would be 1 inch. However, analysis showed that for this length line to provide the required flow rate with reasonable inlet pressures a size on the order of 1.6 inches would be needed. Again, the larger the tank size, the less this effect would be.

e. Calculations for the chilldown of a warm line with a cold receiver tank containing residual liquid showed that the pressure rise could be significant. Thus a vented receiver tank might be desirable for this case to minimize the required supply and receiver design pressure requirements.

f. For the case where receiver tank venting was accomplished during tank chilldown it was found that the condition of the fluid being vented overboard had a significant effect on the vent quantity required. Also, this mass could be a significant proportion of the loaded mass (up to 50% for the conditions considered). It was found that venting fluid at the bulk tank conditions required approximately twice the amount required when venting saturated vapor and the venting of vapor heated to tank wall temperatures required only about one-fifth of that for the saturated vapor case.

g. Due to the uncertainties associated with transfer to a locked-up receiver, the use of vented systems should be considered for most future applications, especially where fairly small tanks and low design pressures are involved. Also, since significant variations in vent performance are possible, the optimum design of such systems should be thoroughly investigated.

5.4 SURFACE TENSION COLLECTION

This section presents the results of work performed to develop reasonable designs and to define performance of capillary or surface tension acquisition devices suitable for the supply of subcritical fluid to space systems operating in earth orbit. Supply bottle sizes of 25 to 150 inches dia for LH_2 and 25 to 100 inches dia for LO_2 and LN_2 were considered. The analyses and preliminary designs were based on the results of a similar study performed for large H_2 and O_2 tanks as reported in References 3-1 through 3-3.

Three basic configurations were considered in the present study; (a) double screen liner, (b) single screen liner, and (c) screened channels. These three systems are illustrated in Figures 5-30, 5-31 and 5-32. Definitions as to screen material, fluid containment and thermal performance and fluid residual calculations associated with these basic concepts are presented in the following paragraphs.

5.4.1 SYSTEM DEFINITIONS. Initial calculations were performed to determine limitations and/or unique requirements imposed on the design of the surface

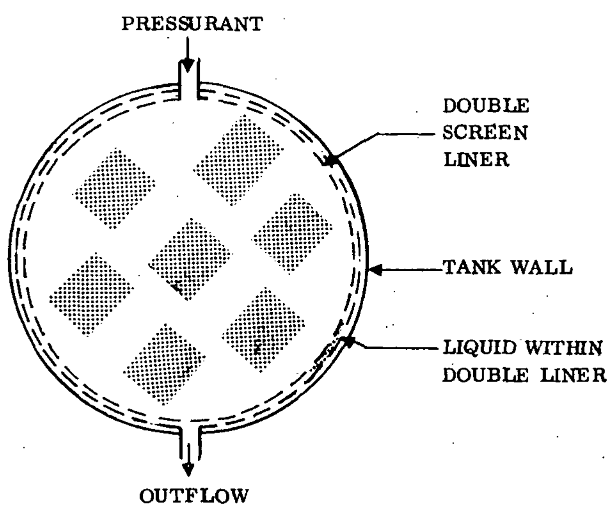


Figure 5-30. Double Screen Liner Configuration

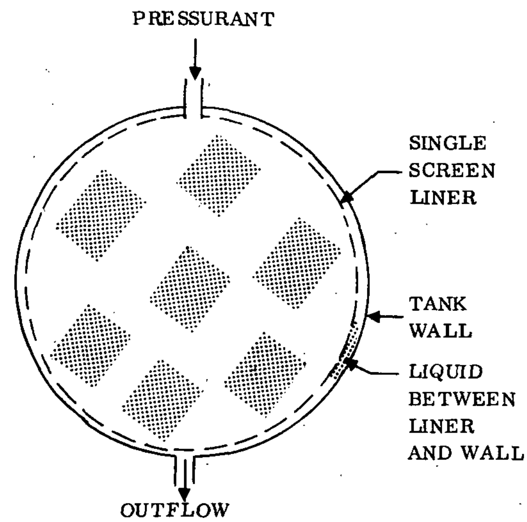


Figure 5-31. Single Liner Configuration

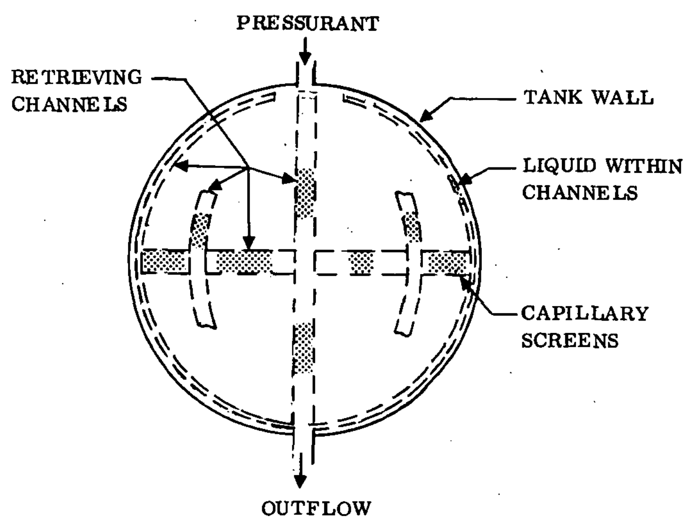


Figure 5-32. Channel Surface Tension Configuration

tension system by disturbing accelerations up to 10^{-4} g's. Satisfactory operation of the surface tension concept requires that vapor not break through the screen in a manner such that a direct path could exist between the vapor ullage and the tank outlet. To prevent this breakthrough from occurring, the screen must remain wetted at all times during the outflow. The bulk tank liquid must also be in contact with the screen liner or channel surface during the outflow. The configurations being considered (Figures 5-30, 5-31 and 5-32) are located at or near the tank wall so that under low acceleration or Bond number conditions liquid will exist somewhere at the screen. Conditions required for liquid to remain at the wall during imposed acceleration levels are discussed in Reference 5-8. An examination of the test data contained in this reference indicates that for Bond numbers less than 2.0 the liquid would remain at the tank wall and any orientation maneuvers at these low Bond numbers would only result in liquid flow along the wall. For this application the Bond number is defined as

$$Bo = \frac{(\rho_L - \rho_V) R_t^2}{\sigma} \left(\frac{a}{g_c}\right)$$

Bond number calculations were made for H_2, O_2 and N_2 cases using the saturated properties data presented in Table 5-5. Bond numbers corresponding to an acceleration of 10^{-4} g's are plotted in Figure 5-33 as a function of tank diameter. From this data it is seen that none of the fluids have Bond numbers less than 2 at 10^{-4} g's for the bottle sizes under consideration. For disturbing accelerations of 10^{-5} and 10^{-6} g's Bond numbers would be greater than 2 and liquid should exist at the wall under the conditions illustrated in Table 5-6.

Table 5-5. Properties Data Used in Bond No. Calculations

Fluid	$P_{sat.}$ psia	ρ_L lb/ft^3	ρ_V lb/ft^3	σ dynes/cm
H_2	20	4.34	0.11	1.8
H_2	100	3.54	0.55	0.62
O_2	20	70.3	0.4	12.55
O_2	100	64.2	1.7	7.8
N_2	20	49.6	0.4	8.2
N_2	100	43.6	1.8	4.36

Table 5-6. Required Conditions for Liquid at the Wall

Fluid	$P_{sat.}$ psia	Max. Dia, inches for Liquid at the Walls 10^{-5} g's	10^{-6} g's
H_2	20	58	> 150
H_2	100	40	130
O_2	20	38	120
O_2	100	31	100
N_2	20	36	114
N_2	100	29	90

It is noted that normal orbital drag would be on the order of 10^{-6} g's which would allow the Bond number criteria to be met for tank sizes of interest. The system operation would also be satisfactory if acceleration disturbances were higher, but in only one direction during the transfer.

If adverse disturbing accelerations of greater than 10^{-6} g's are to occur during the final stages of the transfer, a modification to the basic designs shown in Figures 5-30, 5-31 and 5-32 would be necessary. One possibility is the use of a reservoir, such as proposed for the LO_2 tanker design presented in Reference 3-1. A typical reservoir

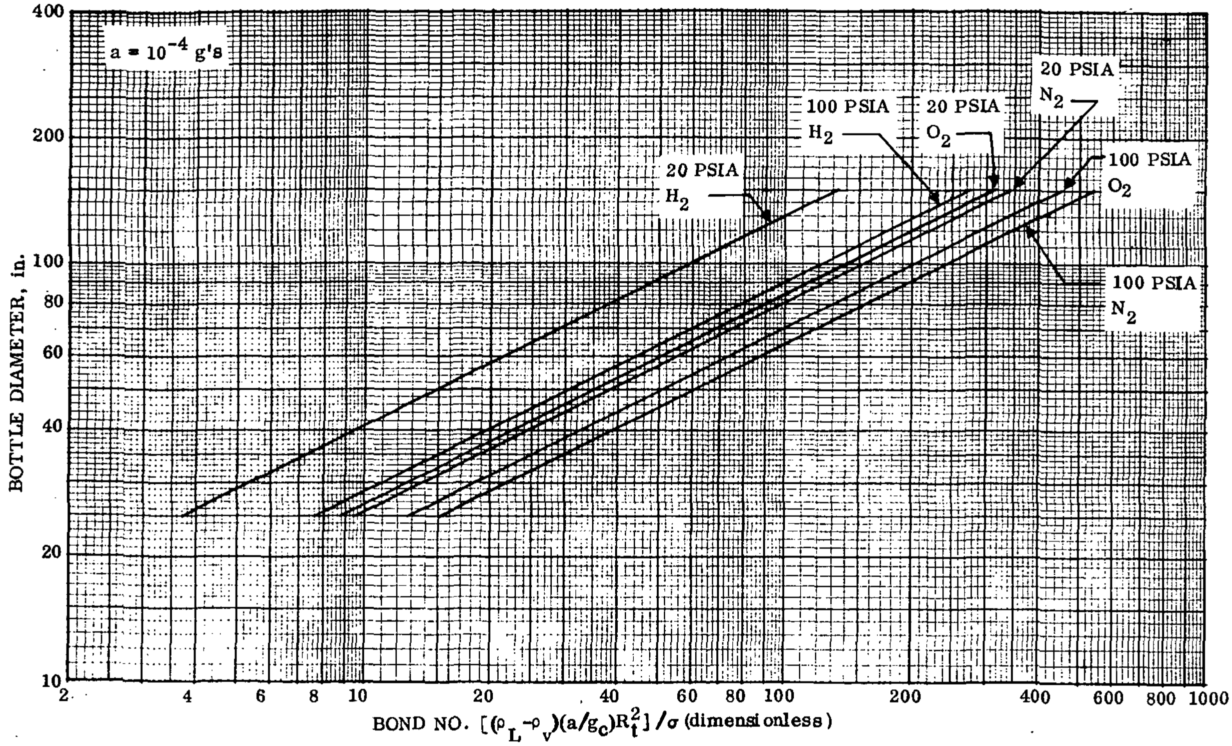


Figure 5-33. Bond Numbers as a Function of Spherical Tank Diameter

configuration is presented in Figure 3-2. The design philosophy associated with the reservoir is that during potential liquid migrations from one side of the tank to the other, the reservoir will provide a continuous liquid outlet and thus prevent vapor breakthrough during these transient periods. Reservoir sizing is based on providing this flow during the maximum time the liquid may be in transit and depends on tank size, acceleration level and normal outflow rate. Time in transit is calculated from the basic laws of motion and the applicable equation is presented below.

$$\theta = \sqrt{2s/a} \tag{5-26}$$

where

θ = time in transit

s = distance to be traveled

a = disturbing acceleration, assumed to be steady over the time interval involved

Solutions to this equation for a range of tank diameters are presented in Table 5-7.

Considering a 52 inch diameter tank and a nominal transfer time of 720 seconds (12 minutes) with a disturbing acceleration of 10^{-5} g's, the reservoir volume would need to be

Table 5-7. Time for Liquid to Travel Across a Spherical Tank

Tank Dia s, in.	Time, sec. $a = 10^{-4}$ g's	$a = 10^{-5}$ g's
25	35.9	113.5
52	52.0	164.5
150	88.0	278.0

$$V_R = V_t \left(\frac{164.5}{720}\right) = 0.228\ V_t$$

or approximately 23 percent of the total tank volume. This would result in a high system weight penalty. Also, it is noted that the use of the reservoir would only be applicable to a single disturbance, since there would be no way to refill the reservoir during the low-g transfer. Furthermore, the above analysis assumes that the liquid traveling across the tank will immediately flow into the channels upon contact and there are some uncertainties in this assumption due to potential geysering at the high Bond numbers.

Another approach analyzed was where the tank was assumed to be divided into compartments such that the liquid within each compartment would be at the respective walls and thus available for transfer, even under adverse disturbances up to 10^{-4} g's. The following configurations were considered; (a) spherical concentric liners, (b) planar screens compartmenting the tank, (c) perpendicular planar screens, and (d) spherical concentric liners with planar screens. In the analysis, a 52-inch dia bottle operating at 100 psia during discharge was assumed. Initial calculations showed that the use of planar screens was inefficient for a spherical tank because of the uneven gaps and increased surface area and numbers of screens required over that of just concentric spheres. The system chosen as best for this present application is illustrated in Figure 5-34. The configuration shown has interconnections between the concentric channels which allows the flow to come from any liner to the outlet. The system weight is taken to be proportional to the sum of the surface areas of the screens. The minimum number of screens required is determined from the following equation.

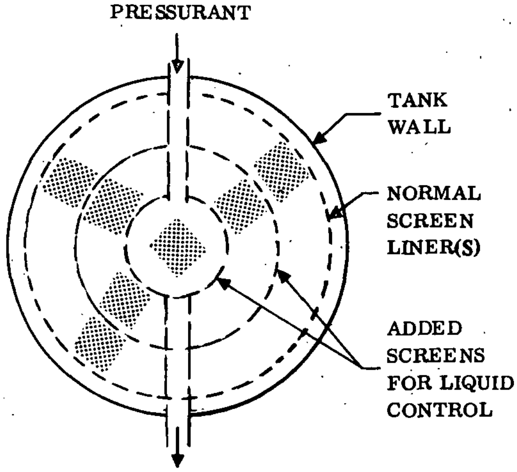


Figure 5-34. Use of Concentric Spheres to Control Liquid for Transfer

$$n = \frac{R-r}{2r} + 2$$

where, from the Bond number stability requirement of 2.0,

$$r = \sqrt{2\sigma/(\rho_L - \rho_V)(a/g_c)}$$

and n includes the basic screen liner(s) near the wall.

R = inner radius of liner(s) at the wall.

The total area of all screens is then

$$A_T = 4\pi \left\{R^2 + [R-r]^2 + [R-2r]^2 + \text{---}[R-(n-1)r]^2\right\} \text{ for a single liner at the wall.}$$

$$A_T = 4\pi \left\{2R^2 + [R-r]^2 + [R-2r]^2 + \text{---}[R-(n-1)r]^2\right\} \text{ for a double liner at the wall.}$$

On this basis, required weights or screen areas for a 48-inch dia tank were found to be increased by 1.4 and 1.8 times when applied to the basic double and single liner systems shown respectively in Figures 5-30 and 5-31.

It thus appears desirable to transfer fluid during a time when disturbing accelerations significantly above 10^{-6} g's would not be applied. This is reasonable since transfer times for this type of system can be made to be short (on the order of a few minutes for a 42.5 ft^3 bottle) and also the main problem with disturbing accelerations would only be during the last phase of the transfer when the tank was nearly empty.

Further analyses were thus performed on the basic systems as presented in Figures 5-30, 5-31 and 5-32. Estimates of required screen pore sizes were made using the following equation.

$$D_{BP} = \frac{4\sigma}{\rho_L \, (a/g_c) \, D_t}$$

where D_{BP} = minimum or bubble point capillary diameter. Assuming the screen is required to hold liquid at 10^{-4} g's, the resulting maximum values for D_{BP} are presented in Table 5-8.

The micron sizes presented in Table 5-8 are all high and thus the 10^{-4} g's do not create significant restrictions on screen geometry. The final screen size is determined from the best combination of head retention and pressure drop for minimizing residuals and maximizing the allowable outflow rate. Previous studies (Reference 3-1) showed that the use of a 200 × 600 mesh screen would be optimum for this requirement.

Table 5-8. Maximum Screen Size for 10^{-4} g's Head

Bottle Size (Dia, in.)	Fluid	Maximum Screen D_{BP} (microns)
25	LH_2	18.1×10^4
25	LO_2	7.54×10^4
25	LN_2	6.99×10^4
50	LH_2	9.05×10^4
50	LO_2	3.77×10^4
50	LN_2	3.00×10^4
150	LH_2	3.02×10^4
150	LO_2	1.26×10^4
150	LN_2	1.00×10^4

The next step in the analysis was to determine the requirements for thermal conditioning to prevent excessive heating from forming vapor within the channels and/or drying out the screen surface. For the double screen liner system, calculations were made to determine if wicking would be sufficient to make up any evaporation which could occur at the screen surface. Based on the expected heating conditions, as determined in Section 5.1 for a locked-up tank without venting or mixing, it was found that the 200 × 600 mesh screen would be capable of providing the necessary wicking. Thus no significant modifications would need to be made to the system shown in Figure 5-30.

In the case of the single screen liner some active means must be provided to prevent vapor formation at the walls of the tank prior to transfer. Cooling of the entire wall, such as with a heat exchanger vent, was not considered practical for the present application. The approach taken was to use a mixer to maintain low pressure rise rates in conjunction with providing a full tank of liquid prior to transfer. Analyses performed in Section 5.1 showed that for initial ullages less than 18 percent, a pressure rise to 100 psia under mixed non-vent conditions would result in a full tank of liquid. Thus the liner would be assured of being full of liquid and transfer accomplished in a manner as presently used for non-cryogenic systems. The required addition to the basic system shown in Figure 5-31 for thermal conditioning would then be a mixer device.

In the case of the channel system, a vent providing active cooling can be used to control heat leak into the channels. This type of system was analyzed in Reference 3-1 and would consist of adding to the Figure 5-32 system a bulk mixer and cooling coils at the tank wall next to each of the surface tension channels. In this case the mixer is required for efficient tank pressure control while venting and not to insure a liquid full tank. The potential advantage of this system over the other systems is that the tank pressure can be controlled at lower pressures and where applicable the tank weights can be reduced. The primary application for this system would be for longer duration missions where use of a locked-up tank is undesirable from a weight standpoint and the increased complexity of the vent system is warranted.

Weights for the three surface tension systems initially presented in Figures 5-30, 5-31 and 5-32 are given in Figure 5-35 as a function of tank diameter and include mixers and wall heat exchangers where required. Weights for the basic screens and support structures are based on data from Reference 3-1 and are assumed to be the same for H_2, O_2 and N_2 applications.

Analyses to determine the residuals associated with the various surface tension configurations are presented in the following paragraph.

5.4.2 <u>LIQUID RESIDUALS</u>. Residuals were determined for both LH_2 and LO_2 tanks, encompassing the full range of anticipated flow rates and tank sizes. The single liner, used as the reference configuration, was analyzed using a computer program described in Reference 4-1 which was developed specifically for determination of surface tension system residuals.

For the LH_2 system, three tank diameters, 25", 52", and 150" were analyzed at volume outflow rates of 0.07083, 0.7083 and 2.125 ft^3/sec for each bottle. Additional flow rates were run for each bottle to define the limits of operation. Liner spacing to provide minimum residuals was determined for each size and flow rate. Minimum liner spacings used, based on fabrication considerations, are shown in Figure 5-36. The minimum residuals

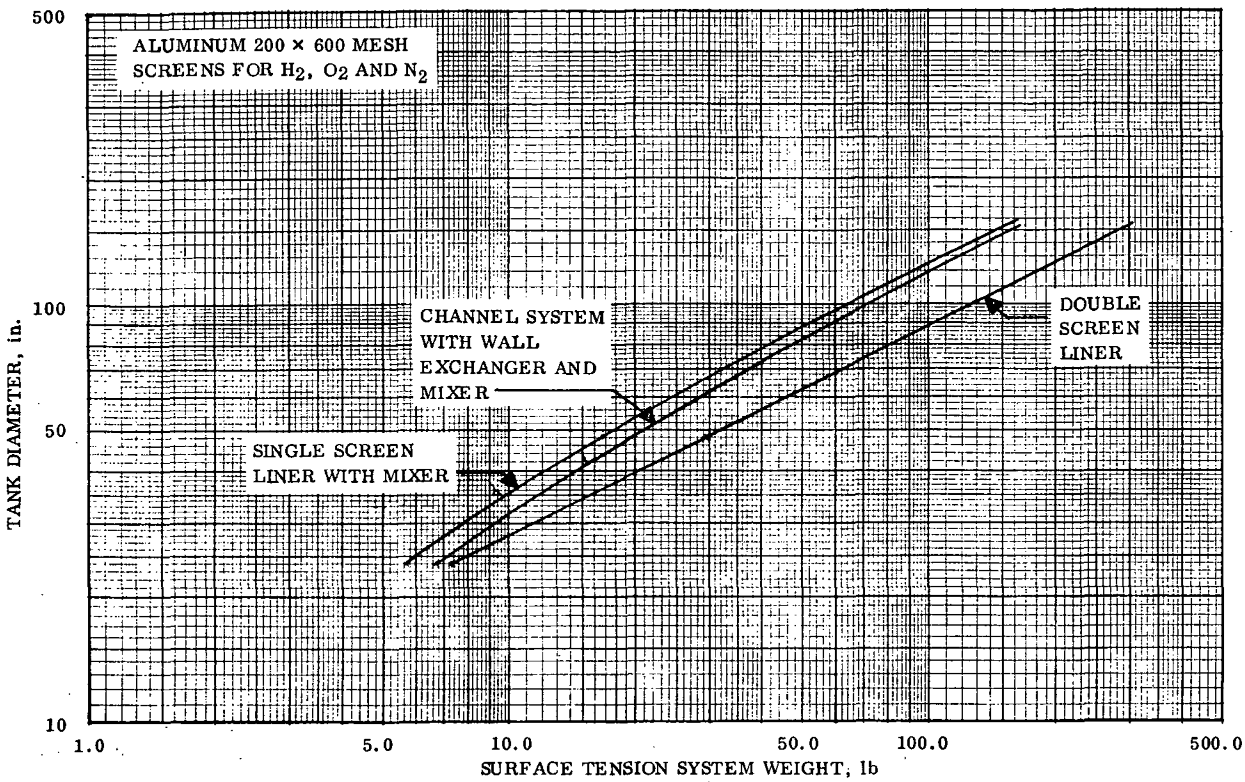


Figure 5-35. Surface Tension System Weights

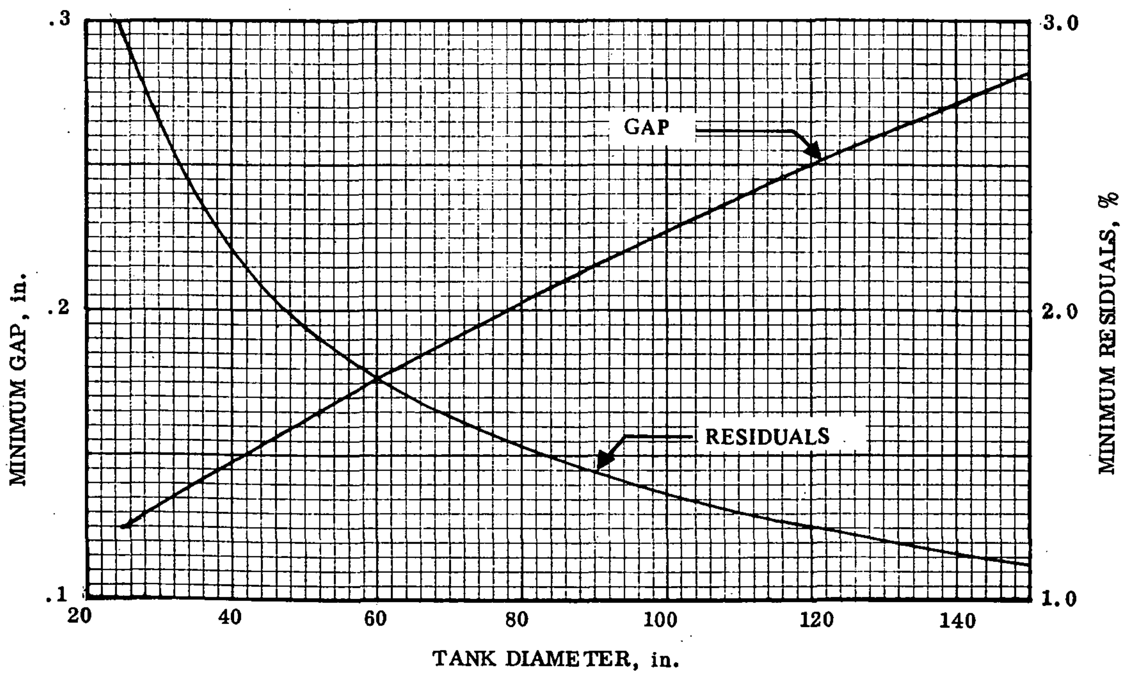


Figure 5-36. Minimum Gap and Residual Requirements for S.T. Screen Systems

presented in Figure 5-36, assume the liner to be full at the end of transfer.

The basic model used for predicting residuals was the single liner with liquid in the tank at the opposite end of the tank from the outlet and the tank divided into eighteen channel segments for pressure drop calculations. The pressure drop in the capillary device is taken as the sum of the screen, channel, and momentum losses. When the pressure drop in the capillary device is equal to the surface tension retention head of the capillary device screen, breakthrough of vapor into the capillary device is imminent. The computer program developed for computing tank residuals with a single liner capillary device determines, by iteration, the liquid surface area which must be wetted by the pool in order to produce the surface tension pressure drop in the liner for a given liner spacing and tank outflow rate. Residuals are taken as the volume in the liner plus the pool residuals. For simplicity, pool residuals are computed for a flat interface which yields to higher residual predictions than would actually occur. Since the optimum (minimum) residual case for each condition analyzed had low pool residuals, this assumption does not appreciably affect minimum residual predictions.

Both pool and liner residuals were determined for each case. The minimum spacing yielded negligible pool residuals for the low flow rate case for each size tank and thus represent minimum residuals. For the 0.7083 ft^3/sec flow rate, minimum spacings are optimum for the larger bottles while spacing had to be increased above the minimum for the 25 inch bottle to achieve minimum residuals. For the non-minimum spacing cases, runs were made parametrically as a function of spacing. As spacing is increased above the minimum, liner residuals increase, while pool residuals decrease. The optimum point is reached where an increase in liner residuals is not offset by a decrease in pool residuals. Figure 5-37 illustrates a typical set of parametric data used to determine optimum spacing for the smallest tank size analyzed, at the 2.125 ft^3/sec flow rate. The data illustrate the trend of decreasing pool residuals and increasing liner residuals as spacing is increased above the minimum spacing. Optimum spacing for this case was found to be 0.75 inches with residuals of 1.0 ft^3 of LH_2. These types of calculations were employed for non-minimum spacing to develop the curves of Figure 5-38. These data illustrate the spacing which yields minimum residuals for each case considered. As discussed previously, minimum spacings per Figure 5-36 yield optimum residuals for all bottles at 0.07083 ft^3/sec, for the 52 inch and 150 inch diameter bottles at .07083 and 0.7083 ft^3/sec and for the 150 inch diameter bottle up to 4.25 ft^3/sec. Flow rate limitations for each bottle size were determined for minimum spacing, as shown typically in Figure 5-39. Pool residuals increase as flow rate is increased due to increased pressure drop in the capillary device. Pool residuals increase sharply after 5 lb/sec indicating that an increase in spacing is required to minimize residuals for flow rates above this value.

Minimum residuals are shown for the LH_2 tanks in Figures 5-40 and 5-41. Figure 5-40 shows residuals as a function of volume rate for the three bottle sizes analyzed.

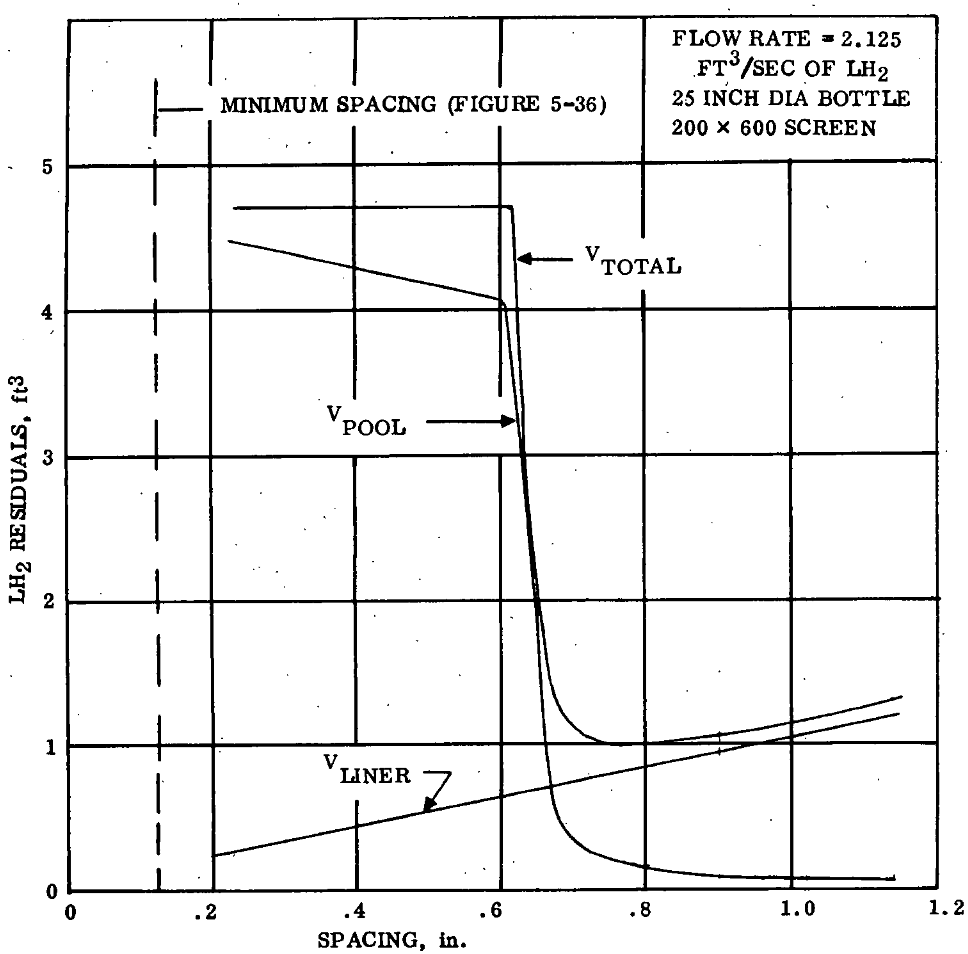


Figure 5-37. Typical Case of LH_2 Residuals Vs Spacing

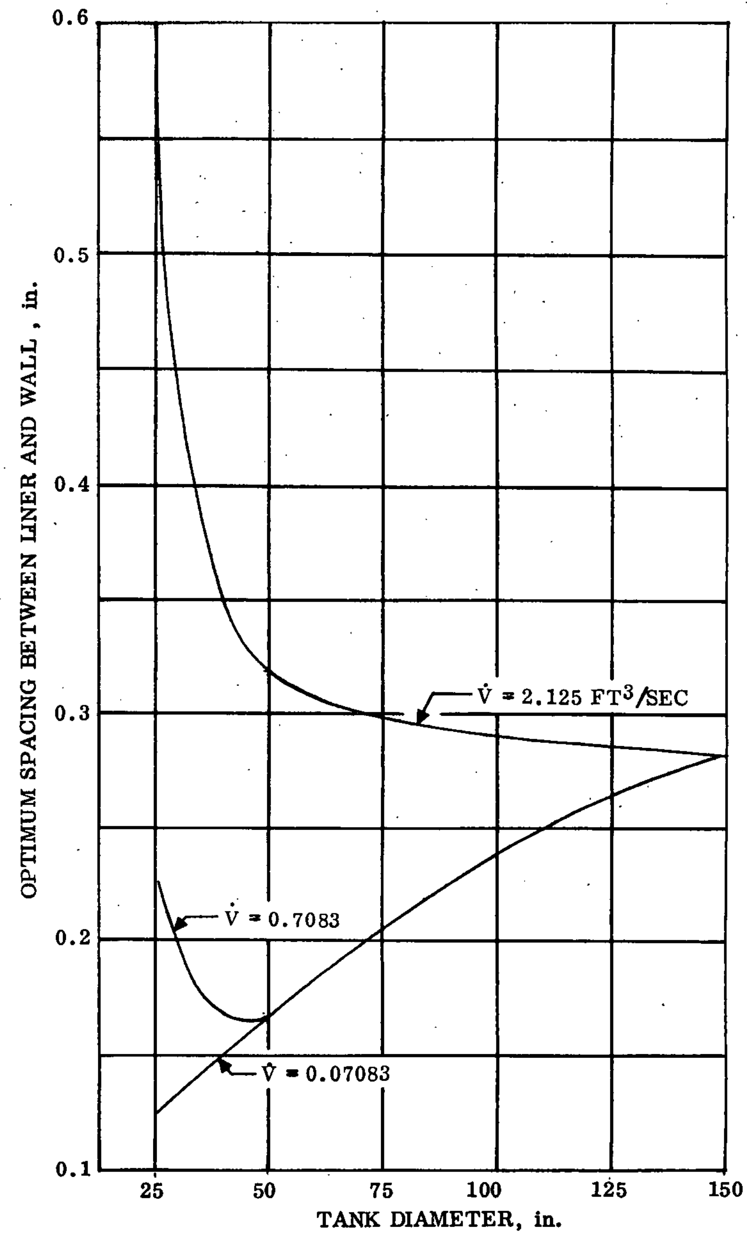


Figure 5-38. Single Liner Spacing for Minimum LH_2 Residuals

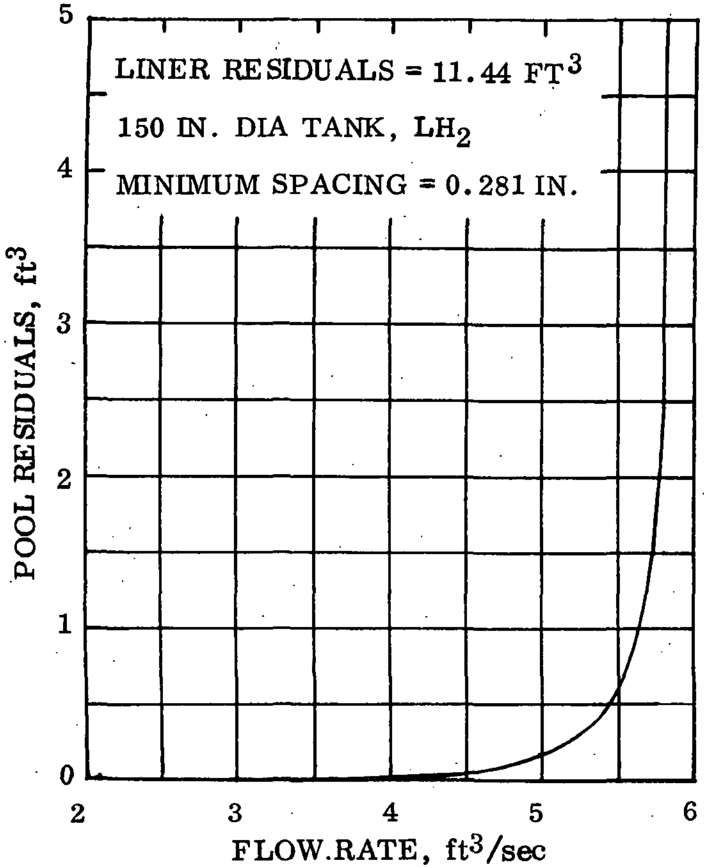


Figure 5-39. Typical Case of Flow Rate Vs Residuals for Minimum Spacing

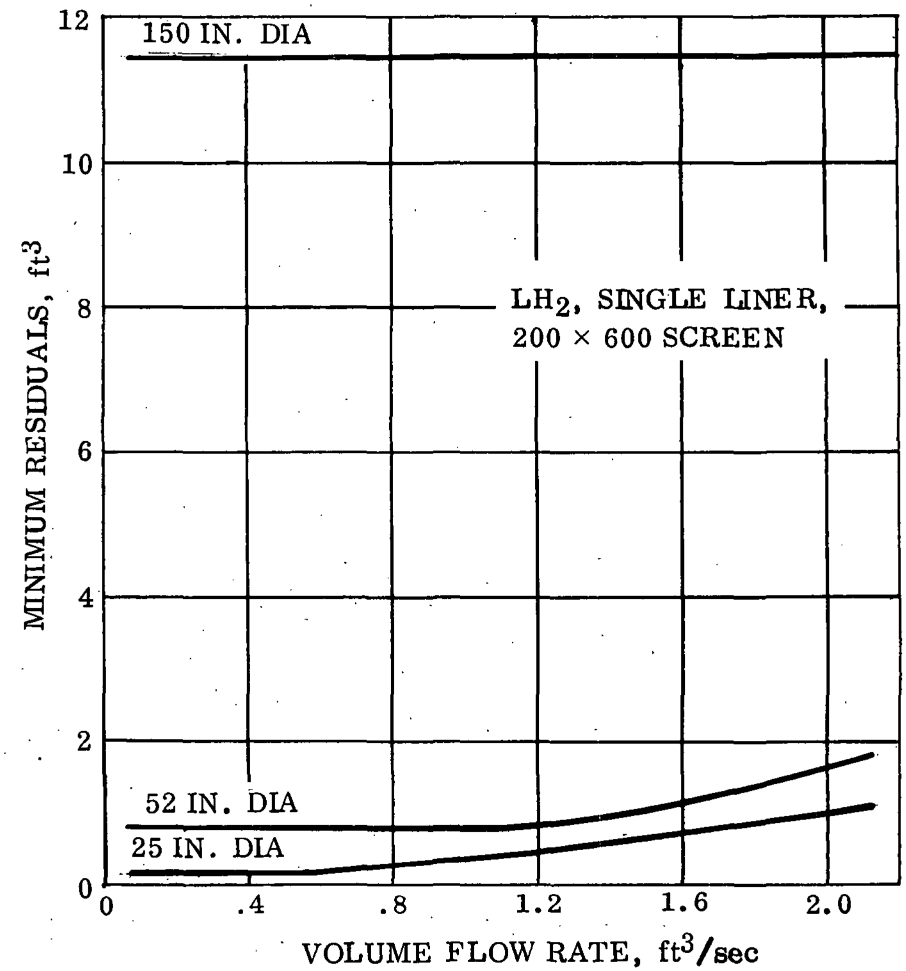


Figure 5-40. Minimum Residuals Vs Flow Rate

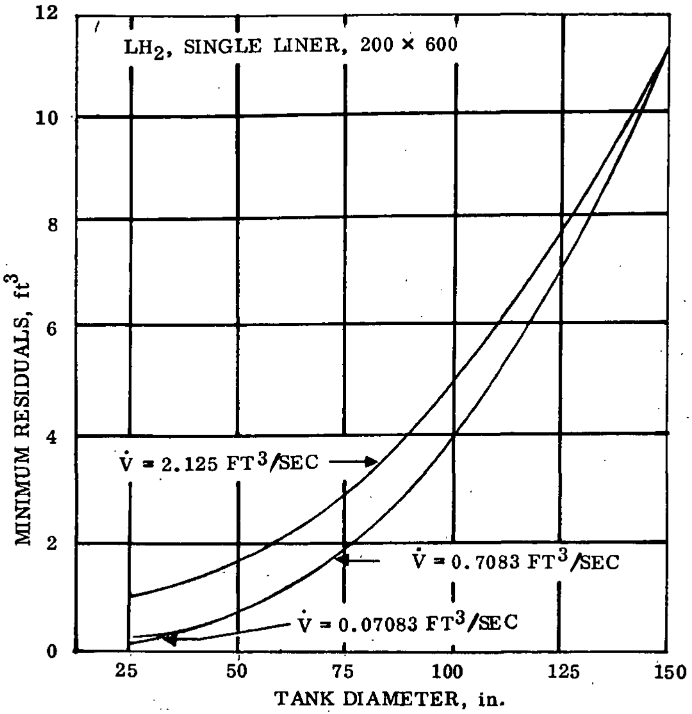


Figure 5-41. Minimum Residuals Vs Tank Diameter

Residuals for the large bottle were the same for all flow rates considered up to 4.25 ft^3/sec. Figure 5-41 shows the minimum residual data plotted as a function of tank diameter for the three flow rates considered. Figure 5-42 shows the variation in percent residuals with flow rate for several bottle sizes. Percent residuals increase for constant flow rate, as bottle size is decreased. Residuals range from about 1 percent to 22 percent for the cases considered. For the large bottle, higher flow rates (up to 9.5 ft^3/sec) were analyzed. Results are shown in Figure 5-43 in terms of percent residuals vs flow rate.

Based on the data from Figures 5-42 and 5-43, the maximum flow rates which would still allow minimum residuals were determined as a function of tank diameter. This information is presented in Figure 5-44 along with corresponding times to empty a full tank at these maximum flow rates.

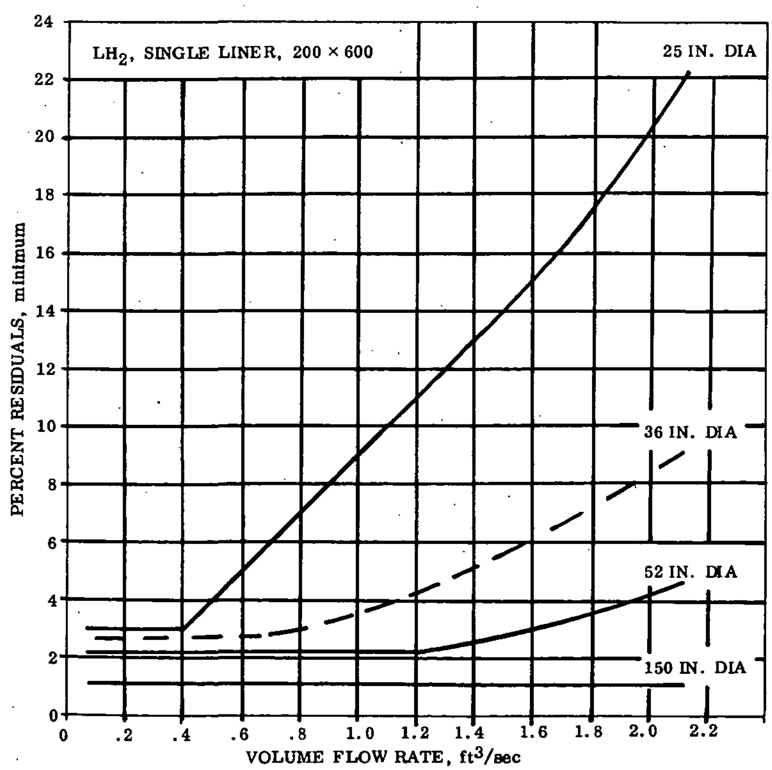


Figure 5-42. Minimum Percent Residuals Vs Flow Rate

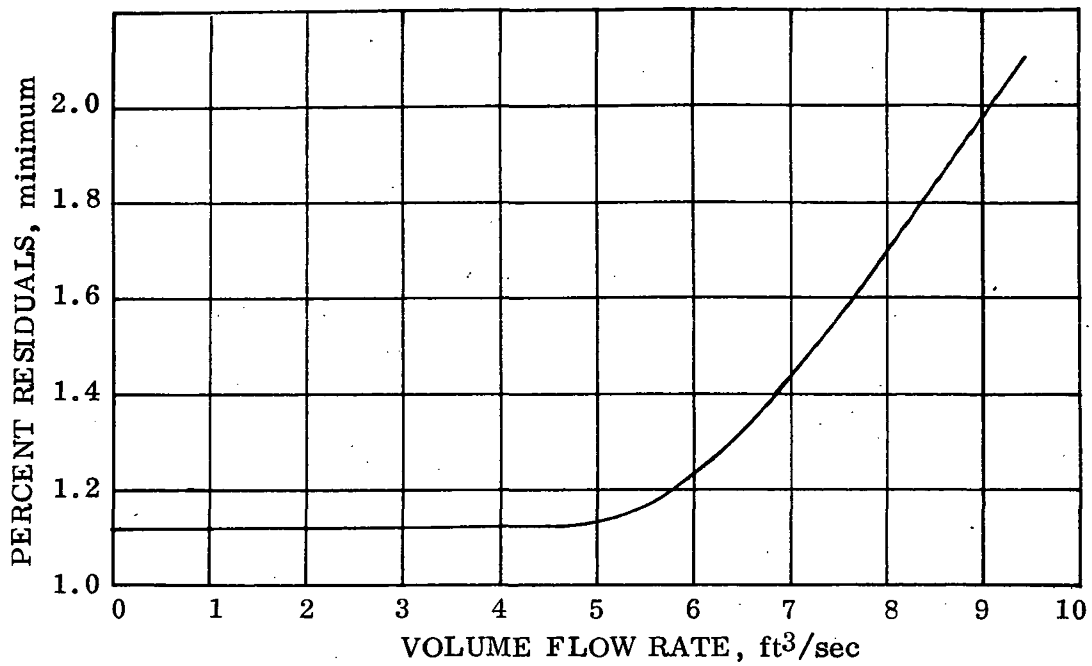


Figure 5-43. Percent Residuals Vs Volume Flow Rate for 150 In. Diameter Tank

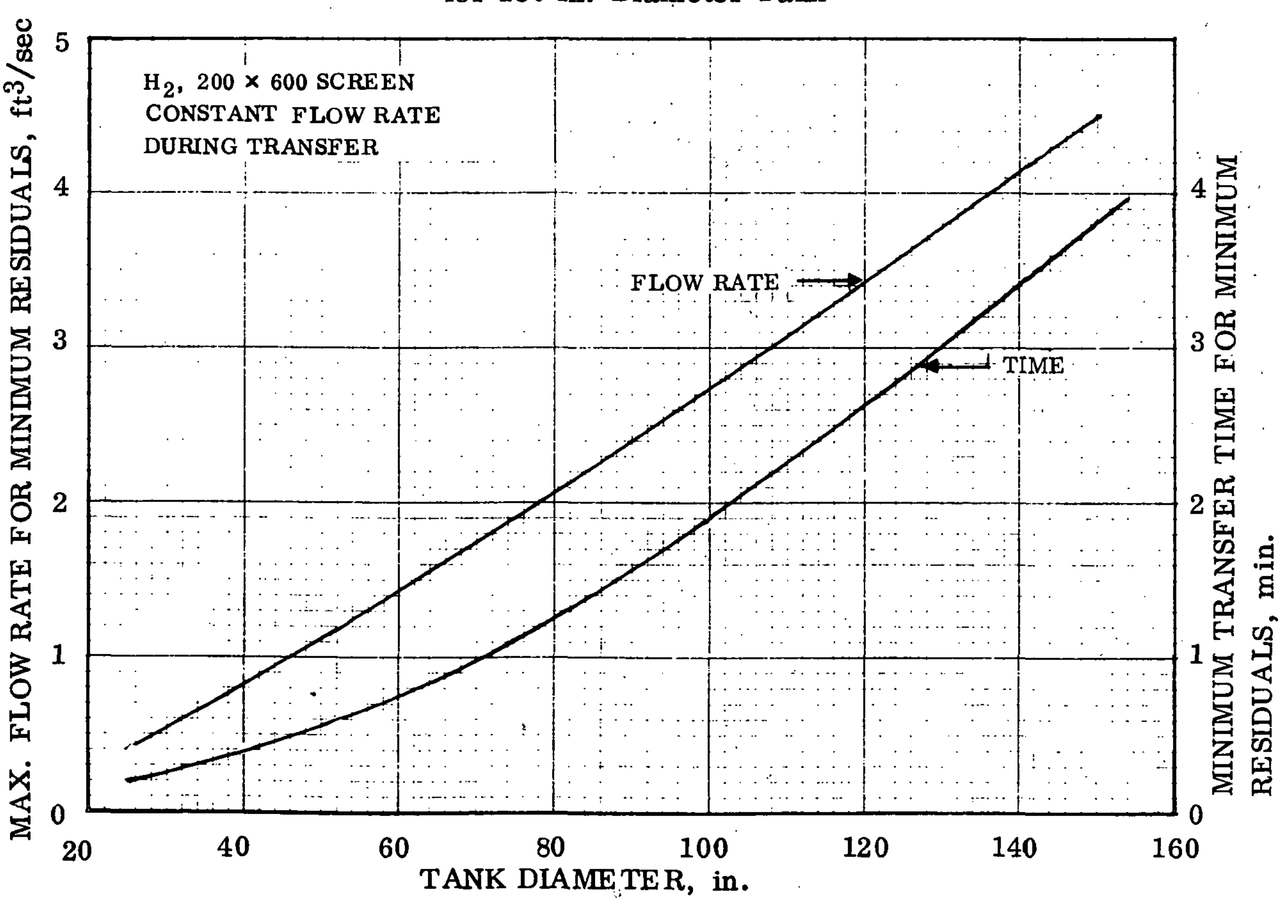


Figure 5-44. Maximum Flow Rates for Maintaining Minimum Residuals

In order to compare the transfer of LO_2 with the LH_2 cases, runs were made for a 52-inch diameter tank with LO_2 at 0.07083, 0.7083 and 2.125 ft^3/sec outlet flow rates. Results are shown in Figure 5-45 as minimum residuals and corresponding liner spacing vs flow rate. Spacing and volumetric residuals are greater for LO_2 than for LH_2 by a factor of approximately 4 to 5 for spacings above minimum. This can be seen by comparing Figures 5-40 and 5-45. The increased residuals with LO_2 are due to lower LO_2 retention properties as indicated by lower surface tension to density ratio and by higher pressure drops at the same volume flow rate due to the higher density and viscosity of LO_2 as compared to LH_2.

Residuals were also determined for the channel type capillary device (Figure 5-32) which is similar in concept to the channel design of Ref. 3-1. Residual determinations were made for minimum and non-minimum spacing based on modification of the liner results. Residuals for the minimum spacing case are based on using 25 percent of the liner residuals plus pool residuals between the channels. For non-minimum spacing, channel flow areas were made equivalent to liner flow areas at the tank equator. Pool residuals were computed as in the minimum spacing case. Results are shown in Figure 5-46 for both liner and channels. For the minimum spacing cases, the lower channel volumes resulted in lower residuals than the liner, while for non-minimum spacing, greater pool residuals resulted in channel residuals higher than for the liner.

For the double screen liner (Figure 5-30), residuals should be quite similar to the single screen liner residuals, with pool residuals decreasing somewhat due to a higher surface area exposed to the liquid. Typical residual data for the double liner case, for a tank diameter of 25 in. and a flow rate of 2.125 ft^3/sec is shown in Figure 5-47. As expected, the data is similar to that obtained for the single liner as presented in Figure 5-37.

5.4.3 CONCLUSIONS. Based on the data generated in the foregoing paragraphs, a listing was made of the relative advantages and disadvantages of the three basic surface tension systems illustrated in Figures 5-30, 5-31 and 5-32. This information is presented in Table 5-9. In summary, the double liner system is the simplest and the heaviest while the single liner and channel systems have lower weight with increased complexity.

5.5 BELLOWS SUPPLY SYSTEM

This section contains a summary of the design and development of weight data associated with a positive expulsion system using a metallic bellows. The basic concept is illustrated in Figure 3-5. The design tasks, as reported herein, were to define a reasonably optimum configuration, including pressurization storage and thermal control provisions, and then to develop detailed weight and size data over a range of propellant volumes corresponding to spherical tank diameters of 25 to 150 inches and expulsion pressures of 50 to 500 psia. Details are presented in the following paragraphs.

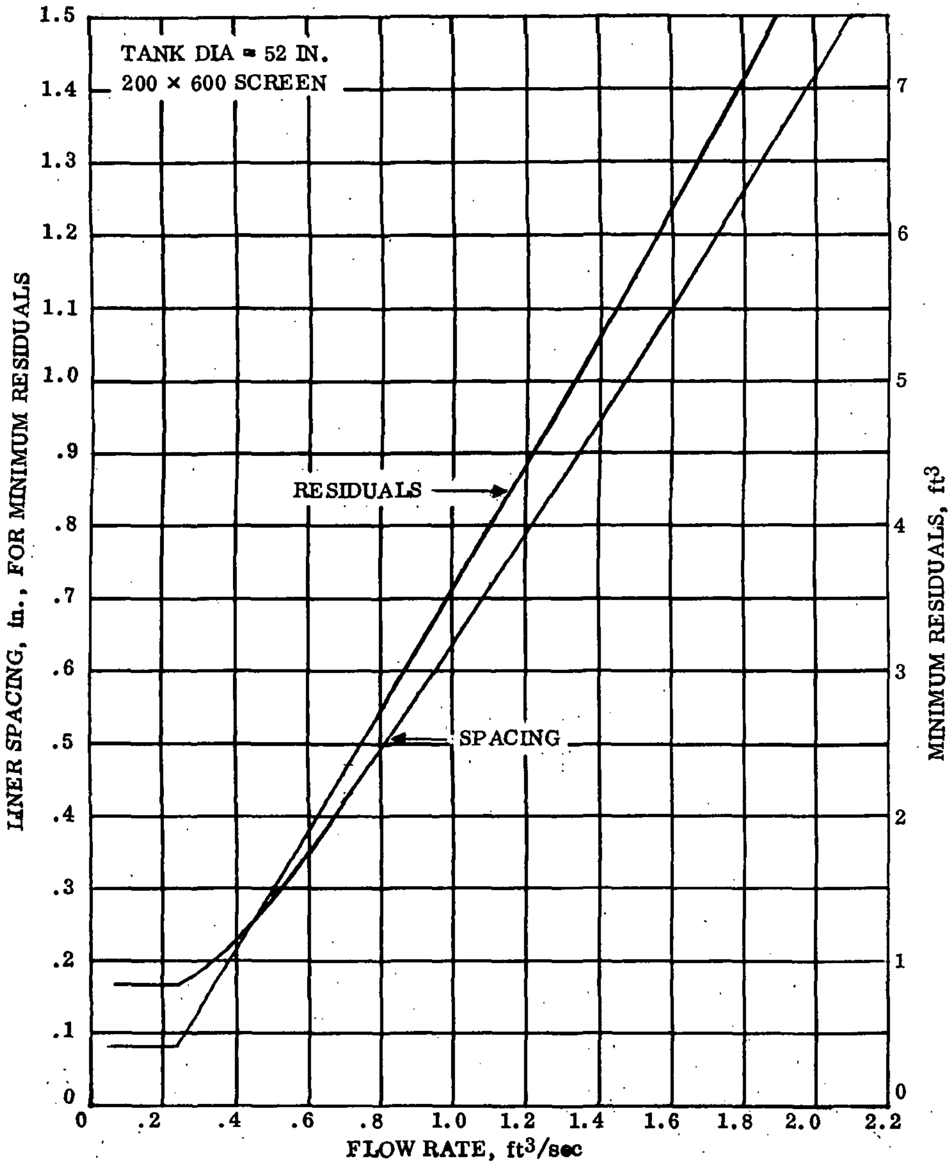


Figure 5-45. Minimum Residuals for LO_2 Transfer

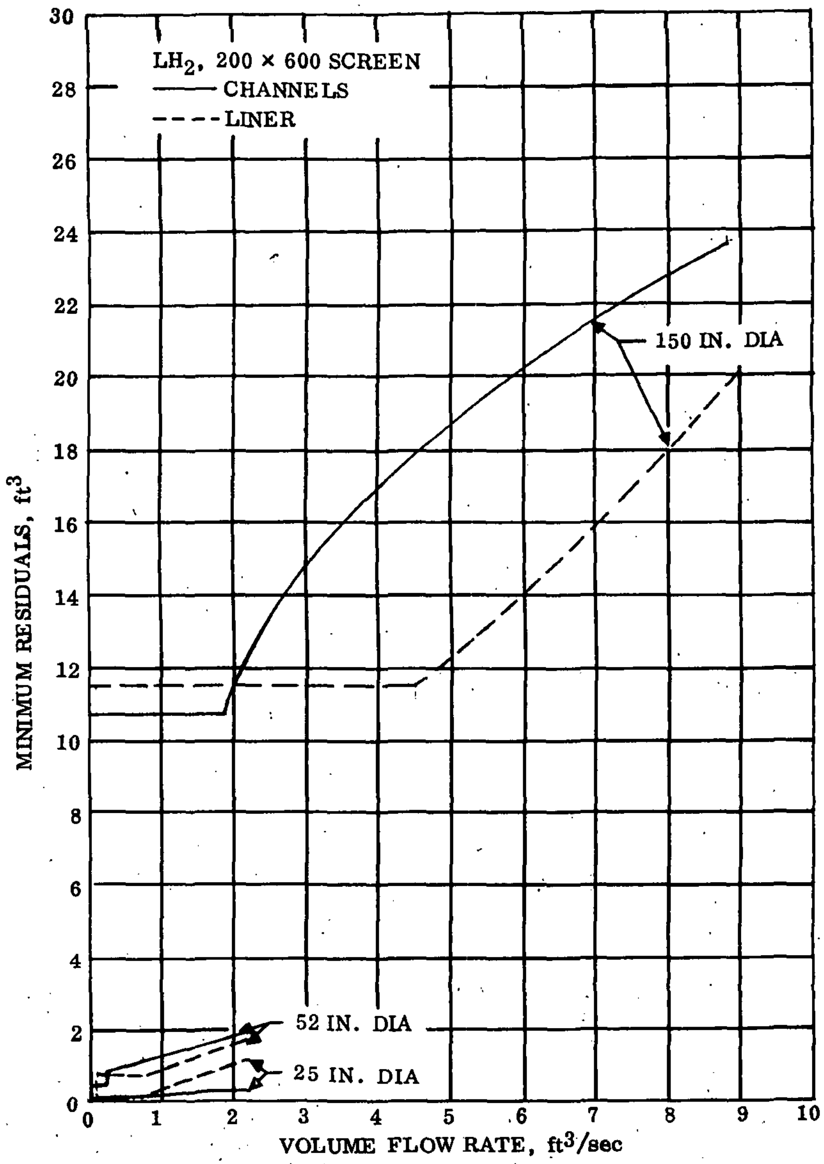


Figure 5-46. Comparison of Liner and Channel Capillary Device Residuals

Table 5-9. Advantages and Disadvantages of Several Capillary Transfer Systems

Concept	Advantages	Disadvantages
(a) Double Liner (Fig. 5-30)	Simplicity. Slightly lower residuals than (b) at non-minimum conditions. Passive. Allows use of locked-up tank.	Fairly high weight. Wicking paths required at all supports and penetrations.
(b) Single Liner (Fig. 5-31) Plus Mixer	Lowest weight. Special wicking paths not required. Lower residuals than (c) at high flow rates. Allows use of locked-up tank.	More complex than (a). External heating controlled to allow 100% liquid fill without excessive pressure rise. Larger minimum residuals than (c).
(c) Channels (Fig. 5-32) Plus Mixer & Cooling Sys. Using Vent Fluid	Lower weight than (a). Also used to control tank pressure and thus may have lower tank wt. Not sensitive to small changes in external heating. Lower min. residuals at low flow rates.	Most complex of the three. Some special wicking paths required. Requires continuous operation prior to transfer.

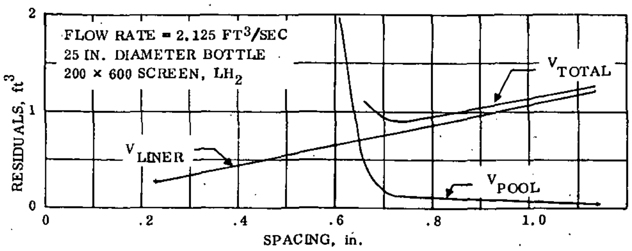


Figure 5-47. Typical Double Screen Liner Residuals Versus Spacing

5.5.1 SYSTEM DEFINITION. As part of this system definition task, different bellows configurations were considered. Each system was designed to contain a total liquid volume of 42.5 ft^3 and be capable of expulsion into a receiver tank at 100 psia. Safety factors of 1.33 on allowable and 1.67 on ultimate were used in all designs. Also, since CRES bellows are the best candidates under the present state-of-the-art, the basic pressure vessel was assumed to be 347 CRES for compatibility of required welding processes between the tank and bellows.

Analysis of both pressurization and thermal control aspects were initially required to obtain a basis for pressure vessel design. One of the trade-offs is whether the pressurant bottle should be located external to the system or packaged as part of the bellows tankage. Also, the effects of overall tank volume efficiency on total weight is important and thus the weight of excess pressurant storage requirements must be included in any weight comparisons. Helium stored at high pressure was assumed to be the pressurant. With respect to pressurization and thermal control requirements, the critical system would be that using liquid hydrogen; therefore, hydrogen was chosen for the baseline case.

Pressurant requirements were based on the data presented in Section 5.2. Assuming a bellows tank pressure requirement of 200 psia, the required helium pressurant storage bottle size to expell 42.5 ft^3 of H_2 gas was estimated to be about 8.8 ft^3.

Also, from Section 5.2 it was seen that the volume requirement was similar for both helium storage at ambient and for storage within the tank at H_2 temperatures. Furthermore, for similar operating conditions, the helium storage volume would be closely proportional to the actual propellant storage volume. The foregoing relations were used in evaluating the various design configurations.

Next, the basic thermal control requirements were defined. For storage up to seven days prior to use, calculations discussed in Section 5.1 for the present application indicated that for plain spherical bottles the use of superinsulation without tank venting would be optimum. Taking conditions for a 42.5 ft^3 tank with an initial ullage of 5% and an allowable pressure rise to 100 psia, the insulation thickness required would be on the order of six inches. This assumes the use of "Superfloc" high performance insulation with a weight of 1.29 lb/ft^3. An examination of the bellows operation indicates that no special thermal control provisions would be required for this system for the transfer application. Any vapor formed within the bellows could be transferred to the receiver along with the liquid and most of the vapor would normally be compressed and subsequently condensed at initiation of external tank pressurization. Tradeoffs between the various bellows configurations were thus performed on the basis of a locked-up tank with 6 inches of "Superfloc" insulation applied. Even though not optimum per the data of Section 5.1, the use of a small ullage was taken to provide a conservative estimate of insulation requirements for comparisons between the various bellows systems. The use of vacuum jacketing was assumed for consistency and predictability of thermal performance between the various systems. A sandwich core type construction similar to that reported in Reference 5-9 was used for the vacuum shell.

Other design ground rules and assumptions used in developing the overall bellows system weight data are presented below.

Bellows Material:	347 CRES
Bellows Wall Thickness:	0.010 inch
Bellows Type:	Nested-hydroform
Convolution Width:	1.0 inch

The above data were taken primarily from Reference 3-22.

The different bellows system configurations considered are discussed below. The configurations were chosen to determine the overall effects on system weight of bellows and pressurant bottle design and packaging and bellows length to diameter (L/D) ratio.

The first system analyzed is illustrated in Figure 5-48, and a corresponding weight statement is presented in Table 5-10. The bellows L/D was taken to be 2/1, which was considered to be a reasonable maximum value for the present volume requirement. The helium pressurant bottle in this configuration is designed to be part of the CRES pressure vessel and represents a significant part of the overall system weight. It is noted that a false bulkhead with a foam filler is used to minimize the pressurant required.

Since the weight of the helium storage bottle for this system was quite high, a design was developed where the bottle is separate from the tank and thus could be made from high strength titanium.

This design is presented in Figure 5-49, and a corresponding weight statement is given in Table 5-11. The primary difference between configurations A and B is

Table 5-10. Weights Statement for Configuration A. He Bottle Inside Vacuum Shell With Separate Pressure Shell

	WEIGHT, lbs	
PRESSURE SHELL		
Cylinder (40" dia, 80" long)	138.0	
Conic Adapter	23.4	
Bulkhead Transition	6.9	
Bellows Stops	6.0	
	174.3	174.3
EXPULSION CHAMBER		
Bellows (36" dia × 72 Eff)	67.5	
Aft Bulkhead	13.4	
Aft Joint Ring	20.1	
Support Fittings	3.0	
Forward Bulkhead Cylindrical Seg.	9.2	
Spherical Seg.	6.7	
Elliptical Seg.	13.4	
Foam	59.8	
Forward Joint Ring	10.1	
	203.2	203.2
VACUUM SHELL		
Cylinder (54" dia × 92 " long)	80.6	
Spherical Bulkheads (2)	53.7	
Weld Lands, Rings, etc.	32.6	
Adhesive Weight	31.4	
	198.3	198.3
He PRESSURE BOTTLE (CRES)(34" dia)		394.0
SUPERINSULATION		88.6
	GRAND TOTAL	1058.4

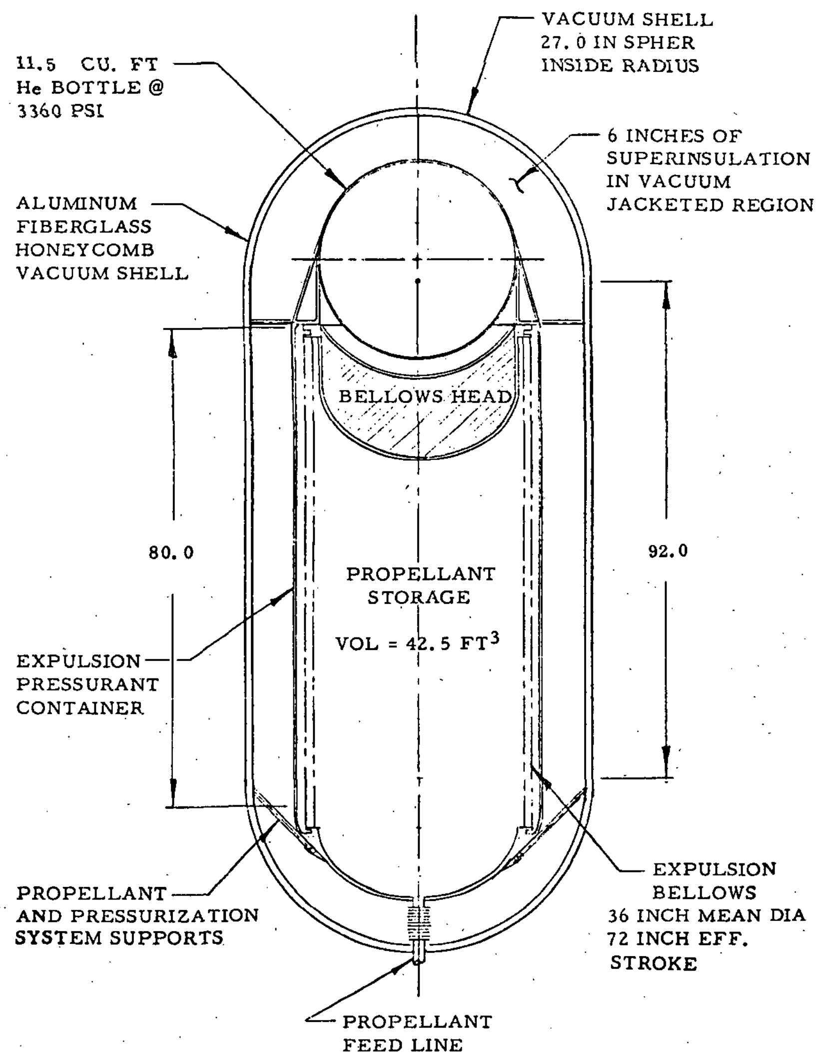


Figure 5-48. Bellows System, Configuration A

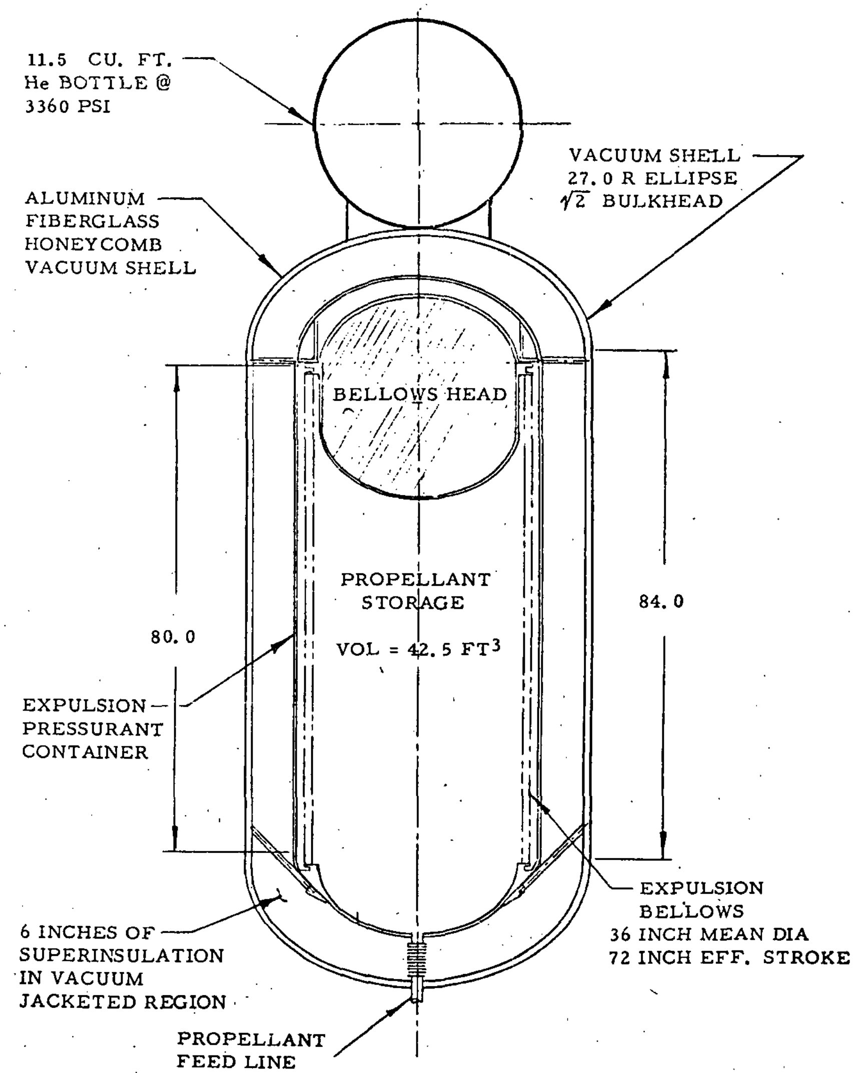


Figure 5-49. Bellows System, Configuration B

Table 5-11. Weights Statement for Configuration B, He Bottle Outside Vacuum Shell With Separate Pressure Shell

	WEIGHT, lbs	
PRESSURE SHELL		
Cylinder (40" dia × 80" long)	138.0	
Forward Bulkhead	20.4	
Bulkhead Transition	6.9	
Bellows Stops	6.0	
	171.3	171.3
EXPULSION CHAMBER		
Bellows (36" dia × 72" eff)	67.5	
Aft Bulkhead	13.4	
Aft Joint Ring	20.1	
Support Fittings	3.0	
Forward Bulkhead Cylindrical Seg.	9.2	
Elliptical Seg. (2)	26.8	
Foam	111.6	
Forward Joint Ring	10.1	
	261.7	261.7
VACUUM SHELL		
Cylinder (54" dia × 84" long)	82.6	
Elliptical Bulkheads (2)	43.6	
Weld Lands, Rings, etc.	32.6	
Adhesive Weight	29.0	
	187.8	187.8
He PRESSURE BOTTLE (Titanium 34" dia)		191.0
SUPERINSULATION		82.9
GRAND TOTAL		894.7

the location and material associated with the helium storage sphere. Also, elliptical rather than spherical bellows tank bulkheads are used in order to minimize waste volume at the tank ends.

The Table 5-11 data shows an overall weight reduction from that of Configuration A. It is noted that there is some increase in the expulsion chamber weight due to the separation of the bottle from the tank and the addition of a larger false bulkhead. However, this weight increase is more than offset by the reduction in helium storage weight.

The next step was to consider a design where the helium bottle would be located inside the overall tank envelope, but would not be part of the tank and could thus be made from titanium. One such system is shown in Figure 5-50, with a corresponding weight statement in Table 5-12. This system illustrates a somewhat unique approach, in that a separate inner shell surrounding the bellows is not employed. This allows the elimination of the weight associated with this pressure shell, but adds additional weight for bellows stops and guides. It is also noted that the vacuum jacket area must now be pressurized with helium which adds considerably to the pressurant storage weight. The overall result is that this system is heavier than previous ones. It would also be a disadvantage to pressurize the area containing superinsulation, since some damage to the insulation could result. Therefore, this type of system was eliminated from further consideration.

A relatively conventional configuration with the helium bottle separate but located inside the tank was then analyzed as Configuration D. A weight statement is presented in Table 5-13. It is noted that, due to anticipated packaging limitations, a cylindrical helium bottle was used. Due primarily to this factor, Configuration D still results in higher weight than Configuration B, which has an external spherical bottle. A check was then made on the potential of using a spherical bottle in a D type configuration, as presented in Figure 5-51. Weight data are presented in Table 5-14 showing a weight reduction. An examination of the data between Tables 5-14 and 5-11, however, shows that the main difference is now the added bulkhead weight for Configuration B. If it is assumed that the tank is prepressurized, such that waste tank space is not a factor, then Configuration B becomes the lower weight system. This is illustrated by the data presented in Table 5-15.

A summary of the results discussed indicates only a slight difference in weight between systems with and without the pressurant storage inside the propellant tank. Therefore, other considerations such as convenience of access, fabrication and cost will dictate any final selection.

The next step in the tradeoff analysis was to investigate the effects of different bellows L/D ratios. System designs were developed for L/D ratios of 1.5 and 1.0 in order to compare with the L/D = 2 systems previously analyzed. Configurations are presented in Figures 5-52 and 5-53, and weight statements in Tables 5-16 and 5-17. It is seen that the Figure 5-52 system, with an L/D of 1.5, represents the minimum weight. This is primarily due to the overall packaging efficiency of this design and the optimum relation of containing a cylindrical bellows in a tank containing ends which are not flat. That is, a reduction in height with an increase in diameter allows the pressure shell and vacuum jacket to approach a relatively structurally efficient spherical shape, but with an increase in waste volume at the end of the cylindrical bellows.

This (L/D = 1.5) system was thus chosen for further analysis and development of parametric data over the full range of volumes and expulsion pressures required. These parametric data are presented in the following paragraph.

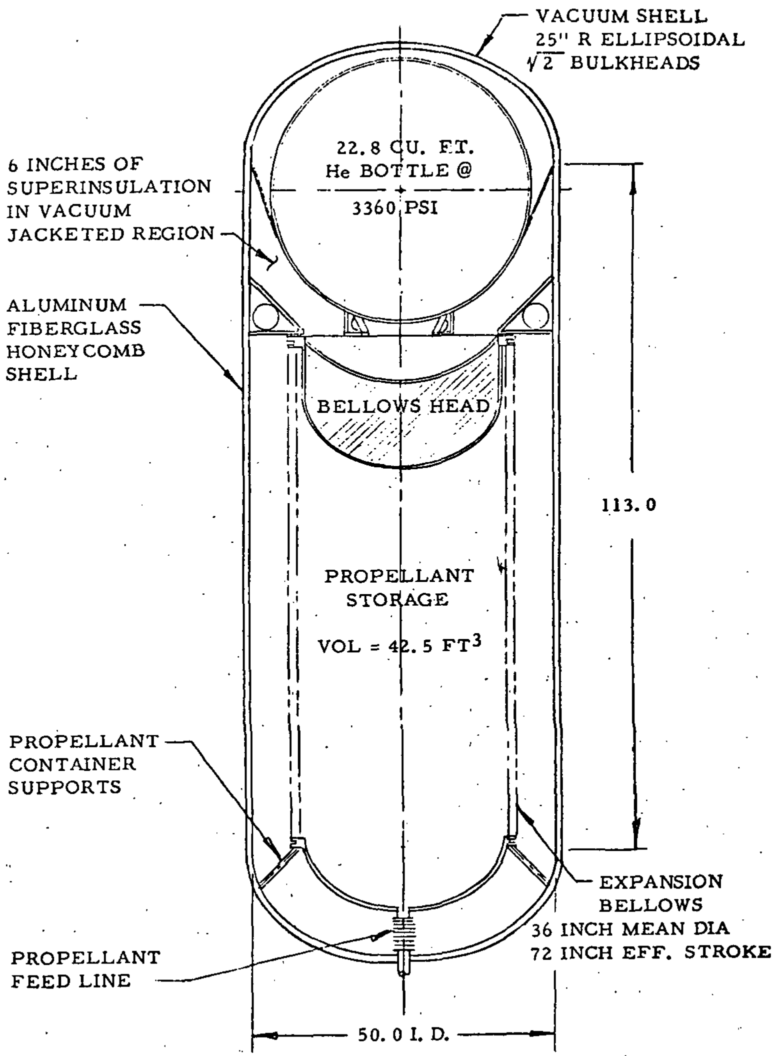


Figure 5-50. Bellows System, Configuration C

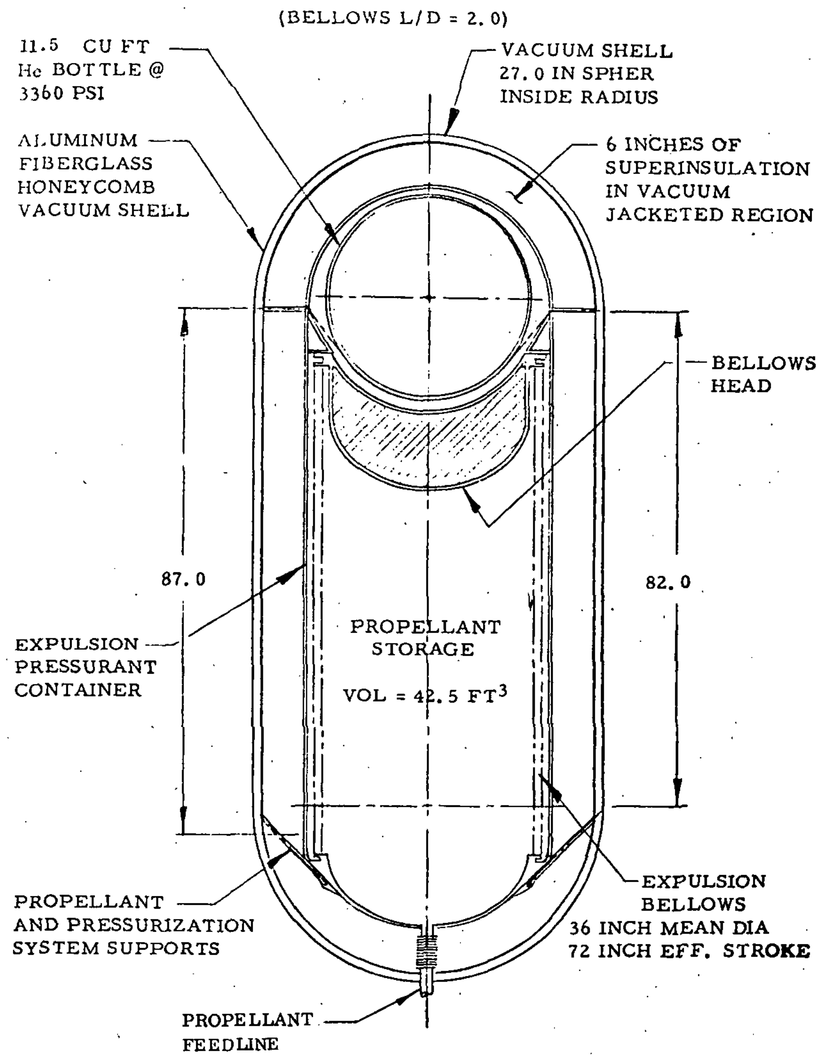


Figure 5-51. Bellows System, Configuration D

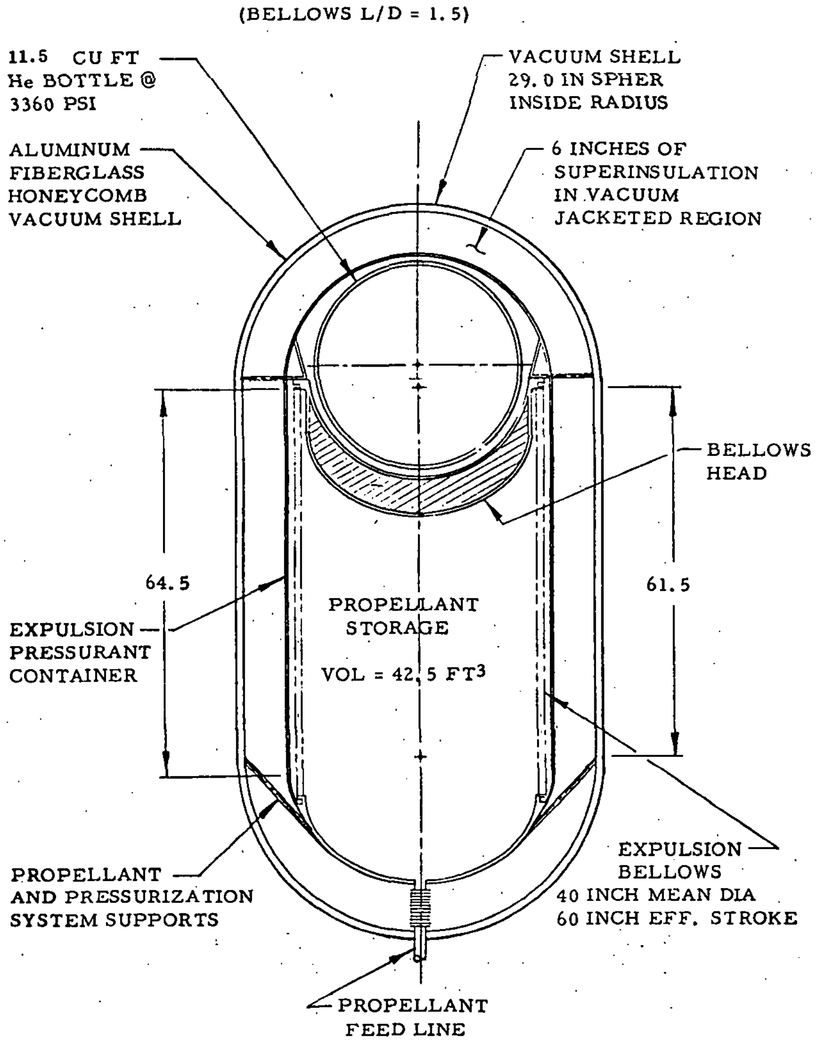


Figure 5-52. Bellows System, Configuration E

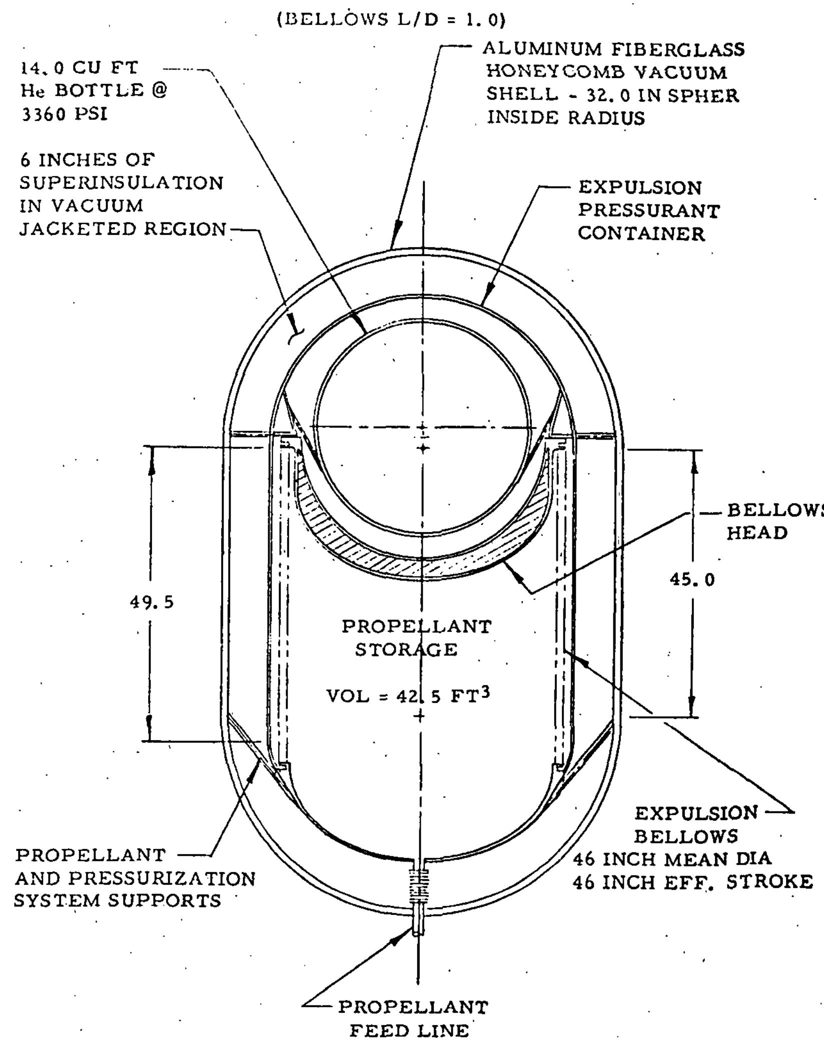


Figure 5-53. Bellows System, Configuration F

Table 5-12. Weights Statement for Configuration C, Combined Vacuum and Pressure Shell, Helium Bottle Inside

		WEIGHT, lbs
VACUUM PRESSURE SHELL		
Cylinder (50" dia × 113" long)	237.3	
Elliptical Bulkhead (2)	86.8	
Weld Lands, Rings, etc.	34.1	
Adhesive Weight	32.3	
	390.5	390.5
EXPULSION CHAMBER		203.2
(Same as Configuration A)		
ADDITIONAL EXPULSION CHAMBER GUIDES AND STOPS		
Bellows Stops	12.0	
Guides	30.0	
	42.0	42.0
He PRESSURE BOTTLE		386.0
(Titanium 43" dia)		
SUPERINSULATION		87.8
GRAND TOTAL		1109.5

Table 5-13. Weights Statement for Configuration D, Separated Helium Bottle Inside Vacuum Shell

		WEIGHT, lbs
PRESSURE SHELL		
Cylinder (40" dia × 87" long)	150.0	
Spherical Bulkhead	17.4	
Bulkhead Transition	6.9	
Bellows Stop	6.0	
	180.3	180.3
EXPULSION CHAMBER		
Bellows (36" dia × 72" eff)	67.5	
Aft Bulkhead	13.4	
Aft Joint Ring	20.1	
Support Fittings	3.0	
Forward Bulkhead Cylindrical Segment	18.4	
Elliptical Segment	26.8	
Forward Joint Ring	10.1	
	159.3	159.3
VACUUM SHELL		
Same as Configuration A		198.3
He PRESSURE BOTTLE		
(Titanium 28" dia w/26.5" cyl. section)		322.0
INSULATION		
Same as Configuration A		88.6
GRAND TOTAL		948.5

Table 5-14. Weights Summary for an Alternate Configuration D With a 34-In. He Bottle (Spherical) at 3360 psi

Pressure Shell	(Conf. D)	180.3 lbs
Expulsion Chamber	(Conf. A)	203.2
Vacuum Shell	(A or D)	198.3
He Pressure Bottle	(Conf. B)	191.0
Superinsulation	(Conf. A)	88.6
	Grand Total	861.4 lbs

Table 5-15. Weight Change With Helium Initially Filling Expulsion Pressurant Chamber (Foam Eliminated in Bellows Head)

Configuration B Weight	894.7 lbs
	-125.0
	769.7 lbs
Configuration D Weight	861.4
	-66.5
	794.9 lbs

Table 5-16. Weights Statement for Configuration E, Bellows L/D = 1.5

		Weights, lbs	
PRESSURE SHELL			
Cylinder and Trans.		132.0	
Spherical Bulkhead		20.6	
Bellows Stop		6.0	
		158.6	158.6
EXPULSION CHAMBER			
Bellows (40" dia × 60" eff)		61.4	
Aft Bulkhead		17.4	
Aft Joint Ring		23.0	
Support Fittings		3.0	
Forward Bulkhead	Cylindrical Seg.	8.5	
	Elliptical Seg.	17.4	
	Spherical Seg.	12.6	
	Foam	35.7	
Forward Joint Ring		11.2	
		190.2	190.2
VACUUM SHELL			
Cylinder		66.4	
Spherical Heads		63.5	
Adhesive		27.3	
Rings, Weld Lands, Brackets, etc.		34.2	
		191.4	191.4
He PRESSURE BOTTLE			191.0
SUPERINSULATION			83.6
		GRAND TOTAL	814.8

Table 5-17. Weight Statement for Configuration F, Bellows L/D = 1.0

	Weight, lbs	
PRESSURE SHELL		
Cylinder and Trans.	135.0	
Spherical Bulkhead	30.0	
Bellows Stop	6.0	
	171.0	171.0
EXPULSION CHAMBER		
Bellows (46" dia × 46" eff)	54.7	
Aft Bulkhead	26.8	
Aft Joint Ring	28.3	
Support Fittings	3.0	
Forward Bulkhead Cylindrical Seg.	10.5	
Elliptical Seg.	26.8	
Spherical Seg.	20.6	
Foam	32.8	
Forward Joint Ring	12.9	
	216.4	216.4
VACUUM SHELL		
Cylinder	54.8	
Spherical Heads	79.0	
Adhesive	29.4	
Rings, Weld Lands, Brackets, etc	36.6	
	199.8	199.8
He PRESSURE BOTTLE		266.5
SUPERINSULATION		83.8
GRAND TOTAL		937.5

5.5.2 DETAIL ANALYSIS AND DEVELOPMENT OF PARAMETRIC DATA. Based on the system shown in Figure 5-52, parametric weight data were generated over the range of propellant volumes from 4.7 ft^3 (25 in. dia. sphere) to 1020 ft^3 (150 in. dia. sphere) and tank pressures from 50 to 500 psi. The analyses presented in the previous paragraph showed only a small difference in system weight between locating the helium bottle inside or outside of the tank. Data were thus generated in a form which is essentially independent of the pressurant location. Also, weights for an intermediate bulkhead are presented separately so that its use can be independently assessed for any particular design situation. It is noted that the basic weight generation methods are the same as discussed in the previous paragraph. Bellows design and sizing are based on the use of existing fabrication procedures. Even though the actual tooling to fabricate the larger sizes did not presently exist, it was assumed that such tooling could be developed and would show up as a cost item when comparing the bellows with other subcritical transfer concepts.

Weights of the pressure shell and expulsion bellows as an integral system are presented in Figure 5-54. Corresponding vacuum shell weights are presented in Appendix A. These are the two main weight components of the overall system, excluding pressurization.

It is noted that the jacket weight is presented as a function of its diameter. For a given propellant volume this diameter will be a function of the inner tank radius and the insulation thickness. A curve of inner tank radius versus propellant volume is presented in Figure 5-55.

The weight of a second bulkhead and foam filler is presented in Figure 5-56. Also, for reference the weight of the individual bellows assembly is presented in Figure 5-57.

Tank surface areas for determining insulation requirements are given in Figure 5-58 and the weights of titanium helium bottles are presented in Appendix A. The total volumes which are available for pressurant storage within the tank are presented in Figure 5-59. The total pressure shell volume is the sum of the propellant and pressurant storage volumes presented in Figure 5-59. The double bulkhead volume is found from the difference between the two curves presented in Figure 5-59.

Following is an example of the use of the data presented in Figures 5-54 through 5-59. It is desired to determine the total weight of a bellows system designed to operate at 200 psia and contain 42.5 ft^3 of fluid. The insulation is to be six inches thick, and to allow for weld lands and mounting bosses, the vacuum shell is to be 14 inches larger in diameter than the inner shell. A double bulkhead is to be employed. The resulting weight statement is presented in Table 5-18. Comparing these data with that presented in Table 5-16 shows 605 lb versus 624 lb. Most of this difference is due to a refinement in the assessment of insulation volume for the present calculations. Some difference is also found between the vacuum shell weights, primarily due to a difference in the space allowed between the jacket and the inner shell.

To determine the helium bottle weight, an iteration is required since the helium requirement depends on the volume to be pressurized and vice versa. For an 11.5 ft^3 helium bottle, assumed to be stored in the tank; from Figure 5-59 the volume remaining to be pressurized is equal to 24.6 ft^3 - 11.5 ft^3 + 42.5 ft^3 = 55.6 ft^3. Ratioing the helium storage volume from that required to pressurize 42.5 ft^3, results in V_{He} = 8.8 (55.6/42.5) = 11.5 ft^3, which checks with the initial assumption. Then, from Appendix A for a 3,000 psia storage case the bottle weight is 198 lb. The total dry system weight is thus 803 lbs. This same analysis can then be accomplished for other propellant volumes and pressurant storage conditions.

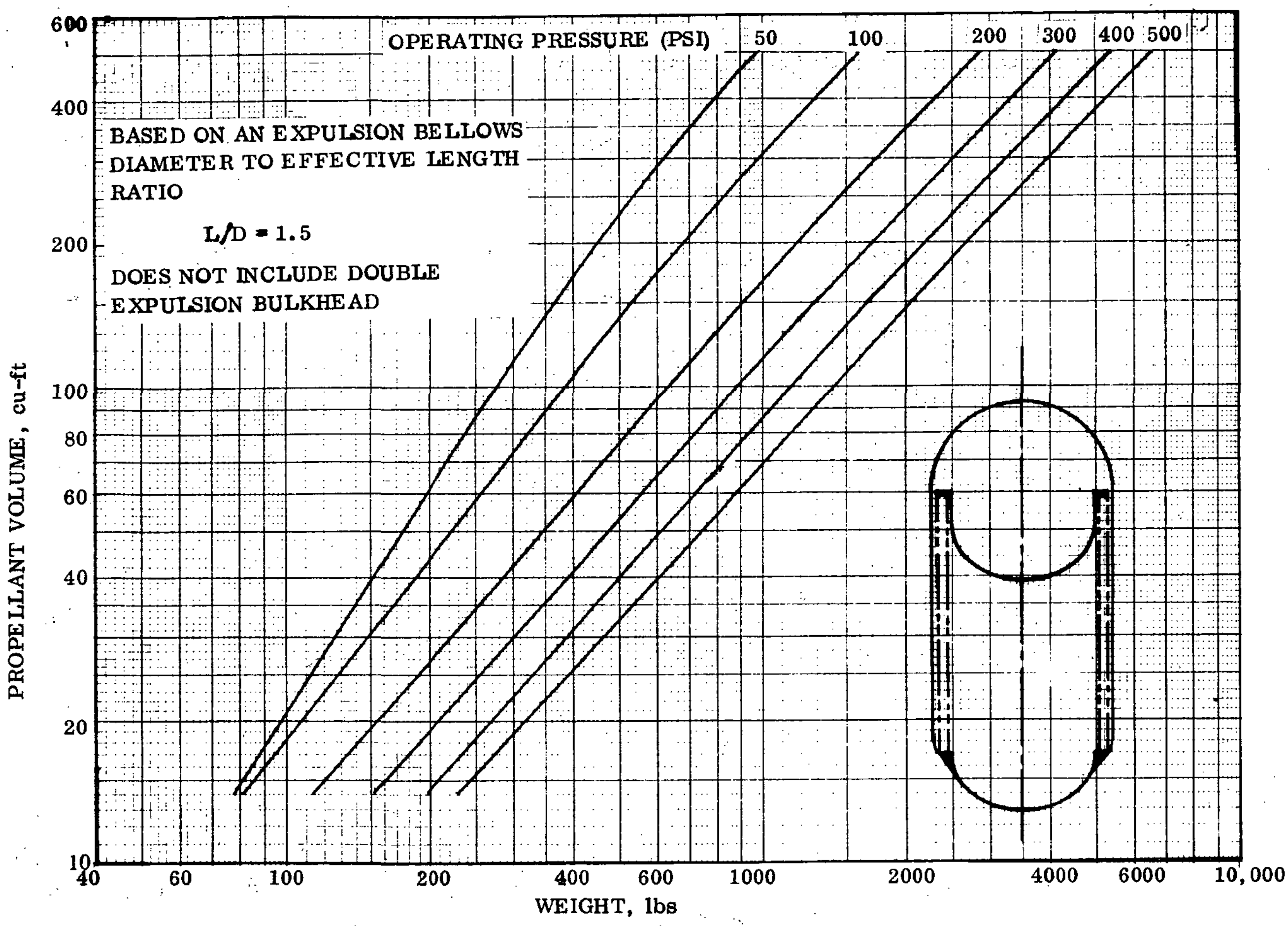


Figure 5-54. Pressure Shell and Expulsion Bellows Weight Data

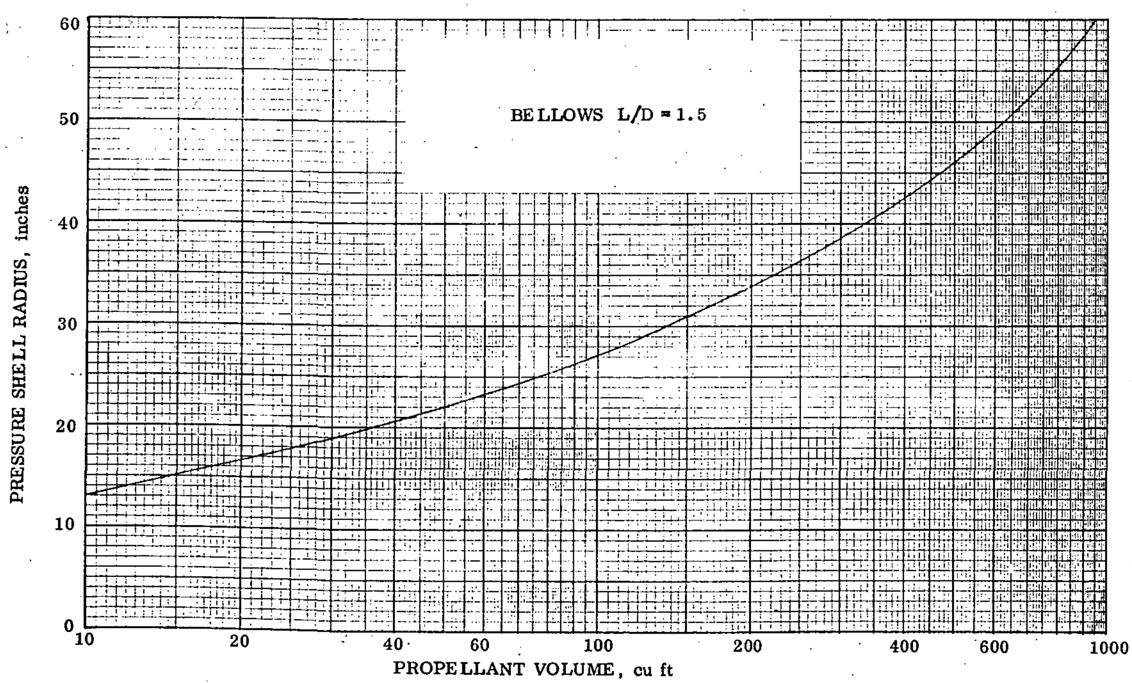


Figure 5-55. Bellows Expulsion System Pressure Shell Radius Vs. Propellant Volume

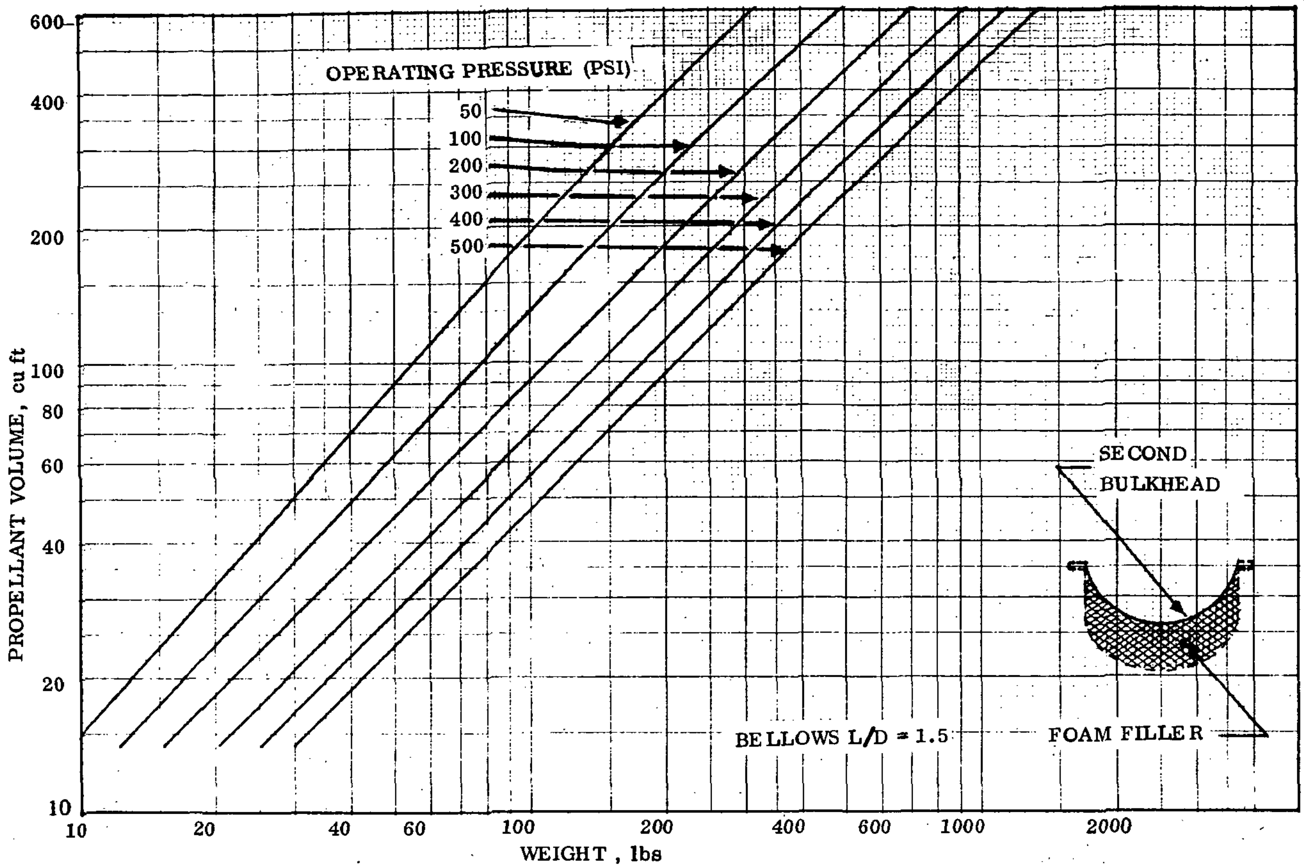


Figure 5-56. Double Expulsion Bulkhead Weight Data

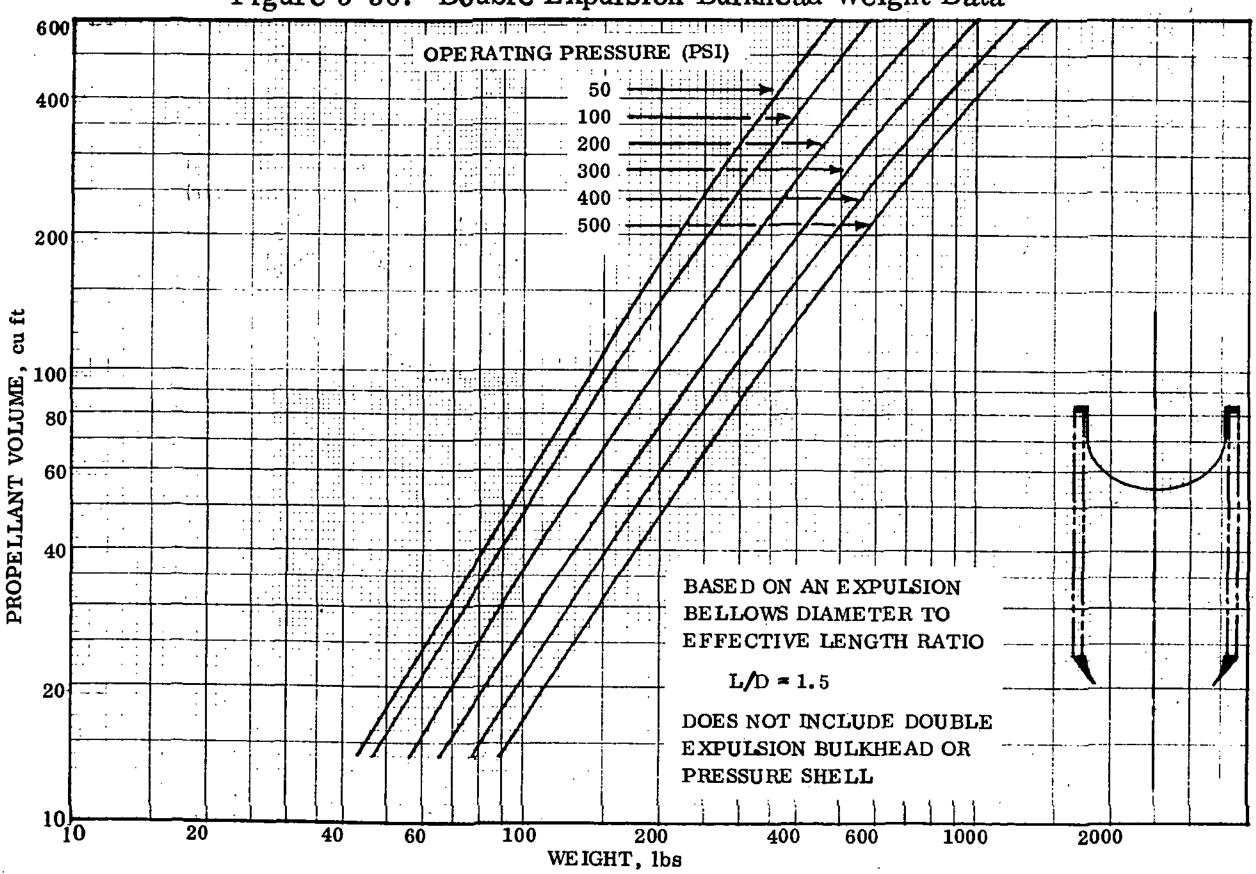


Figure 5-57. Bellows and Expulsion Head Weight Data

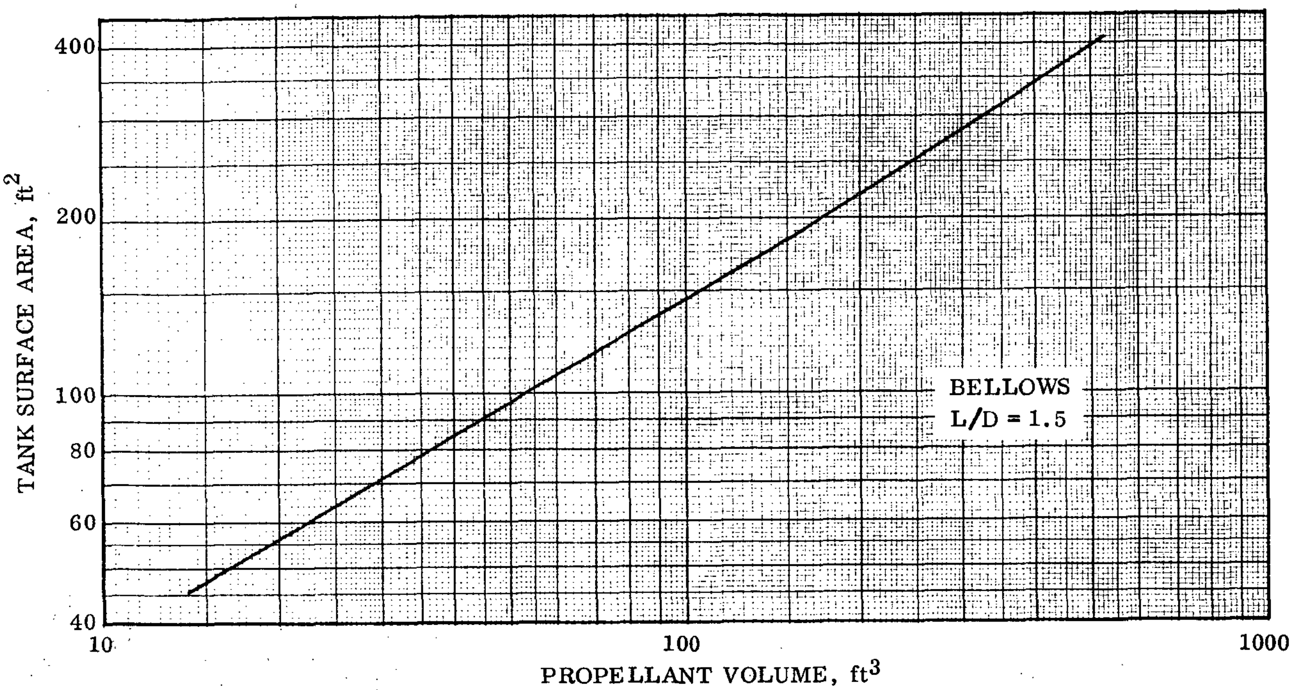


Figure 5-58. Inner Tank Surface Area Versus Contained Propellant Volume

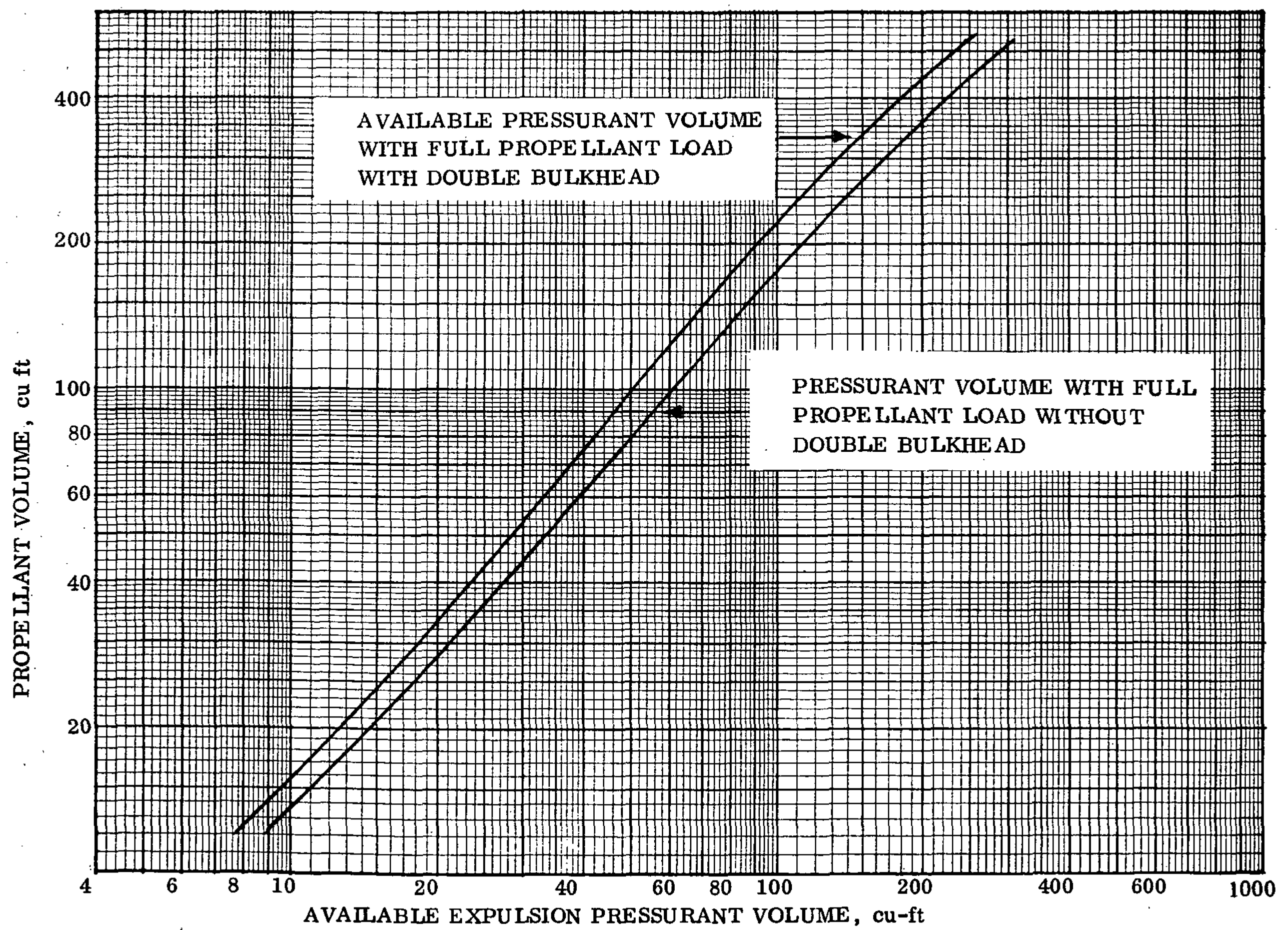


Figure 5-59. Bellows Expulsion System Pressurant Storage Volumes Available

Table 5-18. Summary Weight Data for Example Case

Item	Size or Pertinent Dimension(s)	Unit Weight, lb	Total Weight, lb	Source of Data
Pressure Shell and Expul. Bellows	42.5 ft^3 (42" dia.)	-	300	Fig. 5-54
Vacuum Jacket	56" dia	-	187	Fig. A-5
Double Bulkhead	—	-	47	Fig. 5-56
Insulation	88 ft^2	1.3 lb/ft^3	71	Fig. 5-58
		TOTAL	605 lb	

5.6 METALLIC DIAPHRAGM SYSTEM

The basic metallic diaphragm system being considered is essentially as presented in Figure 3-6. The data contained in this section is the result of performing a parametric weight analysis of this system. The basic weight data for the diaphragm were obtained from point design data on units built by Arde, as presented in References 3-26 through 3-28. The total diaphragm weight consists of that for the thin membrane plus that for the reinforcing wires. From data available, membrane thicknesses and wire diameters were estimated as a function of tank diameter. Examination of the existing data indicated that the spacing between wires, for stability purposes, was at constant angles around the tank. From the above information total diaphragm weights were obtained as a function of tank diameter and are presented in Figure 5-60 for two cases. One assumes the use of solid wires and the other uses hollow wires with the same stiffness.

It is noted that these diaphragms are CRES and therefore the inner tank shell should also be CRES. Weight data for such a vessel was developed and is presented in Appendix A. This tank configuration is essentially the same as that to be used for other systems requiring a spherical tank. The weld land at the tank circumference, as used for conventional tank construction, will be sufficient for containing the diaphragm flange, and thus an additional weight penalty to incorporate the diaphragm is not incurred. Vacuum jacket design would not be significantly affected and thus weight data for a spherical jacket presented in Appendix A is applicable.

5.7 PADDLE VORTEX SYSTEM

This section presents the results of detailed design and analysis accomplished on the basic paddle vortex system described in Section 3.3 (Figure 3-7). The paddle vortex system operates by creating a centrifugal acceleration on the supply liquid to maintain it at the tank outlet for transfer.

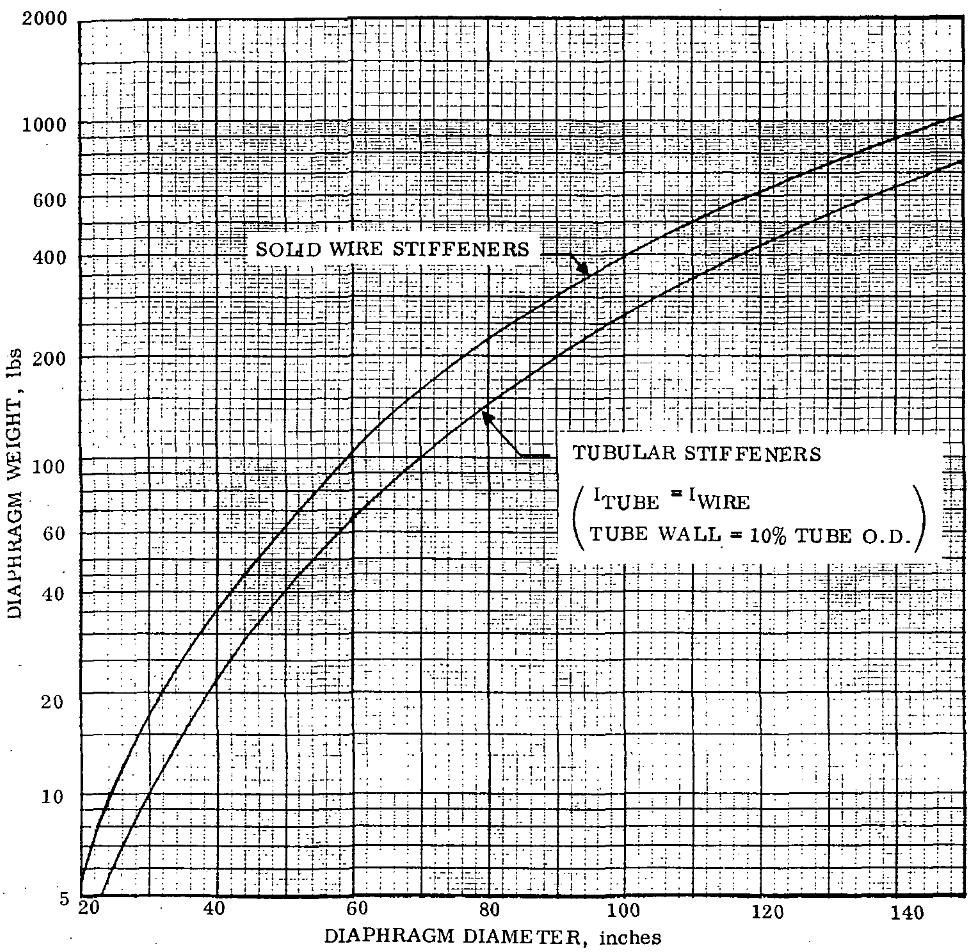


Figure 5-60. Expulsion Diaphragm Weight Based on an Extrapolation of Data for CRES Membranes

A significant characteristic of this system is that some powered means is required to drive the paddle. For the present case an electric motor operating through a hermetically sealed flex spline was chosen to have the best potential. Such a system provides a positive seal between the drive motor and tank fluid and thus allows the motor to be external to the tank and separated from the tank fluid. Also, the flex spline system allows a large reduction in speed between the motor and paddle, as required by the present system since paddle rotational rates are desired to be very low.

The resulting system configuration for a 52-inch diameter tank is presented in Figures 5-61 and 5-62.

One requirement was to overcome surface tension forces which when acting in conjunction with imposed accelerations away from the outlet could result in unwanted vapor expulsion. The other criteria or requirement was that liquid be pumped away from the inner radius of the paddle.

The equations below for determining rotation rates required to overcome the surface forces are based on data obtained from Reference 3-32.

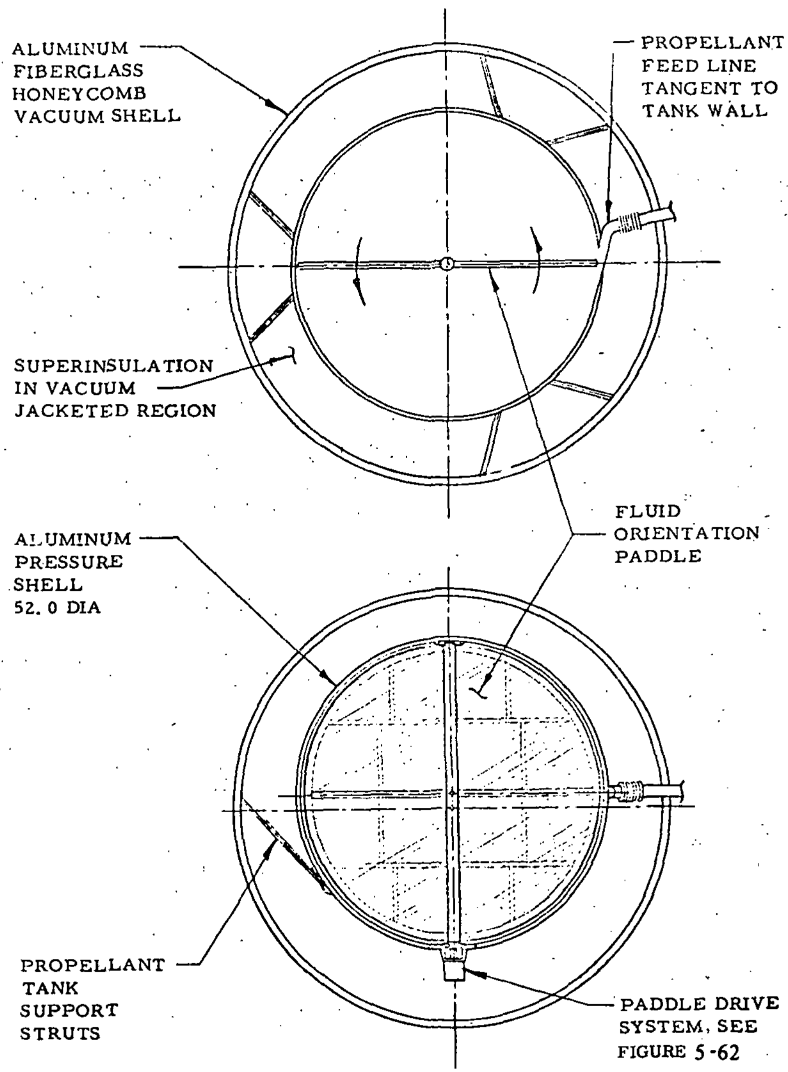


Figure 5-61. Paddle Vortex System

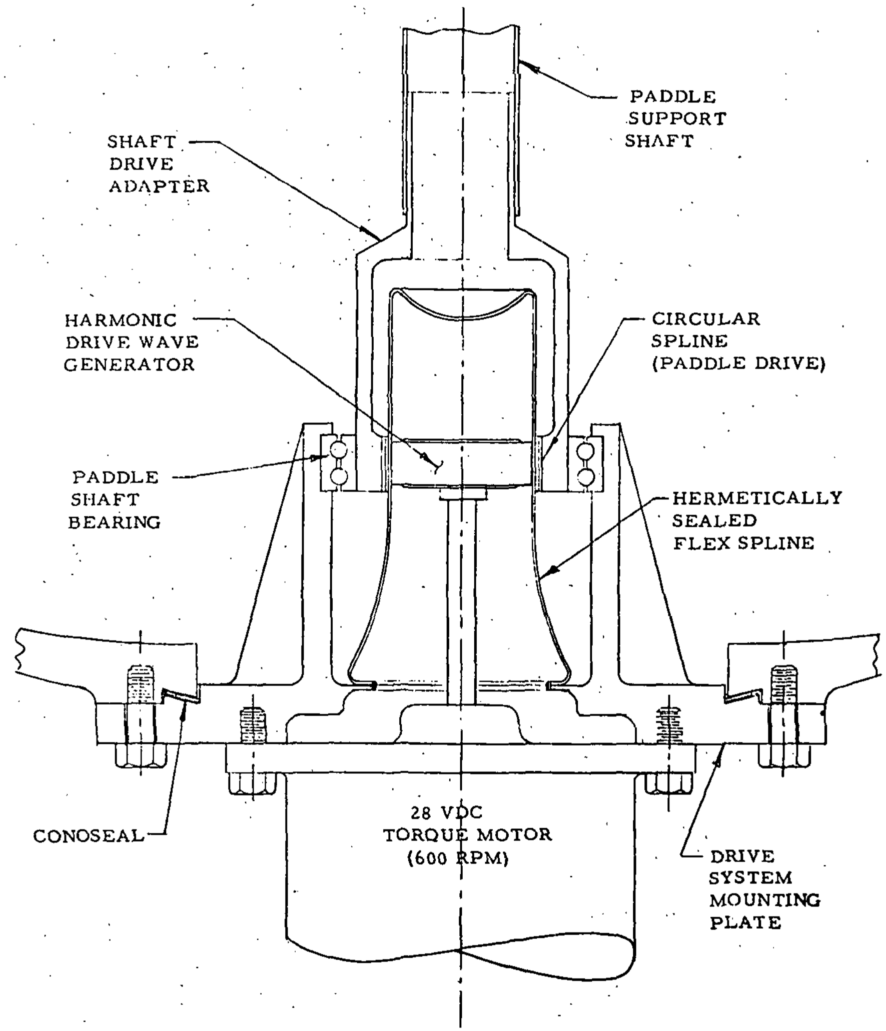


Figure 5-62. Paddle Drive System

$$\Omega_c^2 \equiv \frac{\omega^2 \rho_L R_t^3}{\sigma} \tag{5-27}$$

The above equation is the defining equation for a rotational Weber No. describing the position of the liquid in a rotating body of fluid. For values of Ω_c^2 greater than 4 + Bo/0.2 the liquid would be forced to the outer walls of the tank. Substituting this relation and the applicable definition of the Bond No. ($Bo = \rho_L a R_t^2/\sigma$) into Equation 5-27 results in the following solution for the required rotation rate to overcome the surface forces.

$$\omega = \sqrt{\frac{4\sigma}{\rho_L R_t^3} + \frac{5a}{R_t}} \tag{5-28}$$

Solutions to this equation for an acceleration of 10^{-4} g's for H_2, O_2 and N_2 over a range of tank radii are presented in Figure 5-63. Properties data for fluids saturated

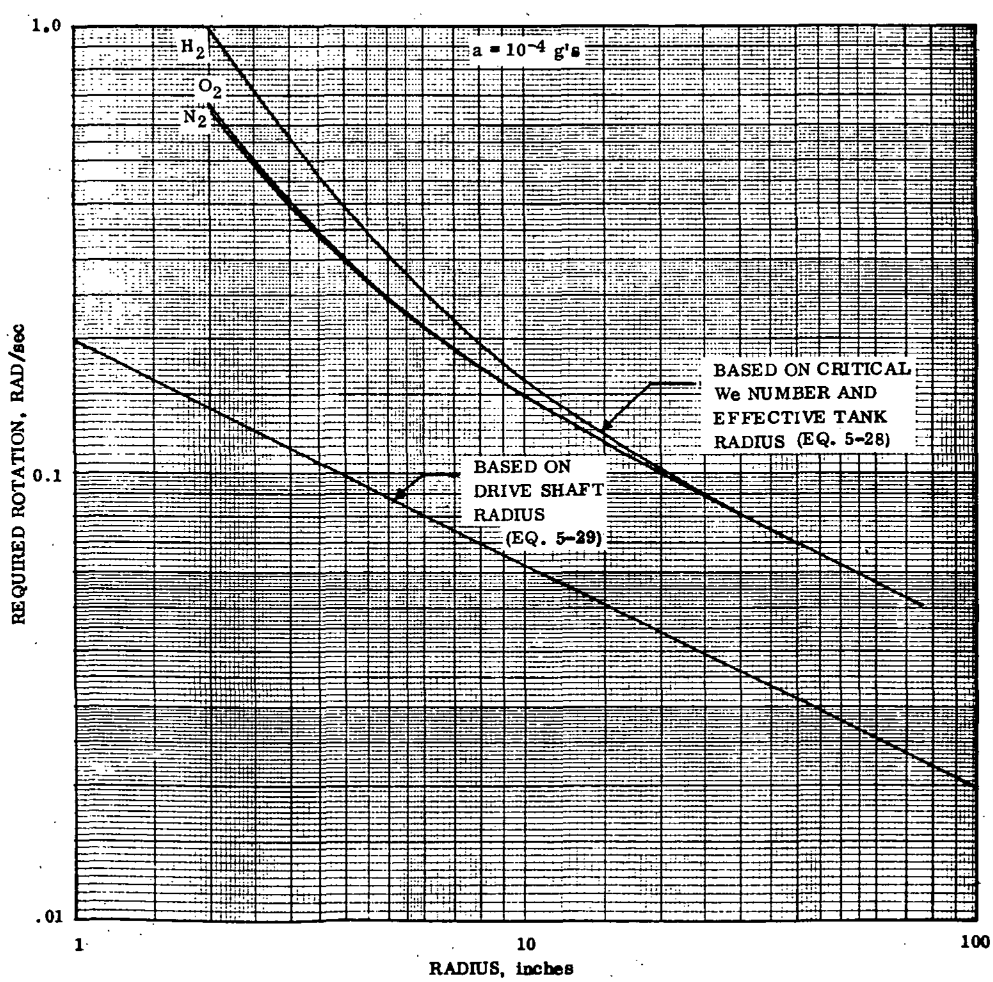


Figure 5-63. Required Rotation Rates for Liquid Positioning Using a Paddle

at 20 psia were used. It is noted that the radius has a significance for a spherical tank only as an effective radius since Equation 5-28 is basically for a cylindrical tank per the data of Reference 3-32. Data specifically for spherical tanks was not found in the literature.

To determine rotation rates required for pumping liquid away from the inner radius, the following equation was used.

$$\omega = \sqrt{a/R_i} \tag{5-29}$$

In this case the inner radius is taken to be that of the drive shaft and solutions to Equation 5-29 for $a = 10^{-4}$ g's are presented in Figure 5-63. For the present case the initial design analysis showed that a shaft diameter of 2 inches would be reasonable for use in the 52-inch diameter tank. On this basis the required rotation rate would be 0.2 rad/sec or approximately 2 rpm.

Assuming an effective tank radius of 13 inches (average value for 52-in. dia tank) the corresponding rotation rate based on the critical Weber No. criteria for H_2 would be 0.13 rad/sec. This indicates a less stringent requirement for overcoming surface tension than for pumping from the inner radius.

For design purposes it is reasonable to assume the shaft diameter to be proportional to the tank diameter and thus the more stringent requirement would always come from Equation 5-29. Taking 2 rpm as the minimum requirement for a 52-inch diameter tank and multiplying by a safety factor of three results in a design requirement of 6 rpm for this size tank. This fairly high safety factor is required due to the high level of uncertainty involved with the present state-of-the-art and associated calculation methods for this type of system. Rotation rates at other tank diameters are based on the use of Equation 5-29 and the assumption that shaft diameter is proportional to tank diameter. Thus to overcome an acceleration of 10^{-4} g's

$$\omega \text{ rpm} = 14.4/(D_t \text{ in.})^{1/2} \tag{5-30}$$

The next step in the analysis was to determine power requirements to drive the paddle as a function of tank size. The required power is taken to be equal to the drag force times the moment arm (tank radius) times the angular rotation rate; i.e.

$$\dot{P} = F_D R_t \omega \tag{5-31}$$

where

$$F_D = C_D \rho_L A_p \frac{(V_e)_p^2}{2g_c} \tag{5-32}$$

and

$$(V_e)_p = \omega R_t \tag{5-33}$$

For a 52 inch diameter tank operating with LO_2 where $C_D = 1.0$, $\rho_L = 70\ lb/ft^3$, $A_p = (\pi D_t^2)/4 = 14.8\ ft^2$, and $(V_e)_p = 1.36$ ft/sec, then $F_D = 29.6$ lb and $\dot{P} = 0.0735$ HP. Taking the motor-drive efficiency to be 50 percent then the input to the motor would be 0.147 HP. In order to determine motor input power as a function of tank size, equations 5-30 through 5-33 were used in conjunction with the following assumptions.

$C_D = 1.0$ = constant, motor efficiency = 50% = constant

$A_p = \pi D_t^2/4$

The resulting equation is

$$\dot{P}_{motor\ input},\ HP = 2.08 \times 10^{-9}\ (\rho_L,\ lb/ft^3)\ (D_t,\ in.)^{3.5} \qquad (5\text{-}34)$$

It is noted that in developing the above equation the start-up inertia was assumed to be negligible for the combination of paddle and fluid. Subsequent analysis indicated this to be true for O_2 but not for H_2. To account for start-up or paddle inertia in the case of hydrogen the power requirements were increased by a factor of two. On this basis parametric data are shown plotted in Figure 5-64 for both O_2 and H_2 cases. The following equations were used.

$$HP = 1.46 \times 10^{-7}\ (D_t,\ in.)^{3.5} \quad \text{Oxygen} \qquad (5\text{-}35)$$

$$HP = 1.77 \times 10^{-8}\ (D_t,\ in.)^{3.5} \quad \text{Hydrogen} \qquad (5\text{-}36)$$

$$HP = 1.032 \times 10^{-7}\ (D_t,\ in.)^{3.5} \quad \text{Nitrogen} \qquad (5\text{-}37)$$

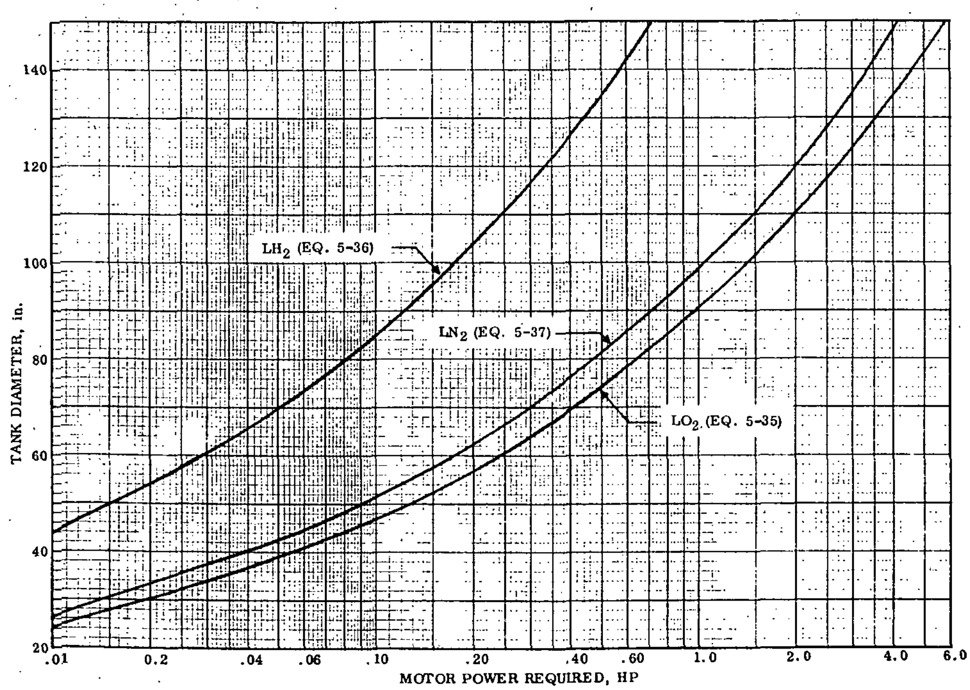


Figure 5-64. Paddle Drive Horsepower Requirements

It is noted that data are not given for motor powers below .01 horsepower since this was considered a minimum practical motor size to obtain a high reliability for the present application.

Based on a perusal of available vendor motor and drive data, the weight curves presented in Figure 5-65 were developed. Motor weight data are presented as a function of horsepower for 600 rpm DC motors and drive weights are presented for both 100:1 and 200:1 speed reduction systems. Totals are also given. Data for an 11,000 rpm motor are presented for reference.

Also for reference, the weights of a power supply system to provide operation over various time periods are presented in Figure 5-66. This data is based on the use of fuel cells with a weight assessment, as presented in Section 3.3, of 94 lb/kw plus 2.9 lb/kw-hr.

Weights presented in Figure 5-66 for 24 hours would represent a maximum. Actual transfer times would depend on receiver and line chilldown considerations and supply tank residuals as affected by outflow rate.

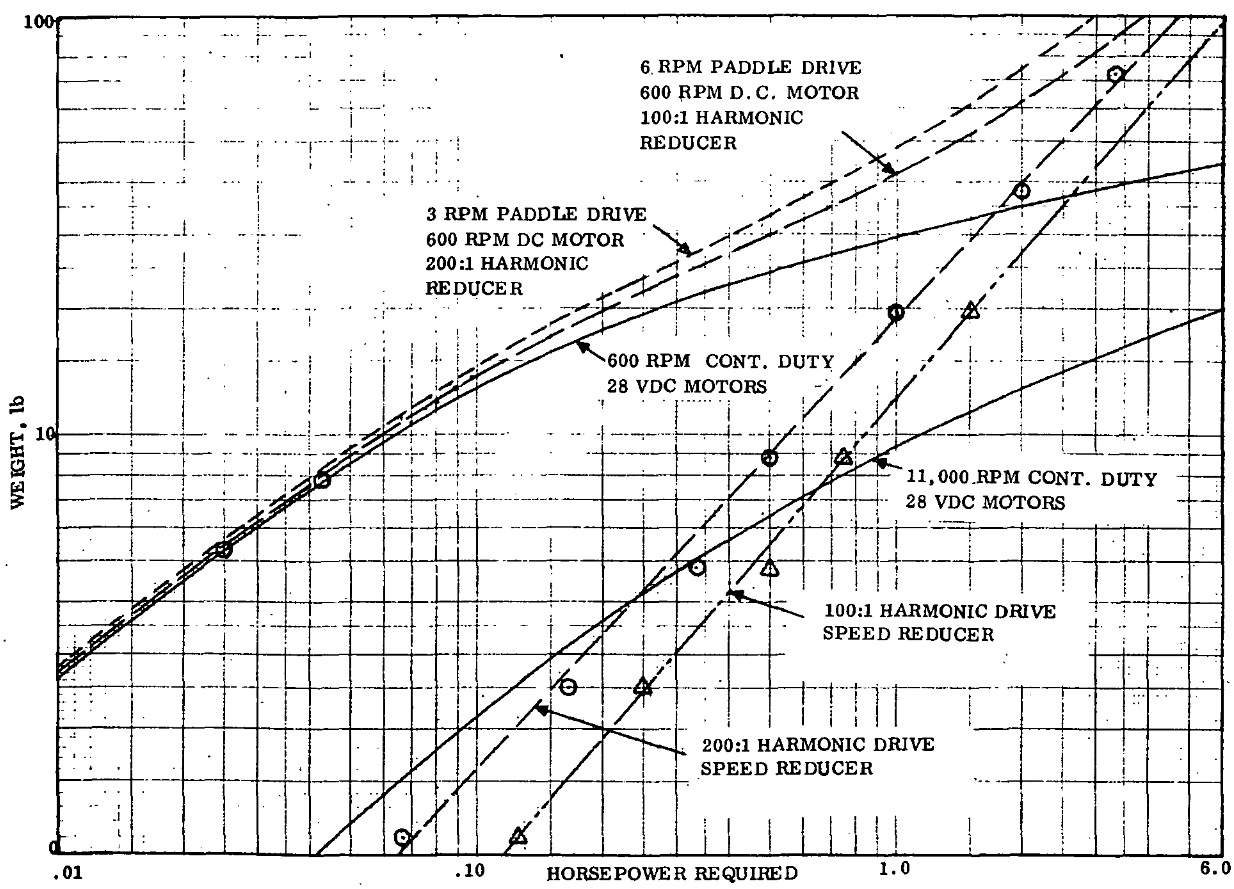


Figure 5-65. Paddle Drive System Weight

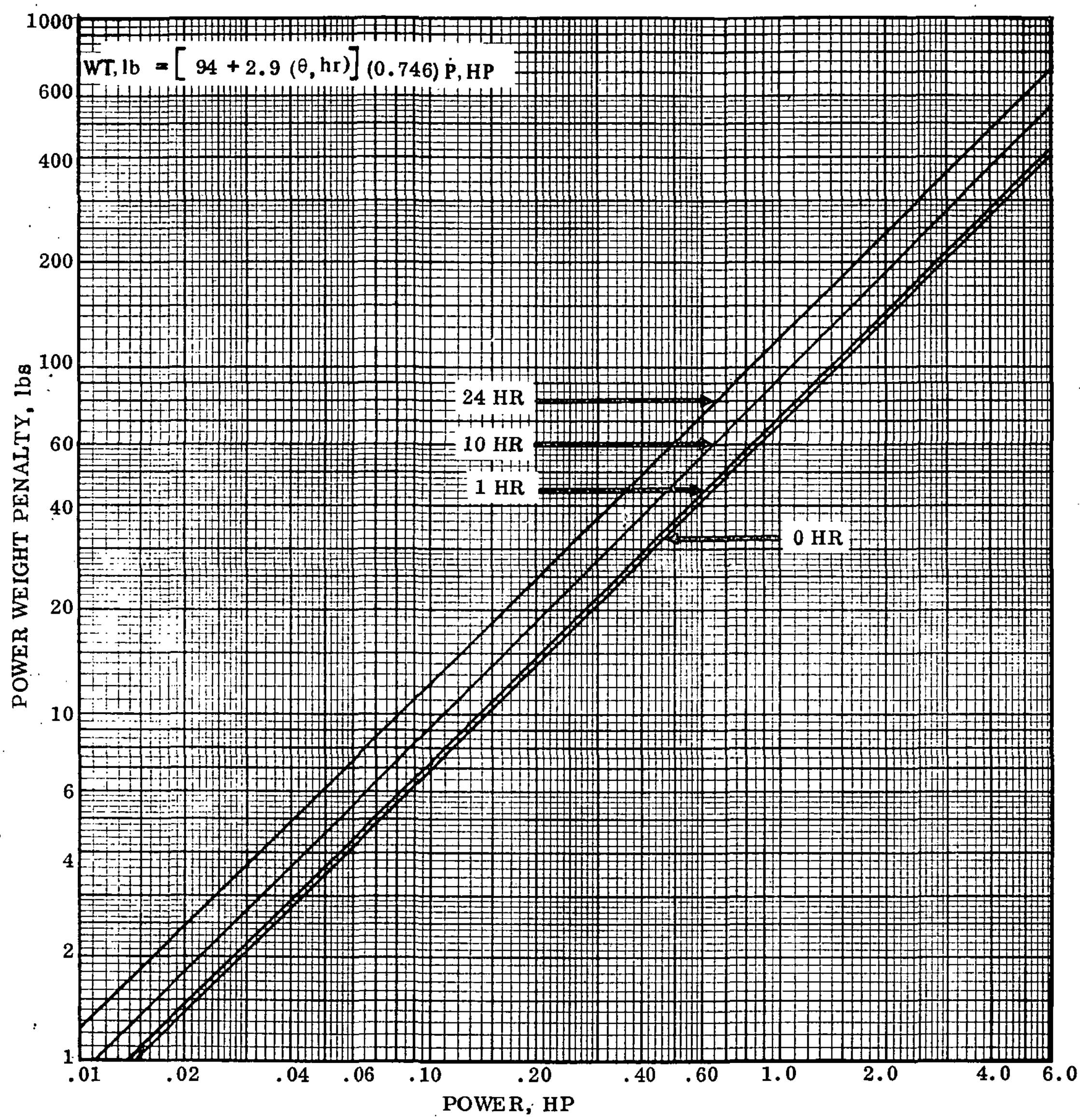


Figure 5-66. Power Weight Penalty Versus Horsepower for a Paddle System

Weights of the paddle itself, including the drive shaft, are presented in Figure 5-67.

One of the main problems anticipated for this system was in the area of fluid residuals, since the liquid/vapor interface is in the form of an annulus at the tank side wall. Thus at vapor pull through there could be a significant volume of liquid remaining in the tank. This is illustrated in Figure 5-68. Also, the liquid existing between the paddle and tank wall may have to be included as residual since a direct centrifugal force would not exist in this area. Analysis was thus performed to obtain reasonable estimates of the expected residuals.

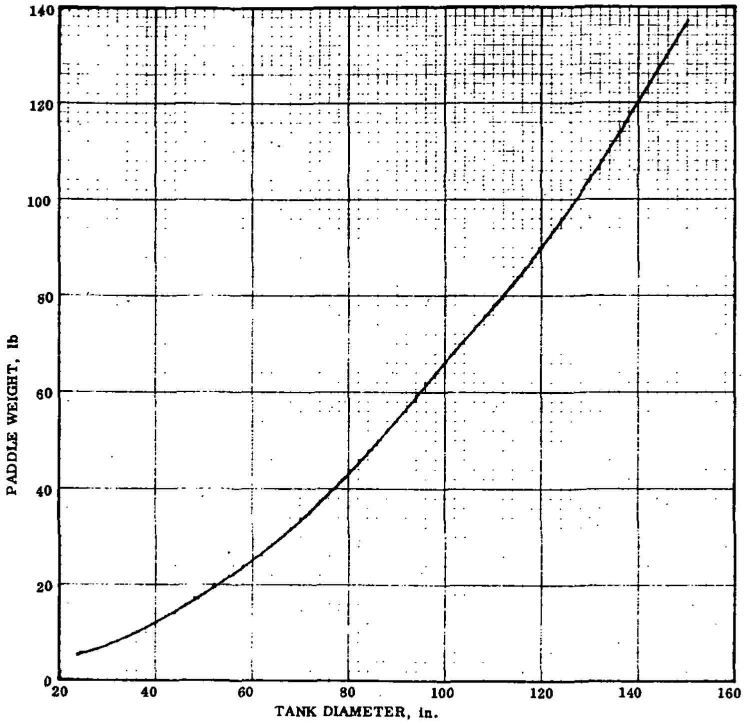


Figure 5-67. Vortex Paddle Weight Versus Tank Diameter

Calculations were initially made using standard pull-through equations developed in Reference 5-10 and presented below.

$$\frac{\dot{V}_o^2}{a_p h_c^5} = 6.5 \text{ for } h_c \gg R_o \qquad (5\text{-}38)$$

$$\frac{\dot{V}_o^2}{R_o^2 a_p h_c^3} = 11.8 \text{ for } h_c \ll R_o \qquad (5\text{-}39)$$

where a = acceleration acting on the fluid, taken to be $R_t \omega^2$.

From geometry considerations, the volume of liquid remaining in the tank is

$$V_L = \frac{4}{3} \pi \left(h_c \, (2R_t - h_c) \right)^{3/2} \qquad (5\text{-}40)$$

This is compared to conventional draining from the bottom of a spherical tank where;

$$V_L = \frac{1}{3} \pi h_c^2 \left(\frac{3}{2} D_t - h_c \right) \qquad (5\text{-}41)$$

Values for pull through height (h_c), as determined from Equation 5-38, are presented in Figure 5-69 as a function of transfer time for a 52 inch diameter tank. The transfer time (θ_T) is based on a simple case of constant outflow where $\theta_T = V_t/\dot{V}_o$.

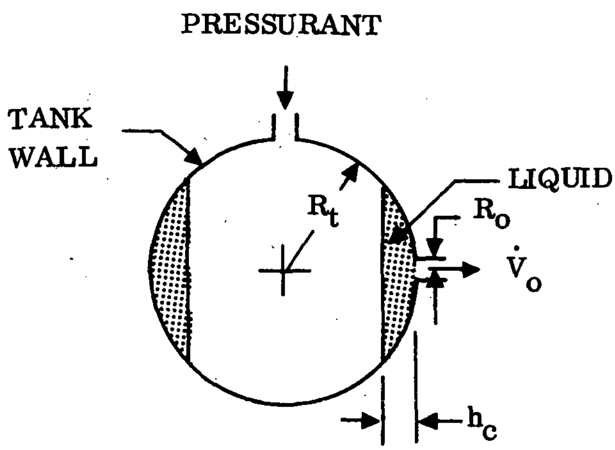


Figure 5-68. Expected Liquid Configuration for Paddle Vortex System

The rotation rate was taken as 6 rpm and the acceleration for expulsion was determined from the relation $a_p = R_t \omega^2$, to be 0.854 ft/sec^2. An examination of Equation 5-30 in conjunction with the above relation shows that the paddle acceleration would be constant for constant disturbing acceleration for the various tank sizes being considered. It is also seen from the Figure 5-69 data that values for h_c are likely to be greater than the exit line radius and thus Equation 5-38 should be applicable.

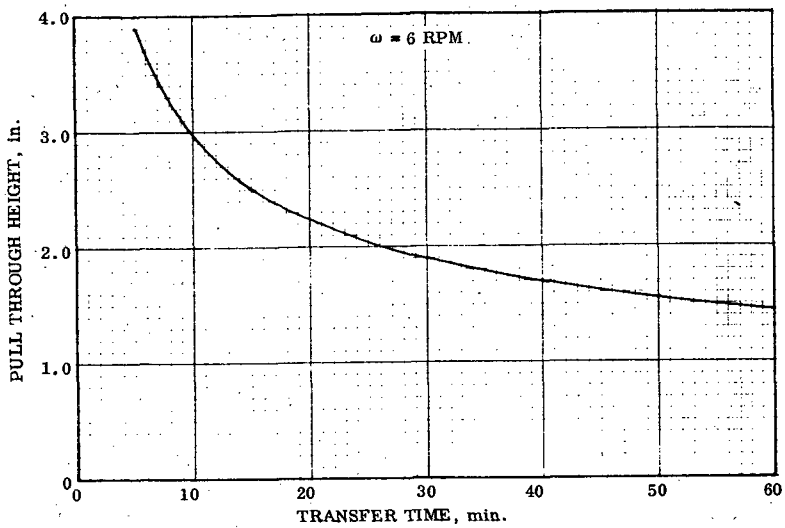


Figure 5-69. Paddle Vortex System Pull Through Liquid Height (62-In. Dia Tank)

Residuals in terms of percent of the total tank volume, as determined from Equations 5-38, 5-40 and 5-41 are presented in Figure 5-70 for various tank sizes. The data shows that the residuals can be quite high for this system. As an example, for a 52-inch diameter tank with a transfer time of 10 minutes, the residuals would be 10 percent. By increasing the transfer time to one hour the residuals were reduced to 3.5 percent which is still high for this size tank.

The effect of changing the paddle rotation rate and thus the centrifugal acceleration is illustrated in Figure 5-71. Increasing the rotation rate to 12 rpm resulted in an estimated residual reduction to 2.5 percent for the one hour transfer. This, however, would result in a significant increase in the system power requirements.

Further analyses were performed to determine the effects of using a sump at the tank outlet to reduce residuals. This approach resulted in a potential reduction of residuals for the 6 rpm 52-inch tank case to nominally 3 percent for transfer times on the order of 10 minutes. Data were also generated as a function of tank size which indicates that this percentage will increase slightly with tank size. The primary problem with sump systems would be in the complication of the basic tank design. Further analysis and testing would be required to accurately predict residuals for the sump systems.

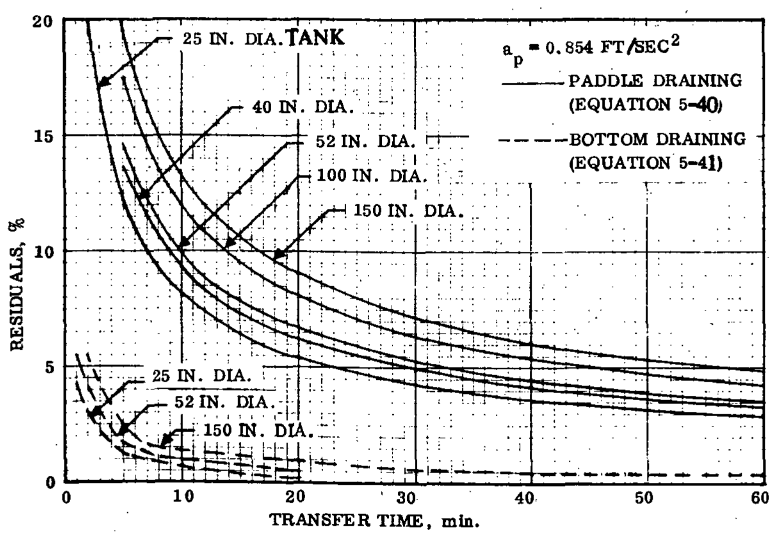


Figure 5-70. Paddle System Residuals

Another potential area for the location of liquid residuals was assumed to be between the paddle periphery and the wall. The minimum clearance between the paddle and the wall was estimated to be the same as that used for the surface tension screen system. On this basis the projected minimum paddle residuals would be comparable to those of a single or full liner surface tension system as presented in Figure 5-36.

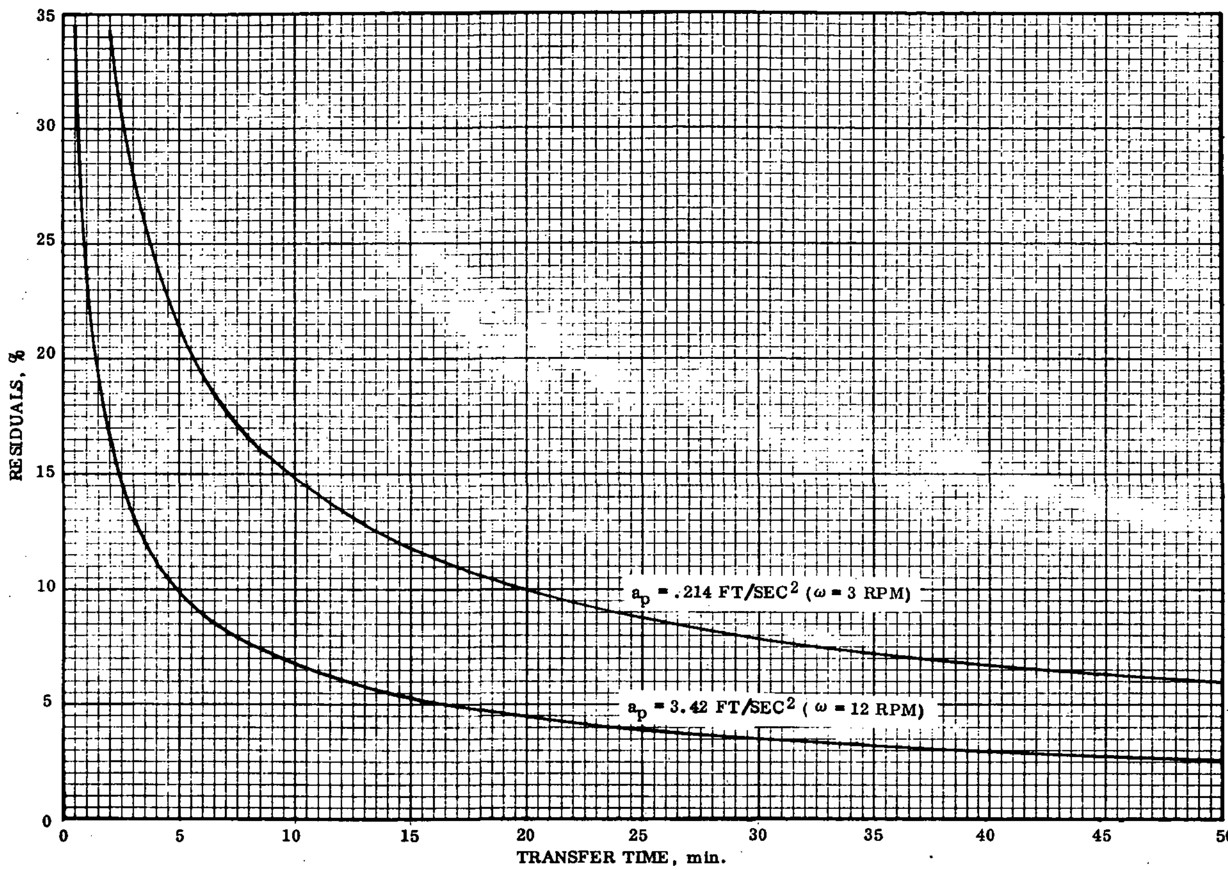


Figure 5-71. Effect of Acceleration on Residuals for 52-Inch Diameter Tank

6

MODULAR TRANSFER

The modular system relies on the exchange of storage tanks between the supply vehicle and space station to accomplish the required fluid transfer. Definition of the hardware requirements and development of parametric weight data for the modular concept are presented in the following sections.

6.1 SYSTEM DEFINITION

The interdependence of Space Station operation and docking, Shuttle operation, stowage and transfer method, and ground launch operations dictates that all must be considered simultaneously for the development of a system optimized for its total operational cycle.

The mechanical docking system and the fluid handling system must be considered jointly. The manufacturing tolerances and mating clearances of the two systems are additive; i.e., fluid sealing and valve actuating loads may be significant fractions of the structural loads borne by the docking mechanism - perhaps dominant in the low-gravity environment.

The approach for the initial definition phase of this study was to select a gross concept providing the best performance for each subsystem; then to integrate the subsystems, while modifying the subsystem concepts as required for compatibility and improvement of overall system performance.

The overall transfer system was divided into the following subsystems according to phases of the overall resupply operation.

a. Shuttle systems to meet ground fill and drain, boost and transfer requirements.

b. Docking system for orientation and lead-in, capture and rigidization at the Space Station.

c. Service systems to provide electrical and fluid connection and sealing at the Space Station

6.1.1 SHUTTLE SYSTEM OPERATIONS. Shuttle System operations will be considered only to the extent that operating philosophies affect the tank-to-spacecraft interface. With respect to the shuttle system, three basic modes of operation of the shuttle vehicle as a modular tanker were investigated. These are:

a. Previously chilled and filled modular dewar tanks loaded on the shuttle at some time before launch.

b. Empty dewar tanks loaded on the shuttle and filled during launch countdown.

c. In-orbit fill, during which tank modules insulated only for service in space are filled just prior to removal from the shuttle, from a dewar tank which remains in the shuttle.

The above operational modes are described and compared in Figure 6-1. The various comparison criteria are discussed below.

Weight. Mode a has the weight advantage in that propellant loading equipment is not carried on board the Shuttle, while Mode c must carry a complete transfer system and additional tank.

Valve Sealing. Mode a requires an additional seal operating cycle per flight with the attendant possibility of damage, while Mode c tanks spend the least time disconnected from an adapter.

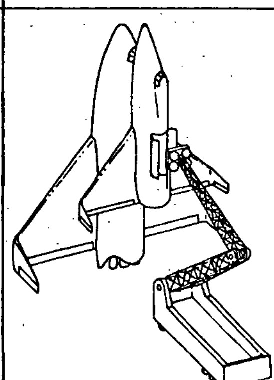
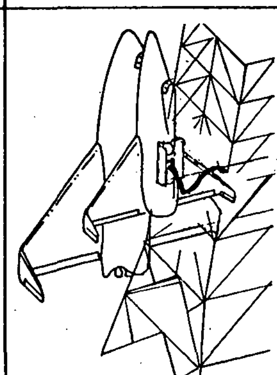
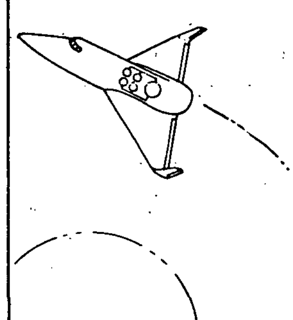
		MODE a - LOAD FULL TANKS	MODE b - FILL PRE-LOADED TANKS	MODE c - FILL IN FLIGHT
OPERATION		PLACE FILLED AND CHILLED TANKS IN SHUTTLE	FILL THRU CONNECTION IN TANK ADAPTER IN SHUTTLE	FILL TANK FROM IN-FLIGHT SUPPLY BOTTLE
				
Tank Module Requirements		Dewar, takes Boost Loads Full	Dewar, takes Boost Loads Full	Tank modules only space-insulated, take boost loads empty. Supply is dewar.
Relative Merit (Higher Number - Greater Merit)	Weight	3	2	1
	Valve Seal Req'd	1	2	3
	Shuttle Mod Req'd	3	2	1
	Complexity	3	2	1
	Operating Flex.	1	3	2
	In-Flt. Work Load	2	2	1
	Safety	1	3	2
	TOTAL	14	16	11

Figure 6-1. Comparison of Shuttle System Operating Modes

Shuttle Modification. In addition to the adapter and pressure monitoring and control required in Mode a, Mode b must add tanking control, and Mode c must add tanking control and transfer control.

Complexity. Complexity is assumed to increase as on-board functions are added and is closely tied to the Shuttle modifications above.

Flexibility. Mode a has no detanking capability after the bottles are installed, other than an abort dump. Mode c requires a pressurant loading to be sequenced during countdown, which detracts from countdown flexibility.

Inflight Work Load. Modes a and b are similar but Mode c requires an additional fluid transfer step.

Safety. The act of loading a full cryogenic tank (Mode a) is inherently more hazardous than the transfer of fluid in pipes and hoses. Mode c probably will require a high pressure pneumatic bottle with its added hazard.

Based on the data presented in Figure 6-1, Mode b was chosen as the operational philosophy used to define configurations that affect the tank-spacecraft interface. Mode a was rejected primarily because of the operating hazard and complexity associated with ground transfer of filled dewars during the Shuttle countdown. Mode c was rejected because of the heavy demands on Shuttle payload capacity of an extra tank and fluid transfer equipment. Additional factors are operational complexity and reliability penalties.

Several assumptions were required to develop the detail design concepts for the mechanical components of the system. The assumptions used are presented below.

a. The initial tank size considered was a 42.5 cubic ft sphere, containing 3000 pounds of fluid at 100 psia. This corresponds to a nominal space station liquid oxygen transfer case.

b. For tanking, a 1-1/2 in. tubing size connection is required. Service and thermal conditioning connections are 3/4 in. size.

c. The detailed means of effecting transfer of the tank from the Shuttle to the spacecraft was not a subject of this study, however, maneuvering accuracies to be expected during the docking operation were taken to be comparable to those experienced in the docking of the Gemini and Apollo spacecraft. Some typical cargo handling modes to be expected are illustrated in Reference 6-1.

d. Boost phase loading is 4.8 g longitudinal and 0.48 lateral with the service load being 10^{-4} g in any direction. The boost phase requirements are based on the possibility of using the Saturn vehicle to transport space station elements to orbit.

e. A 12 conductor harness is used to monitor the mass and temperature of the tank contents.

f. Two sets of "hard" points must be provided: one set for attachment to the device which is to remove the tank from the Shuttle and mate it to the spacecraft, and one set, including fill and drain, vent, and monitoring connections for attachment to the Shuttle or the spacecraft.

g. The tank is to be transferred between the Shuttle and the spacecraft in the locked-up condition.

The orientation of the bottle in the Shuttle is governed by several considerations. Both a vent/pressurization connection and a fill/drain connection are required. To provide a more rapid drain capability in the event of an abort, it is desirable that the liquid connection be open directly to the bottom of the tank as it sits in the Shuttle.

Also, to minimize the loads induced by transverse booster acceleration, the support points should be arranged symmetrically about a line parallel to the roll axis of the booster. For insertion and withdrawal from the adapter by the transfer device, the attach points should be well clear of the Shuttle and the motion in an inboard-outboard direction.

Based on the above considerations the configuration shown in Figure 6-2 was developed. The adapter shown in Figure 6-2 is locked in the position shown for fill and drain and in flight. For transfer of the tank, the adapter is unlocked and rotated 90° so as to deploy the tank for ready access by the transfer device with the attach points exposed. This arrangement also facilitates the insertion of the empty bottles in the horizontal Shuttle.

6.1.2 DOCKING SYSTEM.

6.1.2.1 Orientation and Lead-In. Three alternate configurations of orientation and lead-in devices which were considered are shown in Figure 6-3.

A system which permits the tank to be docked in any orientation incurs a weight penalty, due to the structural requirements of a "hard" ring capable of withstanding the structural attachment loads at any point, in contrast to the minimum three hard points necessary when the tank can be oriented. Experience with Gemini and Apollo indicate that precise docking maneuvers are not only feasible, but relatively easy to perform.

The comparison in Figure 6-3 shows a weight and operational advantage for the triangular probe and receptacle configuration, which is compatible with both Shuttle and spacecraft. The weight advantage stems from the substitution of "hard" lines for "hard" surfaces to resist docking impact. This configuration envisions steel wear

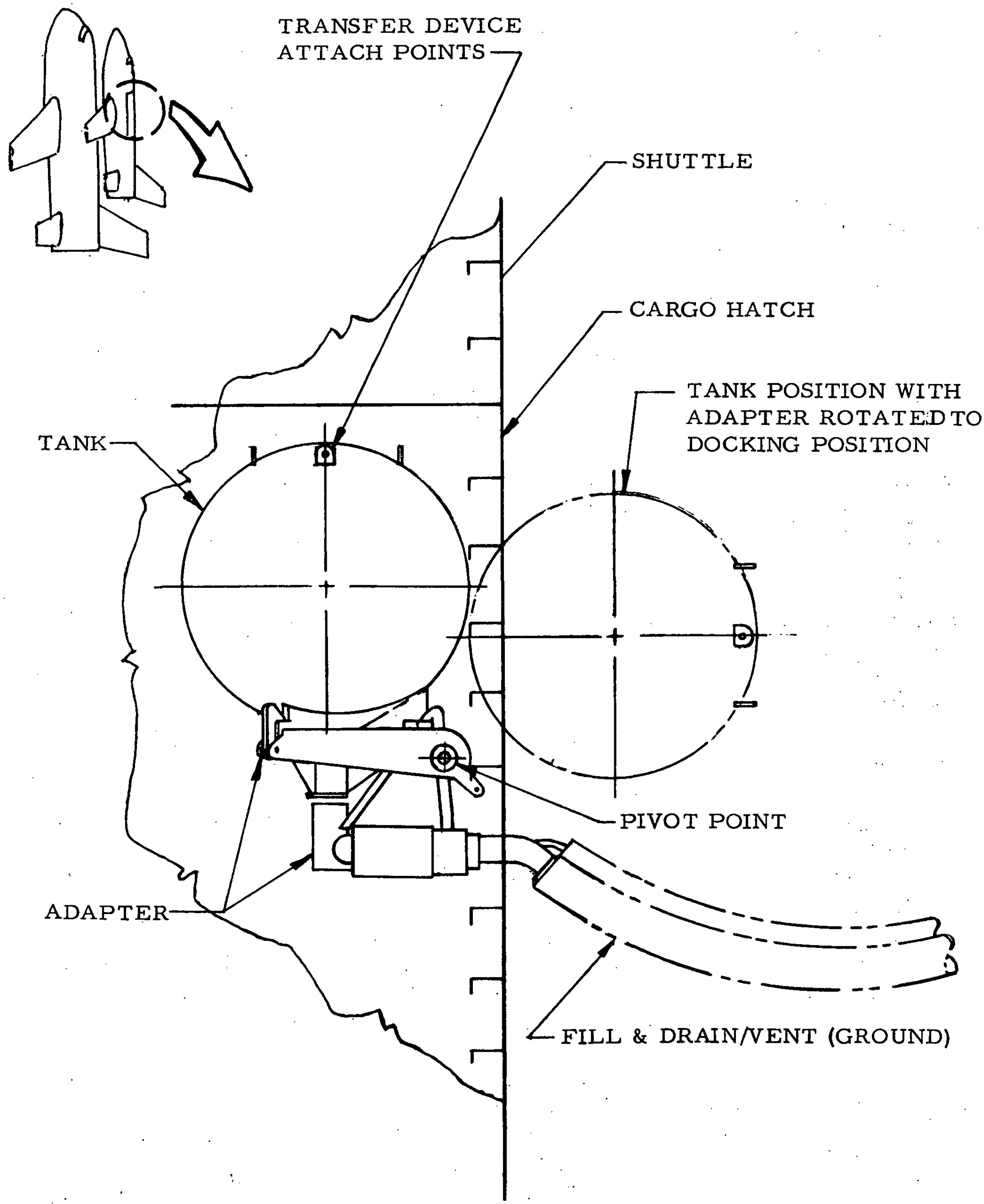


Figure 6-2. Modular Tank Configuration as Installed in the Shuttle

plates protecting the inner corners of the spacecraft receptacle and replaceable aluminum bumpers on the tank probe corners. These bumpers would be replaceable during refurbishment as required. The initial contact must be made while oriented in roll within 45° of the final position; the tank is then guided mechanically to a precise capture orientation.

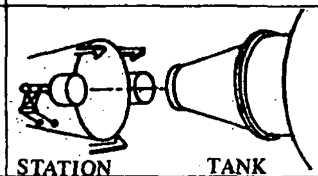
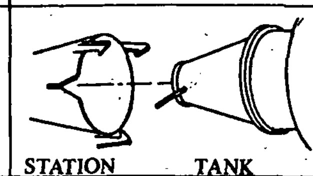
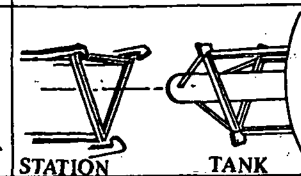
6.1.2.2 Capture. The capture mechanism considered to best meet all the requirements of the present application was the spring-loaded hook. This device is considered to be the most simple, therefore the most reliable. Other possibilities include a ball-lock arrangement or an interrupted thread. All configurations shown in Figure 6-3 use only 3 hooks, to simplify rigging and eliminate tolerance problems associated with larger numbers.

6.1.2.3 Rigidizing. The generous clearances required for a practical capture maneuver are in conflict with the precise alignment required for the successful mating of the fluid and electrical connectors and the rigid attachment required for structural integrity. Therefore an additional operation is required to "cinch-up" the tank after it is captured, and before the service connections are mated. This operation may be accomplished in several ways, as shown in Figure 6-4.

All illustrated methods appear feasible, with the final selection more dependent upon tank size than upon any intrinsic advantage. Relative motion between the hook and the abutting surface of the receptacle, as shown in methods 1 and 4 of Figure 6-4, is, however, the most universally applicable over the full range of bottle sizes when considering the requirement for compatibility with the Shuttle as well as the spacecraft. Method 5 is competitive only if a gasket type seal is used. The various systems are discussed below in conjunction with the comparison criteria.

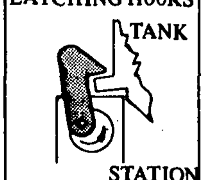
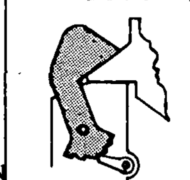
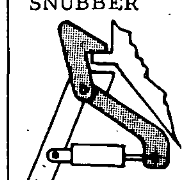
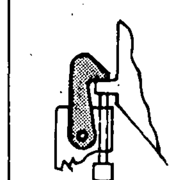
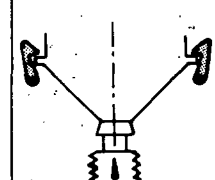
Complexity. Method 3 adds only a 3-way valve to the basic capture mechanism to admit pressurized fluid (assumed to be available from other systems) to the snubbers, causing them to act as actuators, forcing the tank outward against the hooks. Methods 1 and 4 add three electric-motor-driven actuators. Method 5 may require two probe actuators, due to incompatibility between seal pressure and rigidizing force requirements. Method 2 requires that the hooks be mechanically released and then driven open to unlock. Locking forces would be high due to the feedback of hook loads to the locking device.

Weight. Method 3 adds only a valve and tubing, but in common with method 5 it increases hook loads (under lateral accelerations) over those induced by other methods due to the central location of the outboard component of the restoring couple. Method 2 may require a powerful unlatch drive to overcome the tendency of friction to prevent hook disengagement. Methods 1 and 4 differ only in the small weight penalty to convert rotary to linear motion in the rigidizing actuators.

	CONICAL DROGUE, MECHANICAL ROLL AFTER CAPTURE	GEMINI	TRIANGULAR DOCKING GUIDE AND RAILS
			
WEIGHT	1	2	3
COMPLEXITY	1	2	3
ROLL SENSITIVITY	3	1	2
REFURBISHMENT	2	1	3
TOTAL	7	6	11

NOTE: HIGHER NUMBER INDICATES GREATER MERIT

Figure 6-3. Orientation and Lead-In System Comparison

METHOD	1 LATCHING HOOKS	2 WEDGE HOOK	3 PRESSURIZE SNUBBER	4 JACKS	5 SEAL PRESSURE
					
COMPLEXITY	3	1	5	4	2
WEIGHT	5	1	3	4	2
LEAKAGE	5	5	1	5	5
PRECISION OF POSITION	5	5	1	1	1
TOTAL	18	12	10	14	10

NOTE: HIGHER NUMBER INDICATES GREATER MERIT

Figure 6-4. Comparison of Rigidization Methods

Leakage. Only method 3 incurs the penalty of added leakage paths - and that is in the snubber actuating fluid system.

Positioning. To obtain precise location of the fluid coupling, the tolerances of the hooks and their actuators should be removed from the overall tolerance buildup. Therefore methods 3, 4, and 5 were penalized in this area.

Based on the data presented in Figure 6-4, method 1 was chosen as having the best potential for the present application.

6.1.3 SERVICE SYSTEMS.

6.1.3.1 Electrical Connection. It is assumed that fluid quantity and condition monitoring requirements will be met by tank mounted equipment connected to the parent vehicle by a 12 conductor harness.

Assuming concentric fluid connections, it would be the electrical interface that imposes orientation requirements on the tank docking maneuver. Several methods of making the electrical connection in such a manner as to avoid the orientation requirement were examined and discarded as discussed below.

a. Arrange contacts concentrically to the fluid connection. This requires a spacecraft-mounted disconnect and switch to assure proper connections from a randomly oriented tank. This imposes a reliability and weight penalty considered to be unwarranted by the operating convenience of freedom from orientation requirements.

b. A conventional connector with sufficient harness length to permit manual connection from any orientation by a crewman external to the spacecraft. The requirement for EVA severely penalized this proposal; however if other simultaneous external operations require the presence of a crewman, this method should be reconsidered.

c. An arrangement similar to the above, but being connected by a through-hull manipulator, was rejected because of its complexity and weight.

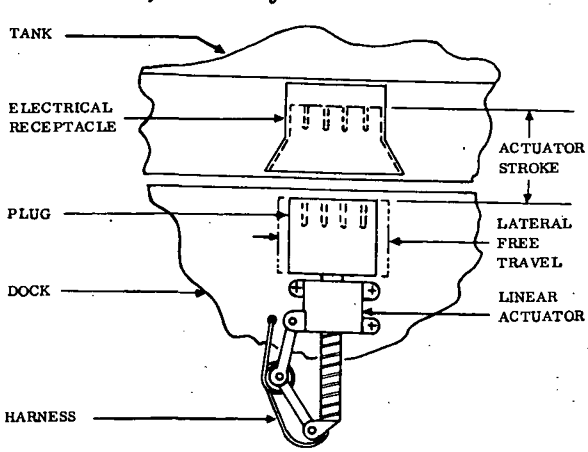


Figure 6-5. Remote Electrical Connection

If the tank is correctly oriented, the electrical connection can be made remotely by a relatively simple, straight-line actuator with limited misalignment tolerance and a generous lead-in. This system, as shown in Figure 6-5, was chosen as best for the present application.

6.1.3.2 Fluid Connection and External Sealing.

Connection. Connection for the transfer of fluids to and from the tank may be made concentrically, or separately by means of flexible or rigid connections as illustrated in Figure 6-6.

All important considerations weigh in favor of concentric fluid connections. This arrangement is the most insensitive to orientation, simplifies operations, and halves external leakage paths for both heat and fluid.

The choice between flexible (hose) and rigid connections is also clear. The hose connections require EVA to connect. Further, it would be more difficult to effectively insulate the hoses. Therefore the configuration shown in Figure 6-6d was chosen for the present application.

Leakage. Acceptable leakage during launch and transfer is an order of magnitude higher than that permitted over the 6-months service. Leakage of the seal between the outlet and the recirculating line, with its pressure differential of 1 or 2 psi, would result only in reduced performance of the fluid conditioning system in service, or the venting of some liquid during chilldown and fill. The most critical seal, therefore, is the tank-to-spacecraft interface, failure of which results in a direct loss of fluid.

This critical seal should be redundant, with the yielding elements mounted on the tank so that they may be maintained during refurbishment on earth.

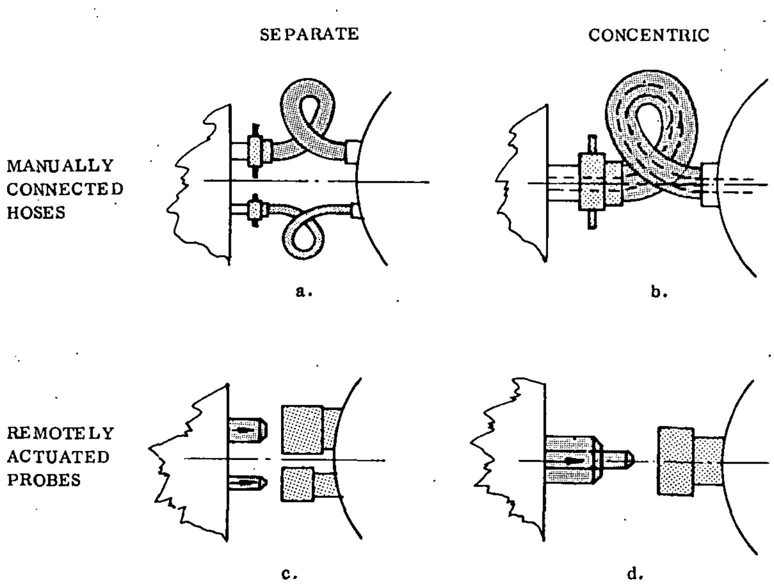


Figure 6-6. Various Fluid Connection Schemes

In the case of a 52-inch tank, considering 10% fluid loss to leakage over the six month service life to be acceptable, a leakage rate of 1.4×10^{-3} cc/sec liquid would be indicated. The required quality of the seal, expressed as permissible leakage, is proportional to the volume of the tank.

A compression seal would require 5000 to 7000 psi gasket load to maintain the required seal integrity. A 2.0 inch dia, .15 inch wide seal would then require that about 5000 pounds thrust be maintained under all conditions of structural and thermal deflections. In contrast, a probe using a self-energizing seal could be inserted, and the seal established, with a force less than 300 pounds. Thus a 500 pound linear actuator would suffice to drive the probe, and, as will be shown in the following discussions, actuate the valves as well. However, the wiping action of the self-energizing seal would incur the probability of damage to the probe from foreign matter in the form of axial scratches. A poppet arrangement would be capable of functioning after sustaining damage from small particulate contaminants. A final configuration was chosen with both a self energizing lip seal and a poppet seal occurring in series to provide a form of redundancy against external leakage.

After the tank is connected and sealed to the spacecraft or the Shuttle adapter, the service and recirculation paths must be opened. This may be accomplished by any of the means presented in Table 6-1.

Table 6-1. Comparison of Fluid Valve Opening Methods

Method	Complexity	Req'd EVA	External Leakage Path	Weight	Total
Electric	2	5	3	2	12
Manual	5	0	4	5	14
Remote External	1	5	4	1	11
Internal Poppet	4	5	5	5	19
Pneumatic	3	5	2	3	13

(High number assigned to greatest merit.)

The high figure of merit for the poppet assumes that it will be actuated by motion of the connecting probe after an external seal is established. Figure 6-7 illustrates the selected configuration for such a valve.

In the illustrated arrangement (Figure 6-7), the initial contact of the probe establishes a seal between the outer tube of the probe and the lip seal. Immediately following the establishment of this preliminary seal, the nose of the probe contacts the recirculation (inner) poppet. Acting against the inner poppet spring and internal tank pressure, continued probe motion lifts the inner poppet from its seat in the outer poppet. This pressurizes the probe cavity with tank contents, with loss of fluid prevented or retarded by the lip seal. Relative motion between the inner and outer poppets continues until the spherical end of the inner probe tube contacts and seals against a conical seat in the outer poppet. This contact seal separates the service outlet from the recirculating connection and is subject to only the pressure differential that exists through the standpipe or recirculating duct to the opposite side of the tank. Continued probe motion now raises the outer poppet from its seal until a spherical collar on the outer probe tube contacts the final seal. Increasing thrust on the probe to establish the required final seal pressure completes the connection sequence.

For disconnecting the tank, withdrawal of the probe reverses the sequence of events.

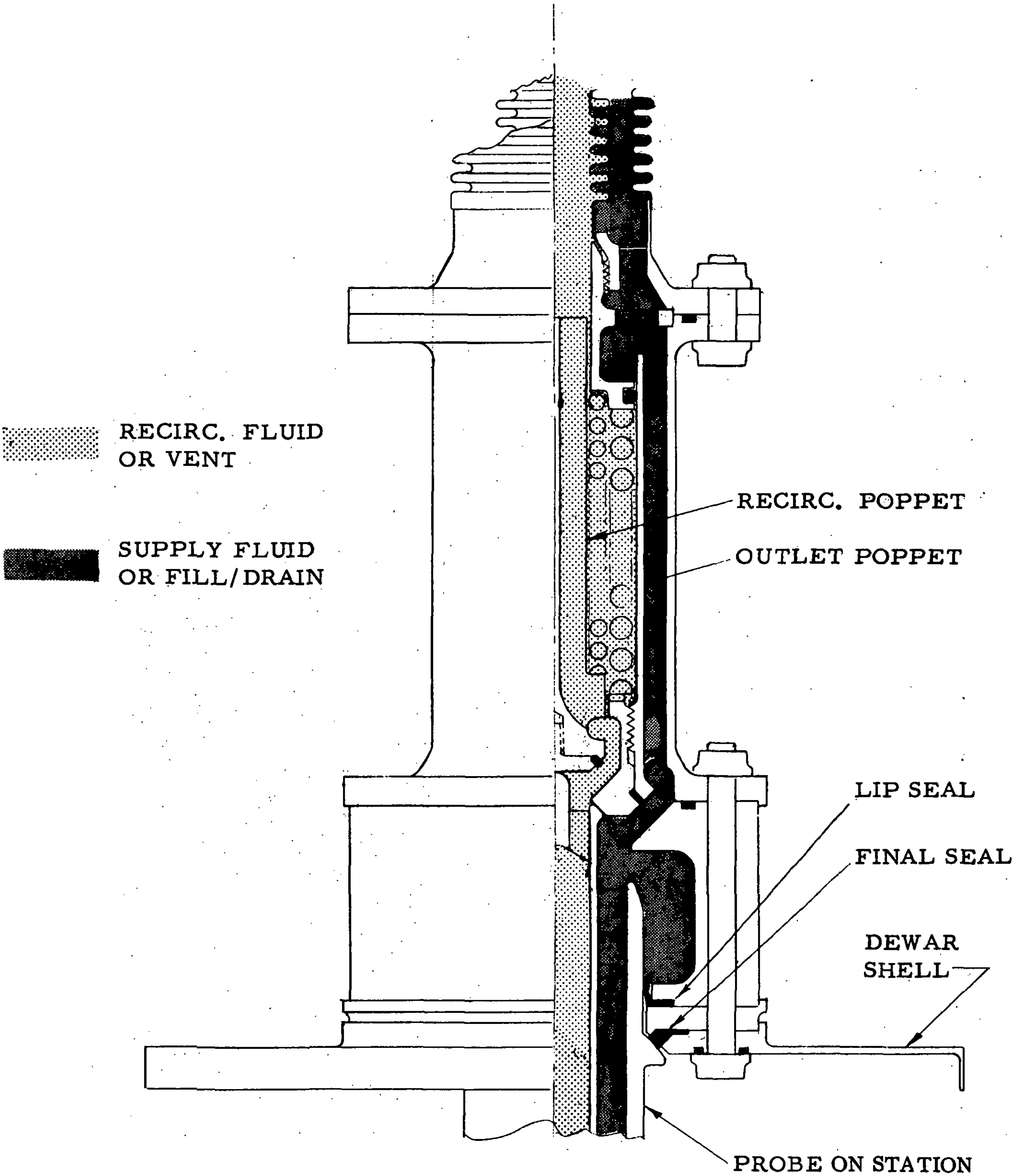


Figure 6-7. Tank Fluid Connection Valve

6.1.4 THERMAL ANALYSIS. Initial design associated with docking and capture mechanisms showed that a considerable amount of structure was required, such that if connected directly to the storage tank a relatively high heat leak to the fluid contents could result. Therefore, these mechanisms must be isolated from the storage bottle itself. This can be accomplished by employing standard dewar design practice where an inner shell is isolated from a vacuum jacket with low conductive supports and long, low-conductive feed lines. Where a dewar is used, the docking and coupling hardware would be incorporated into the jacket. If dewars were not to be used a partial shell designed specifically for docking could be employed.

In the present case the use of a dewar will be assumed and the modular designs developed accordingly. Thermal protection requirements for the mocular system were analyzed to determine any potential problems. Space station boiloff limits per Section 2 were assumed, i.e., 50 percent H_2 and O_2 and 100 percent LN_2 boiloff in 180 days. The applicable equation for determining the allowable tank heating under these conditions is

$$\dot{Q}_a = \frac{V_t\,(F_L)\,\rho_L\,\lambda\ (\%\ \text{BOILOFF ALLOWED}/100)}{(180\ \text{DAYS})} \tag{6-1}$$

Taking H_2 to be the worst case where

$V_t = 42.5\ ft^3$

$F_L = 0.95$ (initially 95% full tank)

$\rho_L = 3.56\ lb/ft^3$ (saturated at 100 psia)

$\lambda = 141.5$ BTU/lb

then from Equation 6-1 the total allowable heat leak would be 2.35 BTU/hr. It is noted that heating during space storage following transfer is then the limiting factor since the data of Section 5.1 showed that 4.25 BTU/hr would be allowable for an unmixed locked-up tank prior to transfer.

For the actual heat transfer through the insulation the following equation is applicable

$$\dot{Q}_i = K_{eff.}\ \frac{A_s\,\Delta T_i}{t_i} \tag{6-2}$$

Assuming the use of a fairly high density, low conductivity insulation such as the NRC type, then from Reference 6-2, $\rho_i = 2.2\ lb/ft^3$ and $K_{eff} = 5.2 \times 10^{-5}$ BTU/hr-ft-°F. For a 42.5 ft^3 ($A_s = 59\ ft^2$) tank with 8 inches of insulation per the baseline presented in Section 2 and with ΔT_i = 398°R then from Equation 6-2, $\dot{Q}_i$ = 1.83 BTU/hr. This leaves 0.52 BTU/hr for supports and penetrations.

Calculations for a 42.5 ft^3 tank with 8 inches of insulation showed that the use of an 18 inch strut would be reasonable. Data from Reference 6-3 showed the total heat leak from six such low conductive struts for the present fluid condition to be a maximum of 0.2 BTU/hr. This then leaves 0.32 BTU/hr for the lines.

Assuming the use of one 1.5-in. × 0.016 in. CRES line and one 0.75-in. × 0.010-in. CRES line and ignoring heat intercepted by the vent fluid, the lengths required to meet this 0.32 BTU/hr requirement was found to be 51 inches. This length is longer than would normally be desired for the present configuration. However, use of expansion bellows to increase the effective conduction length and intercepting a major portion of the line heat by the vent fluid will reduce the heat leakage of a reasonable length line to acceptable limits.

6.1.5 OVERALL SYSTEM. Combining data from the preceeding paragraphs resulted in the design of a tank and docking mechanism as shown in Figures 6-8 and 6-9.

The tank consists of inner and outer spherical shells, with the evacuated annulus containing an eight-inch blanket of superinsulation. A reinforcing ring on the outer shell, or vacuum jacket, distributes point loads introduced by the docking mechanism. The inner shell is supported by low-conductivity struts connected, via fittings through the vacuum jacket, direct to the docking latch seats.

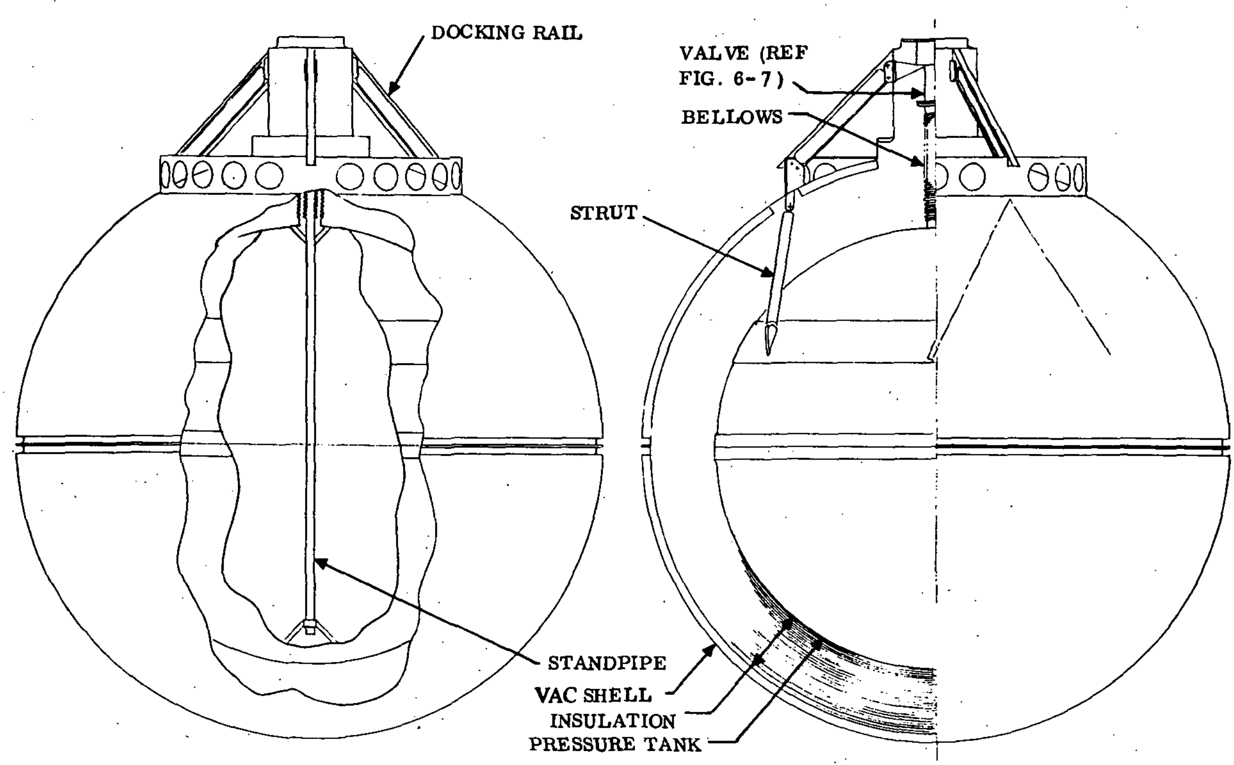


Figure 6-8. Representative Modular Transfer Tank

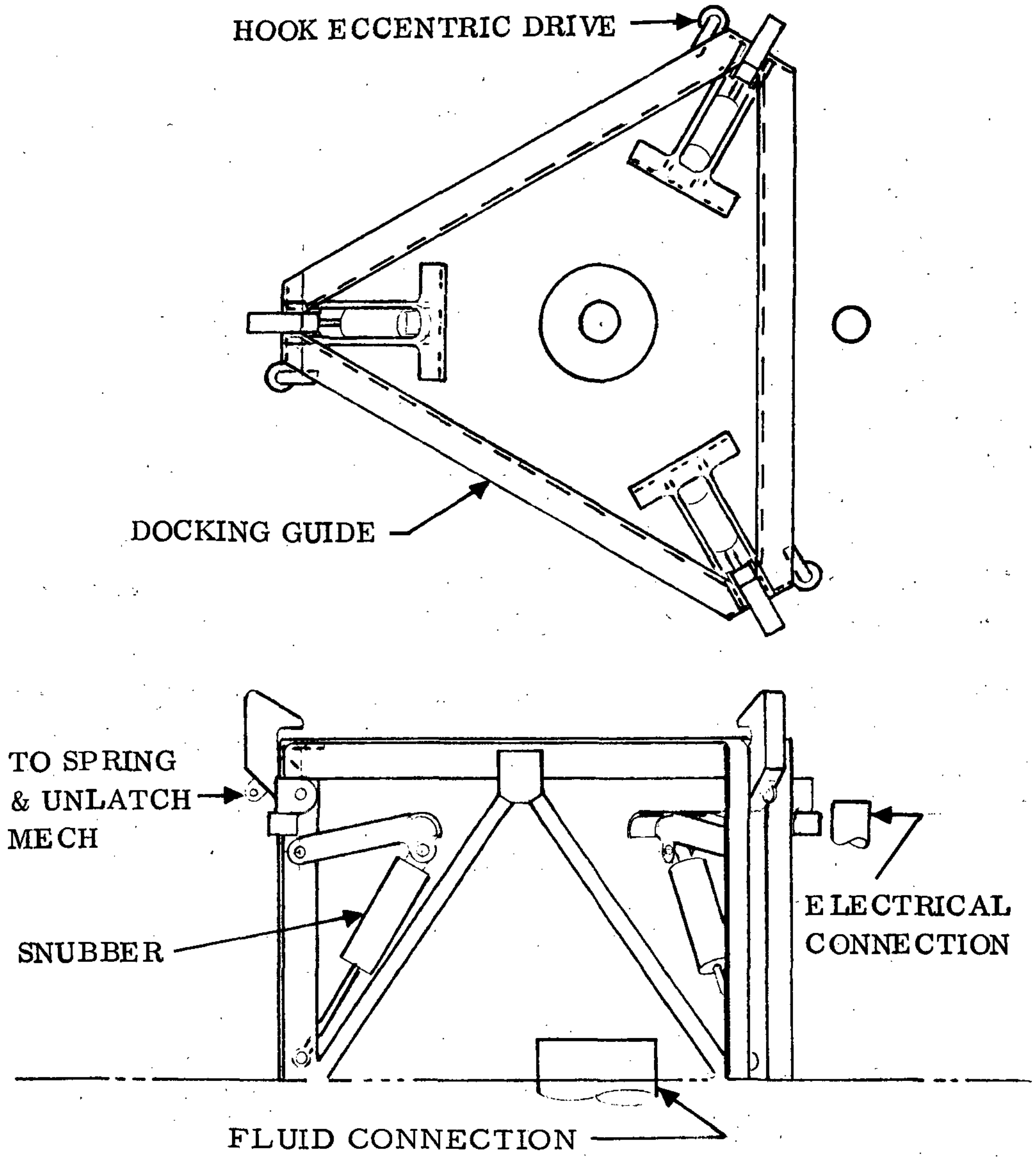


Figure 6-9. Modular Tank Docking Receptacle

These strut design details, developed at Convair, are presented in Reference 6-4. It is noted that the mechanisms associated with the docking probe or system connections are structurally attached to the outer jacket. This allows the heat transfer to the fluid and electrical lines to be isolated from the external environment in a manner similar to that used in conventional dewar design.

The probe, a cylindrical extension of the vacuum jacket, contains and locates the Figure 6-7 valve and its connections. It is supported by the docking rails, which extend from the nose of the probe to the latch seats on the reinforcing ring. The valve, located in the end of the probe, connects to the tank by two concentric CRES bellows. The inner bellows connect to a standpipe which extends to the opposite side of the tank. The annulus contained by the outer bellows connects the valve to the near side of the tank.

The standpipe is the fluid recirculating (and pressurization, if required) connection in flight, and vent connection on the ground or in the shuttle. In an abort condition it will serve as a pressurization connection to assure rapid expulsion of the tank contents.

The docking receptacle shown in Figure 6-9 consists of a triangular array of docking guides arranged to engage the docking rails on the tank at their junction with the reinforcing ring.

As the tank probe enters the receptacle, the docking rails contact the snubber arms, and any excess tank closing velocity is dissipated by the snubbers. The tank is rotated during closing by interaction of the rails and guides to correct any roll misalignment as the rails nest in the apex of the angles formed by the guides. Pitch and yaw alignment is secured when the three latch seats are seated on the surface of the rails.

A spring loaded hook at each apex engages the latch seat on the tank. The outer end of the hooks are ramped so that the latch seats will force the hooks outward until the seats have passed the ramps, when the springs will cause the hooks to engage the seats. The hooks pivot about an eccentric on their shafts. After the hooks are engaged and the tank is captured, the hook shafts are rotated by electric motors to draw the tank into final engagement with the receptacle, taking up the capture clearance, and preloading the hooks to a degree determined by motor stall torque. A worm gear in the motor gear train prevents the hook from driving the motor and becoming loose.

Upon completion of the rigidizing step, the electrical connection can be made. A linear actuator drives the plug on the space station into the receptacle on the tank.

The fluid connections are then made. This is accomplished by a linear actuator, driving the probe through a preloaded, constant load spring, to a predetermined spring deflection. Thus sealing forces will be maintained over the range of thermal and load deflections of the supporting structure, probe and docking mechanism.

Figure 6-10 shows the tank in the final docked position and Figure 6-11 shows details of the latching mechanism.

6.2 DETAIL ANALYSIS AND DEVELOPMENT OF PARAMETRIC DATA

The primary work reported in this section is the development of detailed weight data for the range of tank sizes under investigation. The overall system configuration as shown in Figures 6-8 through 6-11 represents the system for which the weights are generated.

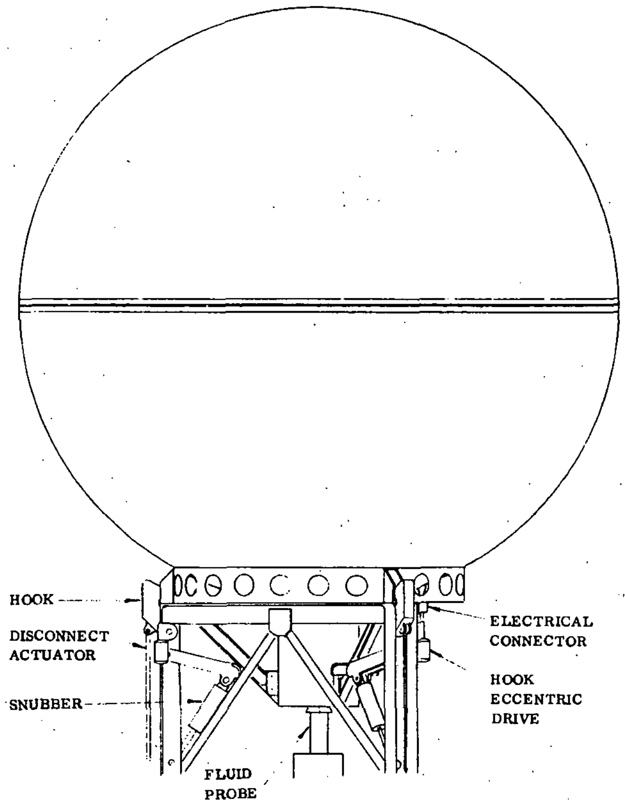


Figure 6-10. Docked Tank

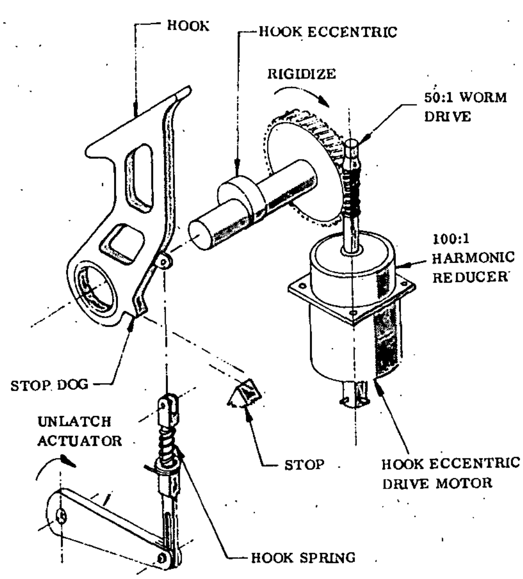


Figure 6-11. Latch Mechanism Details

6.2.1 TANKAGE AND INSULATION. Basic inner tank and vacuum shell weights were obtained from the data in Appendix A. The inner tank is assumed to be aluminum and the vacuum shell is of a honeycomb sandwich type construction.

The actual vacuum jacket weight for a particular tank size is a function of its diameter and thus the corresponding insulation or minimum spacing requirements. The minimum gap or spacing between the inner tank and shell was determined from Figure 5-1 for strut type supports. Insulation requirements are based on the use of Equations 6-1 and 6-2.

Insulation thickness for a 42.5 ft^3 (52 inch) diameter tank was taken as 8 inches per the data in Paragraph 6.1.4. Assuming that the fraction of heating through the insulation in relation to the total heating remains constant, then $\dot{Q}_i \approx \dot{Q}_a$ and from Equations 6-1 and 6-2

$$\frac{V_t\,(F_L)\,\rho_L\,\lambda\,(\text{\% BOILOFF ALLOWED}/100)}{(180\ \text{DAYS})} = K_{eff.}\,\frac{A_s\,\Delta T_i}{t_i}$$

Assuming F_L, ρ_L, λ, % boiloff, K_{eff}, and ΔT_i to be constant as a function of tank size

$$t_i = \text{CONSTANT}\,(A_s/V_t) = \text{CONSTANT}/D_t$$

Referring to the 8 inch insulation with a 52 inch diameter tank

$$t_i\text{, in.} = 415/D_t\text{, in.}$$

This insulation thickness is plotted in Figure 6-12 as a function of tank diameter.

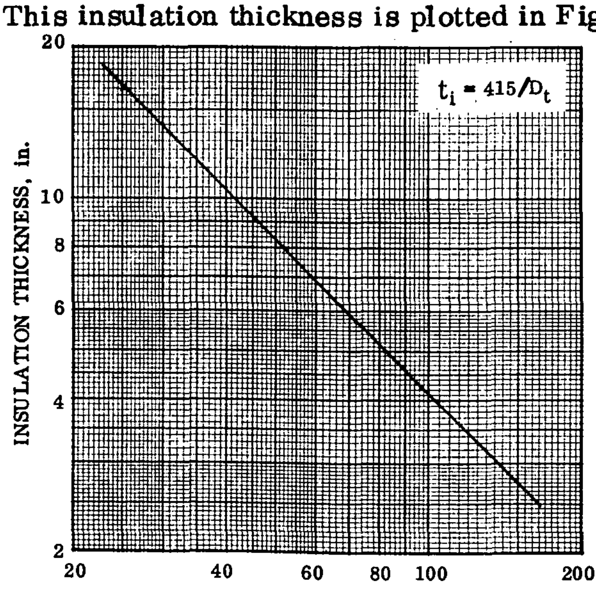


Figure 6-12. Insulation Thickness for H_2 Modular Transfer System

Taking the insulation density as 2.2 lb/ft^3 per Paragraph 6.1.4 and insulation thickness from Figure 6-12, insulation weights were determined and are presented in Figure 6-13.

6.2.2 DOCKING MECHANISMS. To determine a baseline weight estimate for the components peculiar to the modular transfer system, these components were sized for the nominal 52 in. dia tank under consideration. Assumptions used were:

a. Final docking velocity, $V_e = 1.0$ ft/sec.

b. Total docking mechanism deflection

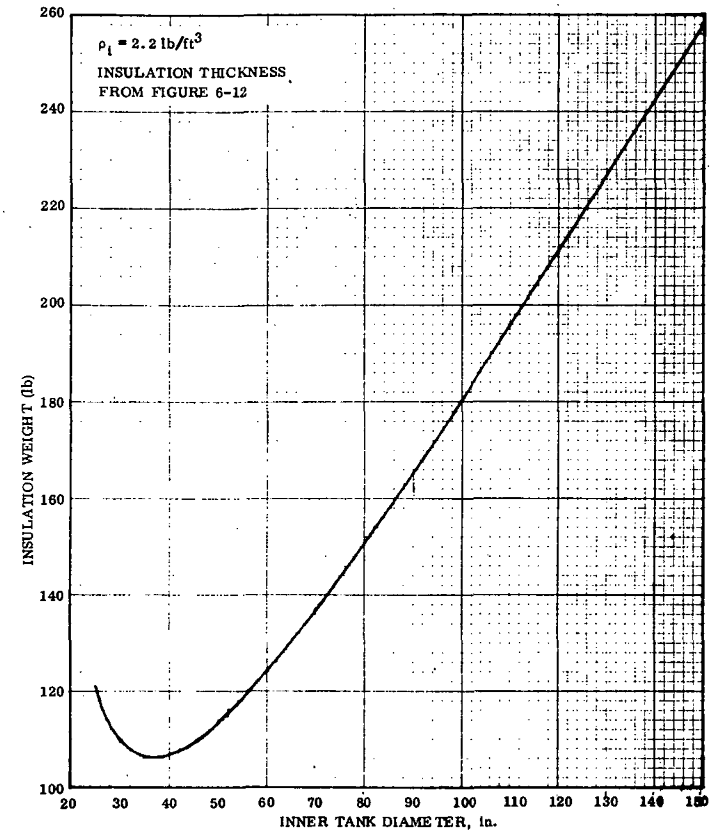


Figure 6-13. Modular System Insulation Weight Based on H_2 Requirements

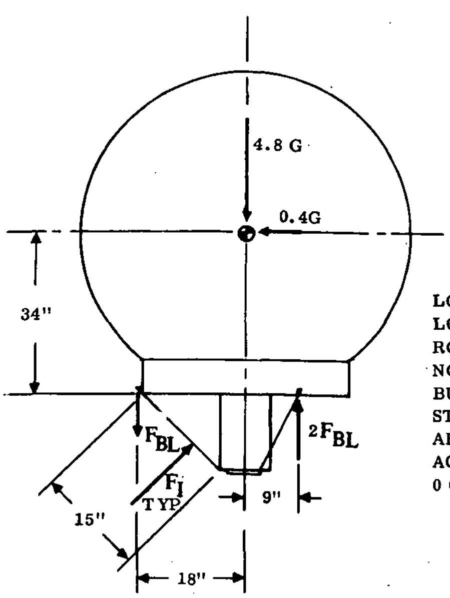


Figure 6-14. Illustration of Design Docking Loads

0.2 in., divided 2:1 between tank rail and receptacle guide.

c. Eight inches of 2.2 lb/ft^3 insulation used.

The docking impact load F_I, and boost phase latch loads F_{BL} are illustrated in Figure 6-14.

Axial travel s to arrest docking velocity

$$s = 0.2\,\text{in.} \times \sin 45° = .282\ \text{in.}$$

Acceleration

$$a = (V_e^2/2s) = 21.3\ \text{ft/sec}^2\ \text{average}$$

Maximum acceleration = 2x average = 42.6 ft/sec^2

From Appendix A and Figure 6-13 the supported weights are

Pressure vessel (100 psi)	73 lbs
Jacket (Dia = 68 in.)	197
Insulation (Common design assumed for both H_2 & O_2)	116
LO_2	2837
	3223 lbs

Axial force = $ma = [(3223 \times 42.6/32.2] = 4260$ lb

Docking impact load, $F_I = (4160/3 \sin 45°) = 2000$ lb

Weights of individual items are developed below.

Docking Rail

Allowable deflection (y) = 0.20 × 1/3 = 0.067 in.

Rail length, L = 15 in.

Required moment of inertia $I = \frac{F_I L^3}{48 Ey} = \frac{2000\ (15)^3}{48\ (10)^7\ (.067)} = 0.210\ \text{in}^4$

Assuming an Allowable Stress (S) = 50,000 psi

$$c_{max} = \frac{4\ IS}{F_I\ L} = \frac{4 \times .210 \times 50{,}000}{2000 \times 15} = 1.40\ \text{in.}$$

The above properties, as well as torsional rigidity to resist eccentric loading from roll misalignment can be met by a hollow rectangular aluminum extrusion 2.12 × 1.00 × .062 wall, weighing .44 lb/ft, or a total of .44 × 15/12 = .6 lb.

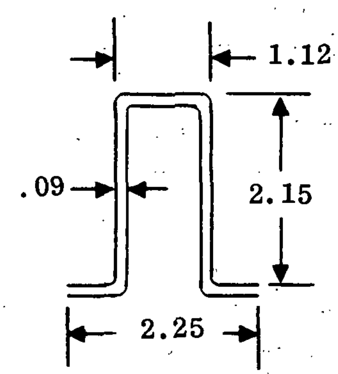
Docking Guide

Deflection, y = 0.20 × 2/3 = .133 in.

Length, L = 32 in.

$$I = \frac{F_I L^3}{48 Ey} = \frac{2000\ (32)^3}{48 \times 3.10^7 \times .133} = .344\ \text{in}^4$$

$$c_{max} = \frac{4\ IS}{F_I L} = \frac{4 \times .344 \times 100{,}000}{2000 \times 32} = 2.14\ \text{in.}$$



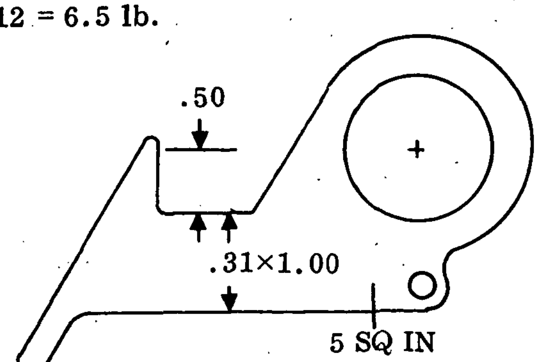
The above properties can be met by the rolled section shown to the right.

At 2.43 lb/ft, or a total weight of 2.43 × 32/12 = 6.5 lb.

Latch System. Assuming a fluid seal load of 9000 lbs, the individual hook load will be

$$\frac{9000}{3} + \frac{0.4 \times 34}{18 + 9} \times 3223 = 4700\ \text{lb}$$

$$S = \frac{4700}{.31 \times 1} + \frac{4700 \times 6}{.31 \times (1)^2} = 108{,}000\ \text{psi}$$



$$\text{Hook Wt} = .31 \text{ in.} \times 5 \text{ in.}^2 \times .29 \text{ lb/in.}^3 = .5 \text{ lb}$$

Additional Weights:

Hook shaft and bearings	1.0 lb
1/16 HP Motor, 5000:1 Reducer, Brake	3.0 lb
Spring, Housing, Mount	1.0 lb
Unlatch Mechanism (at hook)	0.7 lb
Snubber	3.0 lb
Snubber Arm	3.0 lb
Lower Rail Fitting	0.3 lb
Upper Rain Fitting	0.6 lb

A total weight summary of the overall docking mechanism is presented in Table 6-2. It is noted that only weights peculiar to the docking or modular transfer concept are included.

6.2.3 WEIGHT DISTRIBUTION. The modular method of fluid transfer requires installation of equipment in both spacecraft and shuttle. The estimated weights of this equipment, based on the 52 in. example, are presented in Table 6-3. Equipment installed on the tank, indicated by asterisk, is included in both shuttle and spacecraft, as it is assumed that (1) the spacecraft will be launched with tanks on board, and (2) the shuttle will return an empty tank to earth for each tank carried into orbit.

Table 6-2. Docking Mechanisms Weight Summary

		Latch Mech.	Other
Docking Rail	3 × .6		1.8 lb
Docking Guide	3 × 6.5		19.5 lb
Hook	3 × .5	1.5 lb	
Shaft and Bearings	3 × 1.0	3.0 lb	
Motor	3 × 3.0	9.0 lb	
Spring, etc.	3 × 1.0	3.0 lb	
Unlatch Mechanism at Hook	3 × .7	2.1 lb	
Snubber	3 × 3.0		9.0 lb
Snubber Arm	3 × 3.0		9.0 lb
Lower Rail Fitting	3 × .3		0.9 lb
Upper Rail Fitting	3 × .6	1.8 lb	
Remote Unlatch System			5.0 lb

Column 1 contains weights directly proportional to tank weight. Column 2 contains weights peculiar to the latch system and are affected by both sealing loads and tank weight, and column 3 weights are fixed - independent of tank weights.

Generalized equations for equipment weights were then derived from the above data as described below.

For the Shuttle-mounted equipment, weights are the sum of

Table 6-3. Docking Systems Weight Distribution Between Shuttle and Spacecraft

	Shuttle			Spacecraft		
	1	2	3	1	2	3
Docking Guides	19.8			19.8		
Hooks and Related Mechanisms		18.6			18.6	
Remote Unlatch			5.0			5.0
Adapter Pivot Actuator	5.0					
Adapter Pivot Structure	15.0					
Electrical Connector Actuator						5.0
*Docking Rails	1.8			1.8		
*Upper Fittings		1.8			1.8	
*Lower Fittings	0.9			0.9		
*Valve (Figure 6-7)		9.0			9.0	
Probe and Actuator			25.0			25.0
Total, lb	42.5	29.4	30.0	22.5	29.4	35.0

$C_7\ (W_T) = 42.5$ when $W_T = 3223$ (Column 1)

$C_6\ (3000 + .504\ W_T) = 29.4$ when $W_T = 3223$ (Column 2)

$C_5 = 30.0$ (Column 3)

where W_T is the total weight of tank and fluid excluding docking mechanisms.

$$\text{Wt on Shuttle, lb} = 0.00635\ (3000 + .504\ W_T) + .0132\ W_T + 30$$
$$= 0.0164\ (W_T,\ \text{lb}) + 49.05 \qquad (6\text{-}3)$$

For spacecraft-mounted equipment, weights are the sum of

$C_9\ (W_T) = 22.5$ when $W_T = 3223$ (Column 1)

$C_8\ (3000 + .504\ W_T) = 29.4$ when $W_T = 3223$ (Column 2)

$C_{10} = 35.0$ (Column 3)

$$\text{Wt on spacecraft, lb} = 0.00635\ (3000 + .504\ W_T) + .00698\ W_T + 35$$
$$= 0.01018\ (W_T,\ \text{lb}) + 54.05 \qquad (6\text{-}4)$$

Knowing the total weight of tankage, insulation and contained fluid, Equations 6-3 and 6-4 are then used to determine weights of docking systems respectively associated with the shuttle and space station.

6.2.4 EFFECT OF SEPARATE OVERALL DESIGNS FOR LO_2 AND LH_2 TANKS. It is noted that the baseline case considered in previous paragraphs assumed the use of common tankage designs for the LH_2, LO_2 and LN_2 applications. In this case the load carrying requirements are dictated by the LO_2 case and the heat transfer or the amount of insulation is determined by the LH_2 case. In order to determine the effect on weight of designing each tank system for the specific fluid to be loaded, the following analysis was performed.

Assuming LO_2 stored at 100 psia saturated conditions with $\rho_L = 64.2\ lb/ft^3$, $\lambda = 77.5$ Btu/lb, $K_{eff} \approx 1/\Delta T_i$, $F_L = 0.95$, and the hydrogen properties data from Paragraph 6.1.4 the insulation thickness required for O_2 was related to that for H_2 by the use of Equations 6-1 and 6-2. The result is $(t_i)_{O_2}/(t_i)_{H_2} = 0.101$ or the required insulation thickness for the LO_2 case is approximately 10 percent of that for H_2.

A summary of the weights resulting from using common designs versus separate designs is presented below for a 42.5 ft^3 tank.

	Common H_2/O_2 Tanks	H_2 Tankage Only	O_2 Tankage Only
Pressure Vessel (100 psi)	73 lb	73 lb	73 lb
Insulation Weight	116	116	9
Jacket Weight	197	197	142(1)
Fluid Weight for Structural Design	2837(2)	175(2)	2837(2)
Subtotal	3223	561	3061
Docking Mechanisms on Shuttle (Equation 6-3)	102	58	99
Docking Mechanisms on Spacecraft (Equation 6-4)	87	60	85
Total Dry Wt	575 lb	504 lb	408 lb
Weight Saving Over Common Tankage	-	71 lb	167 lb

(1) Gap between inner tank and jacket was one inch based on minumum from Figure 5-1.
(2) Fluid weights based on 95 percent full tank at $\rho_L = 70\ lb/ft^3$ for O_2 and 4.34 lb/ft^3. for H_2.

6.2.5 OVERALL OPERATING SEQUENCE. With a modular fluid transfer system as defined in the preceding sections, the following operating procedure is proposed.

Shuttle Preparation. To prepare the Shuttle for its modular resupply mission, tank adapters, monitoring and control consoles, boiloff ducts, interconnecting harnesses, (and possibly abort pressurization equipment and dump ducts, if filled tanks cause the gross weight of the Shuttle to exceed landing maximum) must be installed and checked. If cargo fluids are common to Shuttle service fluids and single-point fill is used, Shuttle plumbing must be modified and loading logic and control console and wiring installed.

Preflight. During countdown the cargo tanks are filled in sequence with the Shuttle tanks.

Inflight. During boost into orbit, crew duties will consist of monitoring tank conditions and pressure control function. If abort conditions require a reduction of weight, or safety considerations require the jettison of cargo fluid, the tank contents will be blown overboard.

Tank Transfer. After the Shuttle is docked to the spacecraft, transfer of the tanks is begun. The nature of this process is dependent upon the characteristics of the Shuttle. If the Shuttle payload is weight-limited rather than volume-limited, a "blind" adapter, without fluid connections, may be installed in the cargo area to receive the first empty tank. The expended tank, while still on the spacecraft, is vented to approximately 14.7 psia to minimize contamination from any leakage, while preventing collapse of the pressure vessel in the event of a leak in the vacuum jacket on re-entry into the atmosphere.

An alternate procedure, which may be necessitated by presently undetermined spacecraft operating methods, is to vent the tank completely. The Shuttle adapter would then be equipped with a probe to maintain at least one of the tank valves in the open position to vent gas from any solidified residue in the tank, and to prevent a vacuum in the pressure vessel on re-entry.

After the expended tank is secured to the transfer device, it is demated from the spacecraft by withdrawing the fluid probe and electrical connector, then removing latch loads by rotating the hook shafts to the "loose" position, and finally actuating the unlatch mechanism which retracts the hooks. The expended tank is removed from the spacecraft and placed in the adapter in the Shuttle, and secured. The transfer device is transferred to the full tank, which is then demated from the Shuttle in the same manner as the expended tank was demated from the spacecraft. The full tank is transferred to and mated to the spacecraft. Detaching the transfer device from the tank completes the operating cycle.

7

SYSTEM COMPARISONS AND RECOMMENDATIONS

The basic ground rules used in developing the comparative data presented in this section are outlined in Section 2. Overall weight, reliability, cost and crew performance comparisons are presented for the high pressure, subcritical and modular systems defined in detail in Sections 3 through 6. In all cases the emphasis in the present section is on comparative rather than absolute data.

Using the basic information contained in Sections 3 through 6, transfer system weight data are compiled for H_2, O_2 and N_2 over a range of fluid quantities transferred. Also, weight, reliability, cost and crew performance data are presented for the baseline transfer discussed in Section 2; where 1096 lb of H_2, 2480 lb of O_2 and 3,150 lb of N_2 are assumed to be transferred to eight H_2, two O_2 and two N_2 bottles located on a space station.

A layout of typical bottle locations and corresponding transfer line routing is presented in Figure 7-1 for the H_2 case. Supply of the O_2 and N_2 bottles would require a similar layout with the difference that fewer bottles are involved; however, the two bottles would in each case be located on opposite sides of the station to guard against total fluid loss in case of a single bottle failure.

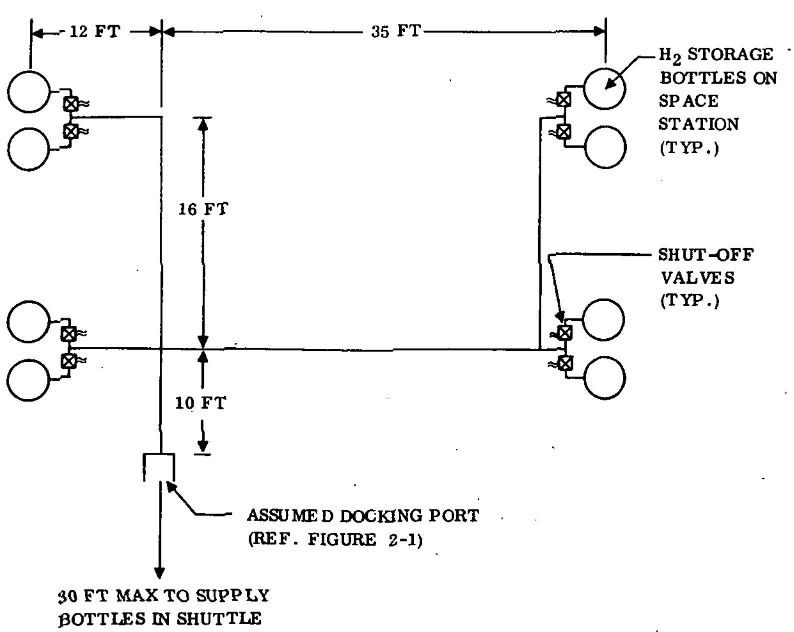


Figure 7-1. Representative H_2 System Bottle Locations and Supply Line Layout

Two supply concepts were considered. In one case individual supply bottles are assumed for each of the receivers and in the other case a single supply tank is used for each fluid.

Basic transfer system configurations and operations for which comparative data were developed are presented in the following section.

7.1 TRANSFER SYSTEM CONFIGURATIONS AND OPERATIONS

The overall transfer operation is outlined below.

a. Loading of empty tanks into the shuttle.

b. Hook-up of tank pressure monitoring equipment, overboard vent lines, fill lines, and abort pressurization as required. Abort dumping is assumed to be through the normal fill line.

c. Filling of tanks.

d. Boost, during which tank pressure conditions are monitored.

e. Rendezvous and docking. In the case of fluid transfer, coupling of fluid lines coming from the shuttle to lines on the space station is assumed to be accomplished during docking.

f. Orbital fluid transfer or tank exchange.

g. Undocking.

h. Return to earth, during which time tank pressure conditions are monitored in "depleted" tanks.

i. Safing and purging on ground.

j. Unloading of "depleted" tanks on the ground.

Individual orbital transfer systems and operations are described below.

The high pressure system determined to be applicable to the present transfer requirements is illustrated schematically in Figure 7-2. It is noted that hardware associated with fluid fill on the ground and usage at the space station are not shown and were not included in the comparisons since such hardware would be essentially common to all the systems. Mass gaging, although important to the transfer process, was also not included due to its commonality between the various systems. The ease with which mass gaging can be accomplished is, however, evaluated on a qualitative basis when comparing the overall advantages and disadvantages of each transfer concept.

Referring to Figure 7-2, high pressure transfer is initiated by opening valves V_1 and V_2, with valves V_3 and V_4 closed, and actuating the electric heater in the supply bottle. Supply tank pressure is maintained fairly constant during the transfer by the heater. In order to do this a heater control, sensing tank pressure

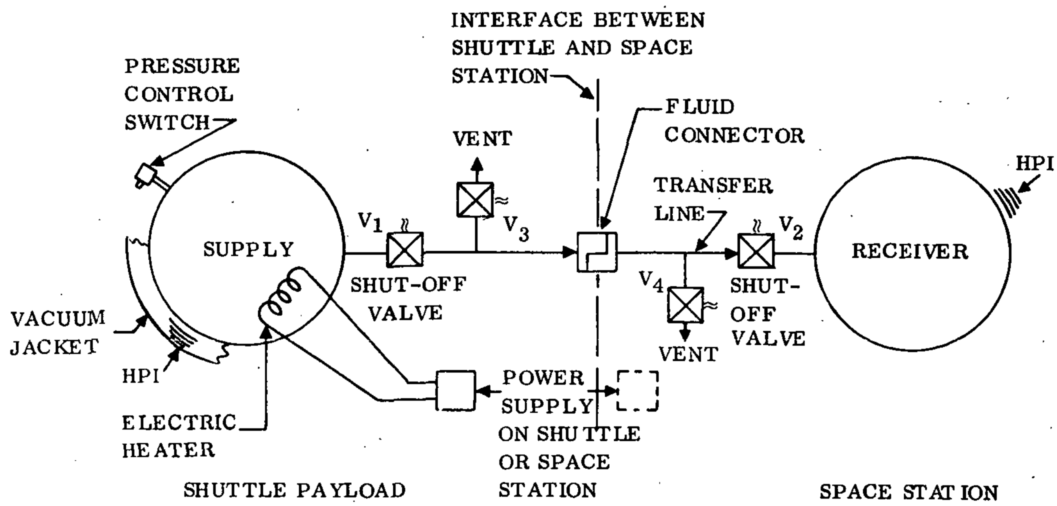


Figure 7-2. Basic High Pressure Transfer Schematic

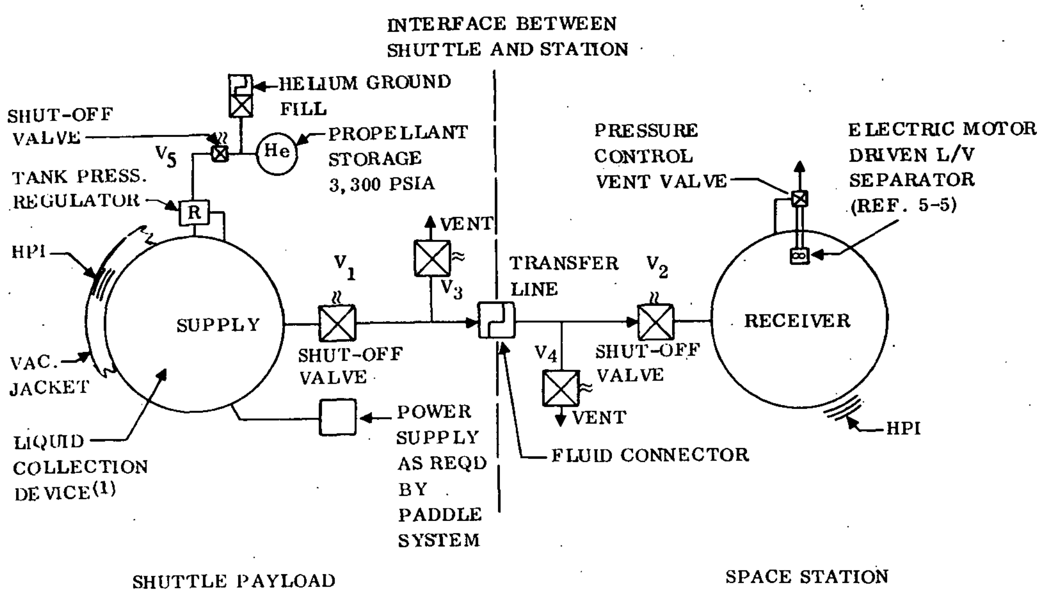
is required. This is assumed to be in the form of a pressure switch controlling heater on-off operation. The transfer is allowed to terminate when the heater, operating continuously, can no longer maintain a supply tank pressure greater than the receiver tank pressure. This condition is sensed by a comparison of the two tank pressures. The shutoff valve V_2 is then closed and the heater turned off. The vent valve V_3 is then opened to vent the supply to 15 to 20 psia at which time V_1 is closed. V_3 remains open and V_4 is opened to insure that both sections of the transfer line are vented prior to undocking. As discussed in Section 4 the initial receiver pressure is taken to be 100 psia with a residual fluid density of 0.15 lb/ft^3 for H_2 and no venting is needed during the transfer.

The power supply for the heater can either be located on the station or the shuttle. Some overall weight saving can be realized by locating on the space station, as discussed in Paragraph 7.2, since in this case a one-time boost would be involved with only the power supply fuel needing to be replenished.

The supply tank has sufficient insulation to allow a locked up (non-vent) mode of operation between final ground filling and fluid transfer at the station. The supply design pressure for use with H_2 is taken as 225 psia. The H_2 is assumed to be loaded in a saturated liquid state at 20 psia.

The basic subcritical system being considered is shown schematically in Figure 7-3. Four different methods of liquid orientation are included as indicated on the figure. In all cases expulsion is by the use of helium pressurant stored at ambient temperature.

As for the high pressure system the supply tank is assumed to exist in a locked-up condition prior to transfer. Calculations presented in Section 5 showed that it would be desirable to operate in this manner where the tank is designed for H_2, O_2 or N_2 transfer at 100 psia. In each case the fluid is loaded as a saturated liquid at 20 psia and the maximum pressure rise condition is assumed, where no mixing occurs.



(1) DETAILS PRESENTED IN SECTIONS 3 AND 5. SURFACE TENSION SCREENS PER FIGURE 5-30. BELLOWS PER FIGURE 5-52 EXCEPT IN PRESENT CASE HELIUM BOTTLE LOCATED OUTSIDE PROPELLANT TANK. DIAPHRAGM PER FIGURE 3-6. PADDLE PER FIGURE 5-61.

Figure 7-3. Basic Subcritical Transfer Schematic

In the case of the surface tension system the double liner configuration presented in Figure 5-30 was chosen for simplicity. In this system wicking is relied upon to maintain the screens wetted at all times.

A mechanical liquid/vapor separator is incorporated into the receiver tank to allow for efficient tank pressure control during chilldown of a warm tank and/or transfer line. Analyses presented in Section 5.3 showed that there is some question as to whether a warm tank of the small sizes and with the fairly long transfer lines of the present program could be chilled to H_2 temperatures without venting. The requirement to fill an initially warm tank could result from having to empty a bottle in order to make repairs or replace critical hardware. Also, analysis showed that even with an initially cold receiver tank, vaporization of LH_2 in the transfer line could cause a significant pressure rise if venting were not accomplished.

The choice of the electric motor driven separator over other schemes is based on work presented in Reference 5-5. This study indicated such a system to have the best potential for receiver tank venting under the current state-of-the-art. Passive methods for the prevention of liquid at the vent were not considered in this study and could possibly show some advantage for future systems. This is further discussed in Section 8.

The basic subcritical operations are described below.

Fluid connections and disconnections are made the same as for the high pressure system. Transfer is initiated by opening valves V_5, then V_2 and V_1. Pressurant is admitted to the supply tank and transfer occurs through the line. Receiver tank pressure is maintained during line and tank chilldown, as necessary, by venting through the liquid/vapor separator shown. Receiver tank chilldown venting is terminated on the basis of temperature sensors located on the line at the receiver tank inlet and on the receiver tank wall at several locations. All sensors reading below a certain temperature will be the vent shutoff criteria. When the receiver tank is full, flow is terminated by closing valves V_2 and V_5. Supply tank and line venting is then the same as for the high pressure system.

The modular transfer system consists of exchanging complete storage tanks between the shuttle payload and the space station. Configuration and operational details are presented in Section 6. The basic modular tank is shown in Figure 6-8. The tank consists of inner and outer spherical shells, with the evacuated annulus containing high performance insulation.

A reinforcing ring on the outer shell, or vacuum jacket, distributes point loads introduced by the docking mechanism. It is noted that the mechanisms associated with the docking probe and system connections are structurally attached to the outer jacket. This allows the fluid and electrical lines to be thermally isolated from the external environment in a manner similar to that used in conventional dewar design. The probe contains and locates the fluid connection valve described in Figure 6-7. This valve connects to the tank by two concentric CRES bellows. The inner bellows connects to a standpipe which extends to the opposite side of the tank and is used for tank filling and fluid expulsion. The annulus contained by the outer bellows connects the valve to the near side of the tank to provide a path for tank venting and pressurization. The docking receptacle, which is located on both the space station and the shuttle, is shown in Figure 6-9. The tank docked to the space station or shuttle is shown in Figure 6-10.

Assuming adequate volume in the shuttle for receipt of an expended bottle prior to transfer of the full bottle, the following operations are involved.

a. The expended tank, while still on the space station, is vented to a nominal value of 15 to 20 psia.

b. The expended tank is demated from the space station by withdrawing the fluid probe (Figure 6-7) and electrical connector (Figure 6-5), then removing the latch loads by rotating the hook shafts (Figures 6-4 and 6-11) and retracting the hooks.

c. The expended tank is removed from the space station and placed in a "blind" adapter in the shuttle and secured. Tank pressure monitoring is assumed to be required and the tank maintained at 15 to 20 psia by venting as required.

d. The full tank is transferred to and mated to the space station. Mating consists of orientation and lead-in to a triangular docking receptacle (Fig. 6-3), capture with spring loaded hooks (Fig. 6-4) and rigidizing with an eccentric drive motor (Fig. 6-11).

e. The final fluid and electrical connections are then made at the space station by actuation of the appropriate probes (Figs. 6-5 and 6-7).

Comparative weight, reliability, cost and crew performance data for the high pressure, subcritical and modular systems described are presented in the following paragraphs.

7.2 SYSTEM WEIGHTS

The basic information required to generate the comparative weight data presented in this section was obtained from Sections 3 through 6 and Appendix A. In these sections weights were presented primarily as a function of tank size. In the present case, in order to provide a valid comparison between the various systems operating at different design pressures and fluid conditions and having different fluid residuals, the weight data are presented as a function of the amount of useful fluid transferred. Also, weights of lines, valves, pressurization systems and other hardware illustrated in Figures 7-1 through 7-3 are included here.

In the case of high pressure transfer, tank sizes, fluid residuals and power system weights are related to the quantity of fluid transferred by the following equations derived from data contained in paragraph 4.2.8.

$$\text{Receiver Volume, ft}^3 = \frac{H_2 \text{ transferred, lb}}{1.1} \quad (7\text{-}1)$$

$$\text{Supply Volume, ft}^3 = \frac{H_2 \text{ transferred, lb}}{3.23} \quad (7\text{-}2)$$

$$\text{Residual } H_2 \text{ in supply, lb} = 0.21 \,(H_2 \text{ transferred, lb}) \quad (7\text{-}3)$$

$$\text{Total } H_2 \text{ boosted to orbit, lb} = 1.21 \,(H_2 \text{ transferred, lb}) \quad (7\text{-}4)$$

$$\text{Residual } H_2 \text{ in receiver at start of transfer, lb} = 0.136 \,(H_2 \text{ transferred, lb}) \quad (7\text{-}5)$$

$$\text{Power Supply Weight Transported on the shuttle, lb} = 0.115 \,(H_2 \text{ transferred, lb}) \quad (7\text{-}6)$$

Power Supply Weight Located on shuttle or space station, lb $= 10.4$ (H_2 transfer rate, lb/hr) (7-7)

The above data are for H_2 only, since the transfer of LO_2 and LN_2 within reasonable time periods was found to result in excessive weight penalties for the present application. The supply tank pressure is taken to be 225 psia for bottle design and during transfer. The receiver tank initial pressure is 100 psia with a final maximum design pressure of 225 psia.

Basic weights for the subcritical systems are obtained from the information in Section 5.0. Conversion of weights presented in terms of tank size to weights as a function of fluid transferred is based on the following conditions.

Basic tank loadings are to 95 per cent liquid saturated at 20 psia. Corresponding liquid densities for H_2, O_2 and N_2 are respectively 4.34, 70.4 and 49.6 lb/ft^3. Residuals are nominally 2 per cent. Therefore:

$$H_2 \text{ transferred, lb} = 4.04\ (V_t,\ \text{ft}^3) \tag{7-8}$$

$$O_2 \text{ transferred, lb} = 65.4\ (V_t,\ \text{ft}^3) \tag{7-9}$$

$$N_2 \text{ transferred, lb} = 46.2\ (V_t,\ \text{ft}^3) \tag{7-10}$$

The above relationships were also applied to the modular transfer cases.

Major elements associated with the subcritical systems for which complete data were not previously generated are the pressurization subsystem and the transfer lines. Specific information on these items used for the current comparisons is developed in the following paragraphs.

7.2.1 PRESSURIZATION SYSTEM PARAMETRIC DATA. The basic equations presented in Section 5.2 are used. Initial analyses were accomplished for the pressurization of spherical hydrogen tanks containing a double screen liner. Aluminum tanks and screens were assumed. Tank weight data are obtained from Figure A-1 and screen weights from Figure 5-35. The sum of these weights are used in Equation 5-17a to calculate the constant (C). Values for the collapse factor (CF) were then determined from Equation 5-16 and are plotted in Figure 7-4 as a function of transfer time for various bottle sizes. The conditions used are presented on the figure.

Data for O_2 systems are presented in Figure 7-5 showing significantly lower values for the collapse factor than for the H_2.

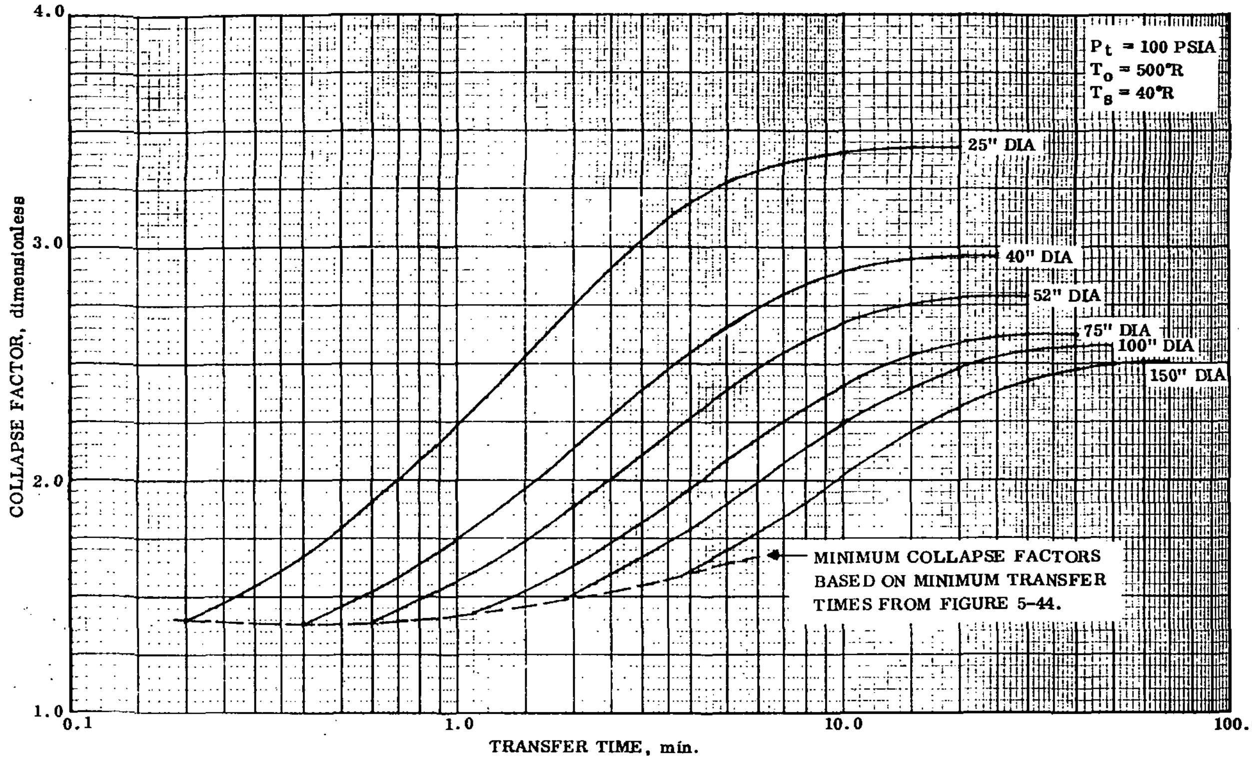


Figure 7-4. H_2 System Collapse Factor as a Function of Time for Surface Tension System

The following equations were used for oxygen,

$$C = 0.157 \frac{W_{TOTAL,\ LB}}{V_t\ ,\ ft^3} \tag{7-11}$$

$$S = 9.5\ \frac{(\theta,\ min.)}{(D_t,\ in.)} \tag{7-12}$$

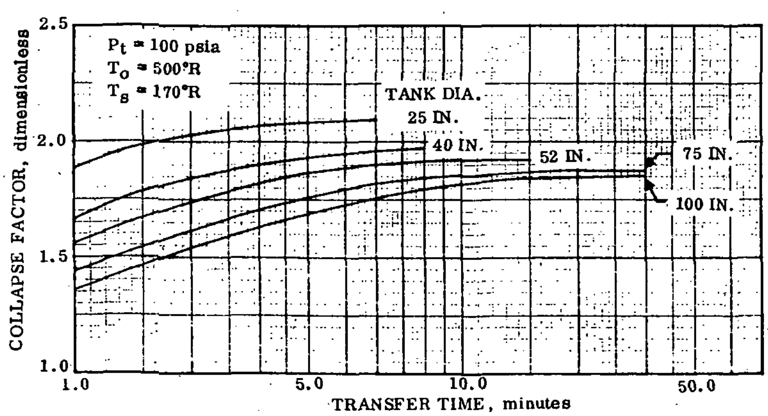


Figure 7-5. O_2 System Collapse Factor as a Function of Time for Surface Tension System

It is noted that a significant range of the collapse factor occurs for different transfer times. This is especially true in the case of hydrogen. For the hydrogen transfer, minimum transfer times shown are based on use of the highest discharge flow rate throughout the transfer which will still allow minimum residuals. These transfer times are taken from Figure 5-44.

The maximum collapse factors occur at transfer times where the value of S in Equation 5-16 has the effect of approaching infinity. In this case Equation 5-16 reduces to,

$$CF = \left[\left(\frac{T^o}{T_s} - 1\right)\left(1 - e^{-P_1 C^{P_2}}\right) + 1\right] \qquad (7\text{-}13)$$

Further examination of Equation 5-16 shows that as the transfer time approaches zero ($S \rightarrow O$) a lower limit of the collapse factor of 1.0 is approached.

The range of data presented in Figures 7-4 and 7-5 would also occur for fixed transfer times with changes in the heat transfer coefficient (h_f). In order to allow for the possibility of fluid sloshing and high heat transfer between the pressurant and surroundings, conservative values for the collapse factor, as represented by Equation 7-13 and the maximums from Figures 7-4 and 7-5 were used in development of the parametric weight data contained in this section.

Using the procedures outlined in Section 5.2 and maximum collapse factors from Equation 7-13 and Figures 7-4 and 7-5, the weight data presented in Figure 7-6 was obtained. Weights presented are the sum of bottle plus total helium. In developing the data the following conditions were assumed

$\rho_{He} = 0.073$ lb/ft^3 at 100 psia and 500°R, such that, He required, lb = .073 (CF) (V_t, ft^3). For the He storage bottle, $\rho_i = 2.15$ lb/ft^3, $P_i = 3300$ psia, and $T_i = 500$°R. Then from Equation 5-20 for $n = 1.4$ and $P_f = 125$ psia, $T_f = 196$°R and $\rho_f = 0.24$ lb/ft^3. Equation 5-19 then determines the helium storage volume required and bottle weights are obtained from Appendix A.

Pressurization system weight data were then generated for the paddle system described in Section 5.7 using the same procedure as above. The only difference from the surface tension system is the weight of the paddle hardware as compared to the screens. The calculations showed a negligible difference between helium system weights of the two systems and thus were taken to be the same for the two cases. Pressurization system requirements for the bellows and diaphragm systems would be affected by additional hardware in contact with the pressurant and by reduced contact between the pressurant and fluid. The magnitude of these effects can not be accurately determined within the present state-of-the-art and for purposes of the present comparisons it was assumed that these effects would be offsetting and the pressurization system weights presented in Figure 7-6 were used for all the subcritical systems.

7.2.2 TRANSFER LINES. Representative transfer line sizes are determined from the following equations

$$\Delta P = \frac{f\, L_\ell}{D_\ell}\, \rho_L\, \frac{\left(V_{e_L}\right)^2}{2\, g_c} \tag{7-14}$$

$$\dot{m}_L = \rho_L\, \frac{\pi\, D_\ell^2}{4}\, V_{e_L} \tag{7-15}$$

Combining Equations 7-14 and 7-15 and solving for line diameter (D_ℓ) results in,

$$D_\ell = 0.959 \left(\frac{f\; L_\ell\; \dot{m}_L^2}{\Delta P\; g_c\; \rho_L} \right)^{1/5} \tag{7-16}$$

Assuming LH_2 transfer with $\rho_L = 4.26\ \mathrm{lb/ft^3}$, $g_c = 32.2\ \mathrm{ft/sec^2}$. $f = 0.015$ for a smooth line, and an allowable pressure drop (ΔP) between supply and receiver of 5 psi,

$$D_\ell,\ \mathrm{in.} = 0.0967\ (L_\ell,\ \mathrm{ft})^{0.2} \cdot (\dot{m}_L,\ \mathrm{lb/min.})^{0.4} \tag{7-17}$$

From Figure 7-1 the maximum distance from the supply to any one receiver is 91 ft. Allowing a 10% increase in the effective line length, to account for bends, valves and fittings results in an L_ℓ of 100 ft. for use in Equation 7-17; or,

$$D_\ell,\ \mathrm{in.} = 0.242 \cdot (\dot{m}_L,\ \mathrm{lb/min.})^{0.4} \tag{7-18}$$

To obtain representative line sizes as a function of mass transferred, flow rates for use in Equation 7-18 were determined by dividing the maximum values from Figure 5-44 by a factor of ten. Conversion from volume rate to mass rate is based on a H_2 density of $4.34\ \mathrm{lb/ft^3}$.

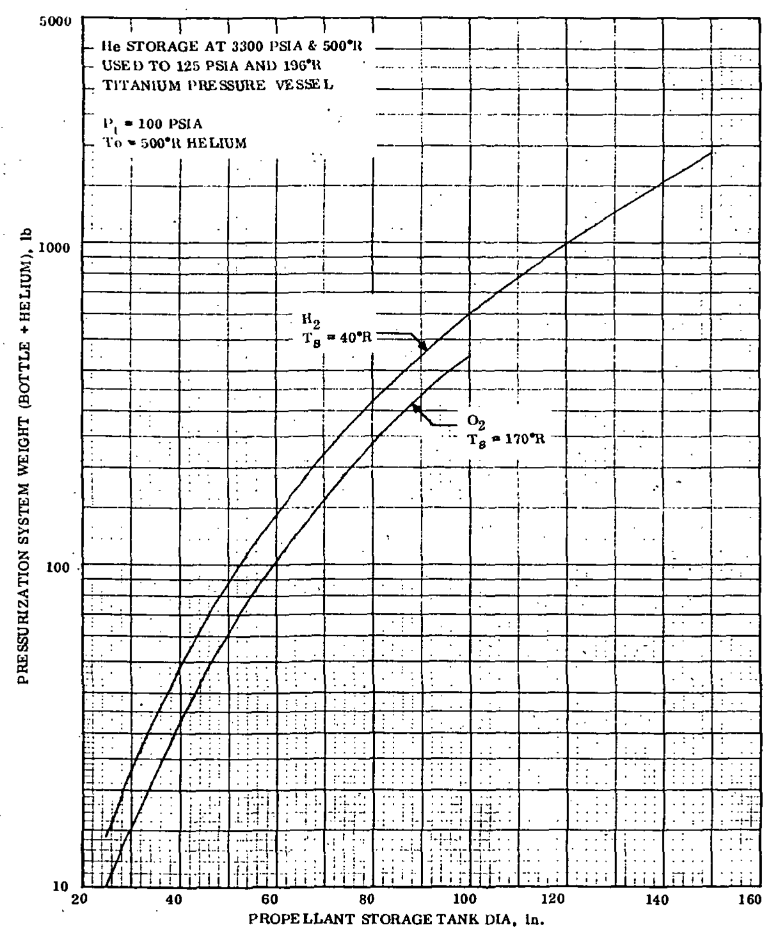


Figure 7-6. Pressurization System Weight for Surface Tension System

Line sizes obtained in this manner are presented in Figure 7-7. Corresponding CRES line weights for total lengths of 30 feet and 100 feet, as estimated to be on the shuttle and station respectively, were then determined from the data of Appendix A. These weights are presented in Figure 7-7.

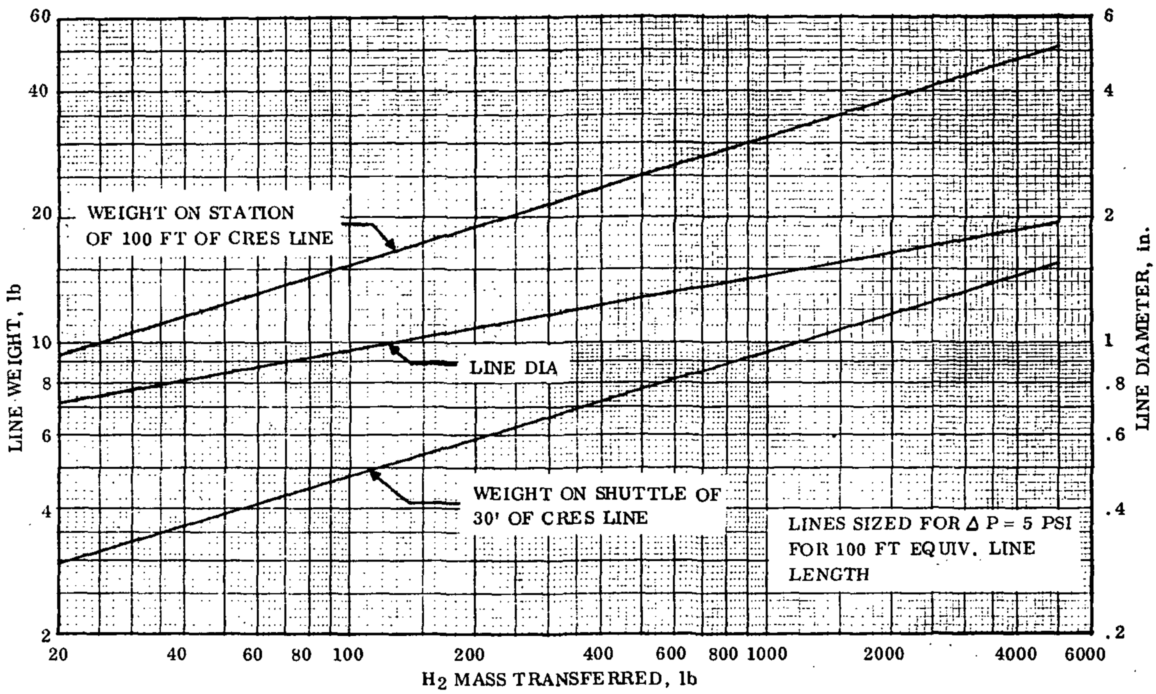


Figure 7-7. Typical Line Sizes and Weights as Function of H_2 Mass Transferred

In the case of O_2 transfer it was determined that use of the same line sizes as for the H_2 would result in an equivalent volume transfer when the transfer time is four times longer than for the H_2. Based on the analyses of Section 5.4, such a ratio would be desirable in order to obtain minimum fluid residuals for the O_2 as compared to the H_2.

7.2.3 WEIGHT SUMMARY. Comparative data for the high pressure, surface tension, bellows, diaphragm, paddle, and modular supply systems are presented in Figure 7-8. These weights represent that boosted into orbit for each resupply mission.

The high pressure supply weights include the inner tank, insulation, 30 ft. of transfer line, one shut-off valve, one vent valve, power supply per Equation 7-6, shuttle half of disconnect, electric heater and control, vacuum jacket and the total fluid contained.

Inner tank weights are obtained from Figure A-1 for a tank design pressure of 225 psia. Insulation weights are based on the use of "Superfloc" HPI and the data and equations presented in Section 5.1. Locked-up storage prior to transfer without mixing with an allowable pressure rise from 20 psia to 225 psia over a seven-day period is assumed.

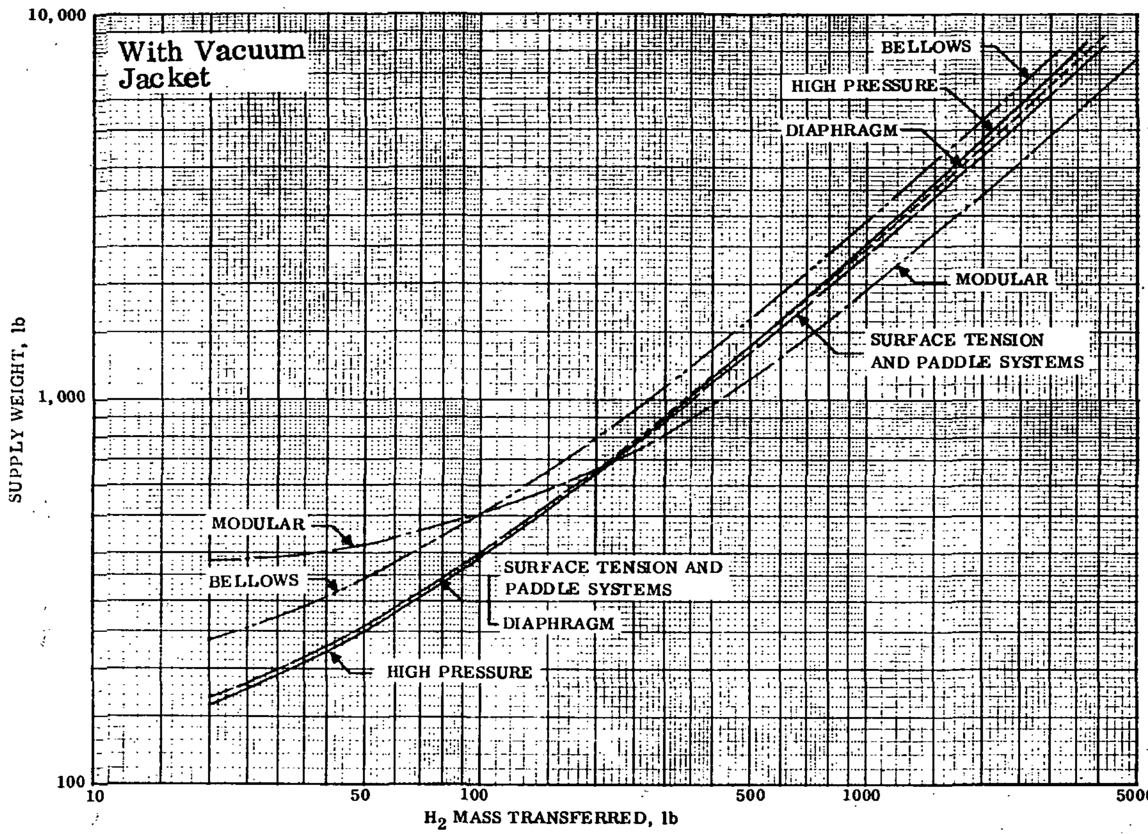


Figure 7-8. H_2 Supply System Weights Including Fluid (Wet)

Transfer line sizes and weights are determined to be the same as for the subcritical systems from Figure 7-7. Valve and vacuum jacket weights are obtained from Appendix A. The jacket diameter is taken to be the larger of that required for the insulation and the minimum gap per Figure 5-1.

Subcritical supply weights include the same basic tankage and transfer line hardware as the high pressure system. The pressurization system with associated valving and the liquid orientation device is included instead of the power supply and electrical heater.

The sum of inner tank weight, insulation and vacuum jacket are obtained for the surface tension, diaphragm, and paddle systems from Figures 5-2, 5-3 and 5-4 at the optimum ullage conditions. In the case of the 52-inch diameter tank the curve for minimum gap spacing was used. Surface tension screen weights were determined from Figure 5-35, and diaphragm weights from Figure 5-60 assuming the use of hollow wires. Paddle hardware and power supply weights were obtained from Figures 5-64 through 5-67.

Bellows plus inner tank weights are from Figure 5-54. Insulation thicknesses for the bellows system are determined from Equation 5-11 where the allowable heat leak ($\dot{Q}_a = \dot{Q}_i$),

($\dot{Q}_a = \dot{Q}_i$), effective insulation conductivity and environmental temperature are taken to be the same as for the other subcritical systems. Tank surface area is obtained from Figure 5-58. Vacuum jacket weight is from Appendix A for a cylinder, with the diameter based on insulation requirements or spacing required for struts from Figure 5-1, whichever is larger. The double bulkhead concept illustrated in Figure 5-56 was not used.

For all subcritical systems, pressurization weights are taken from Figure 7-6 with additions made for lines and associated valving.

Modular weights for the inner tank and vacuum jacket are taken from Appendix A with an inner tank design pressure of 100 psi. Insulation thicknesses and weights are from Figure 6-12 and 6-13. The modular weights presented do not include the mechanisms required to transfer the bottles from the shuttle to the station. Docking and associated hardware weights mounted on the bottle and in the shuttle are included as obtained from Equation 6-3.

Figure 7-8 shows that no system is lightest or heaviest over the full range of transfer quantities. Of the subcritical systems the bellows is significantly heavier than the others, with the surface tension and paddle systems essentially equal and representing the lowest weights. The diaphragm weight is also low at small supply tank volumes and gets relatively heavier at larger sizes or transfer quantities. The high pressure system has the lowest weight at small transfer quantities with a significant increase at larger values.

Weights of the modular system are relatively high at low transfer masses and low at the higher masses. In the modular case a weight penalty results from the requirement to insulate the supply tank for long term storage on the space station. However, transfer lines and pressurization system are not required. Insulation requirements are magnified at small tank sizes and at larger sizes pressurization system weights required for subcritical systems becomes an increasingly significant factor.

It is noted that in each case a single bottle is assumed to accomplish the mass transfer shown in Figure 7-8. Based on the data presented in this figure, the surface tension and paddle systems are the most desirable subcritical systems; further comparisons are made between these and the modular and high pressure system.

Weights without a vacuum jacket for these systems along with the bellows system are presented in Figure 7-9. Essentially the same relationship was found to exist between the systems as in Figure 7-8 where vacuum jacketing was included. The only change was a slight improvement in relative weights of the bellows and modular systems at low transfer quantities. The surface tension and paddle systems still represent the lowest weight subcritical systems.

For assessing the relative weights of the expended supply being returned to earth,

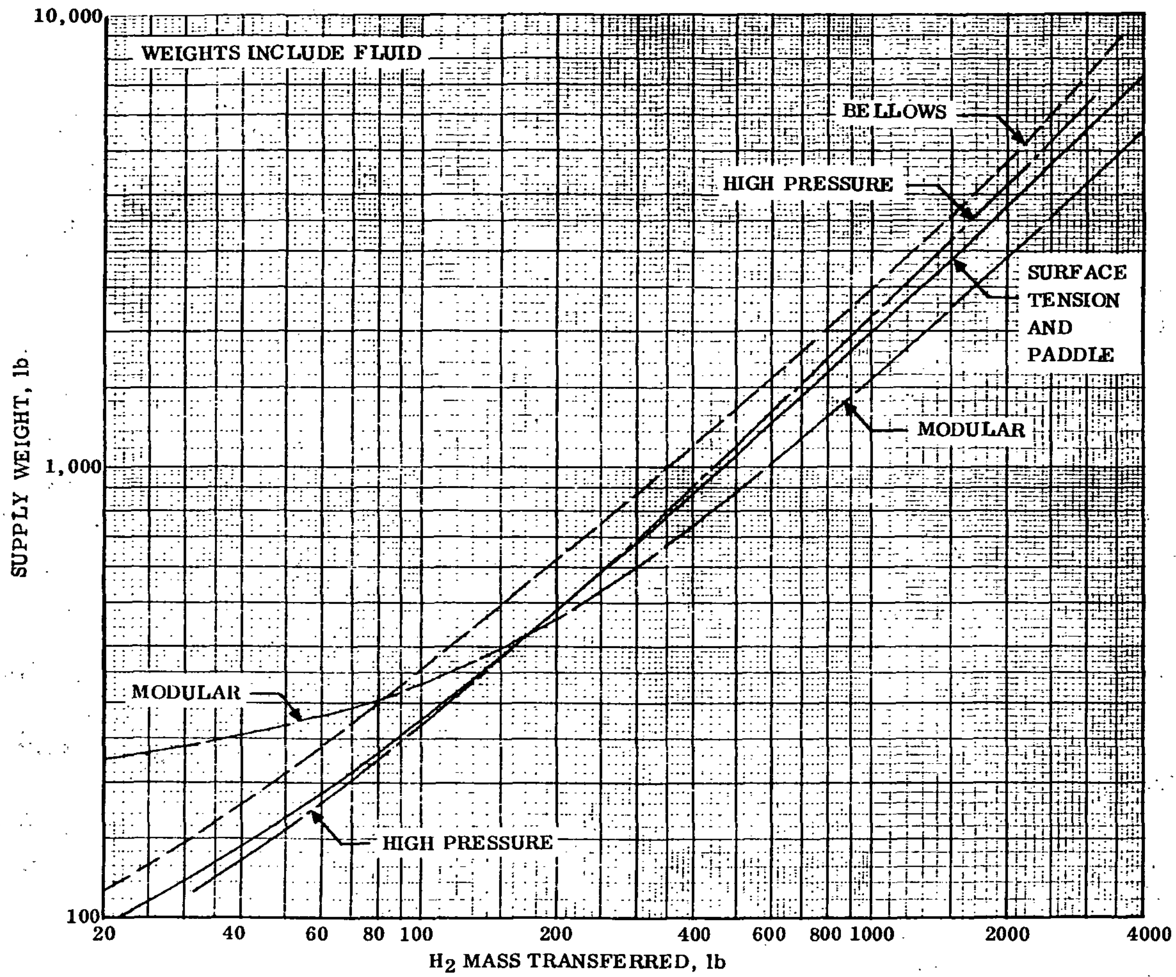


Figure 7-9. H_2 System Supply Weights Without Vacuum Jacket (Wet)

weights without fluid are presented in Figure 7-10 for the high pressure, subcritical and modular systems. The relation between modular and subcritical systems remains the same while the high pressure system shows a relative weight reduction. This weight reduction shows the result of increased fluid residuals for the high pressure system over the other systems.

Receiver weights for the high pressure, subcritical and modular systems are presented in Figure 7-11. High pressure weights include the inner tank from Figure A-1 at 225 psia, insulation from Figure 6-13, and 100 feet of transfer line per Figure 7-7. Associated valve weights are based on the Figure 7-2 transfer configuration. Weights do not include the power supply which is shown separately in Figure 7-12 as a function of total transfer time.

Subcritical system weights for lines and valving are the same as for the high pressure system, except that a vent valve and liquid/vapor separator are included for the subcritical system as shown in Figure 7-3. Weight estimates for this hardware are based on data from Reference 5-5.

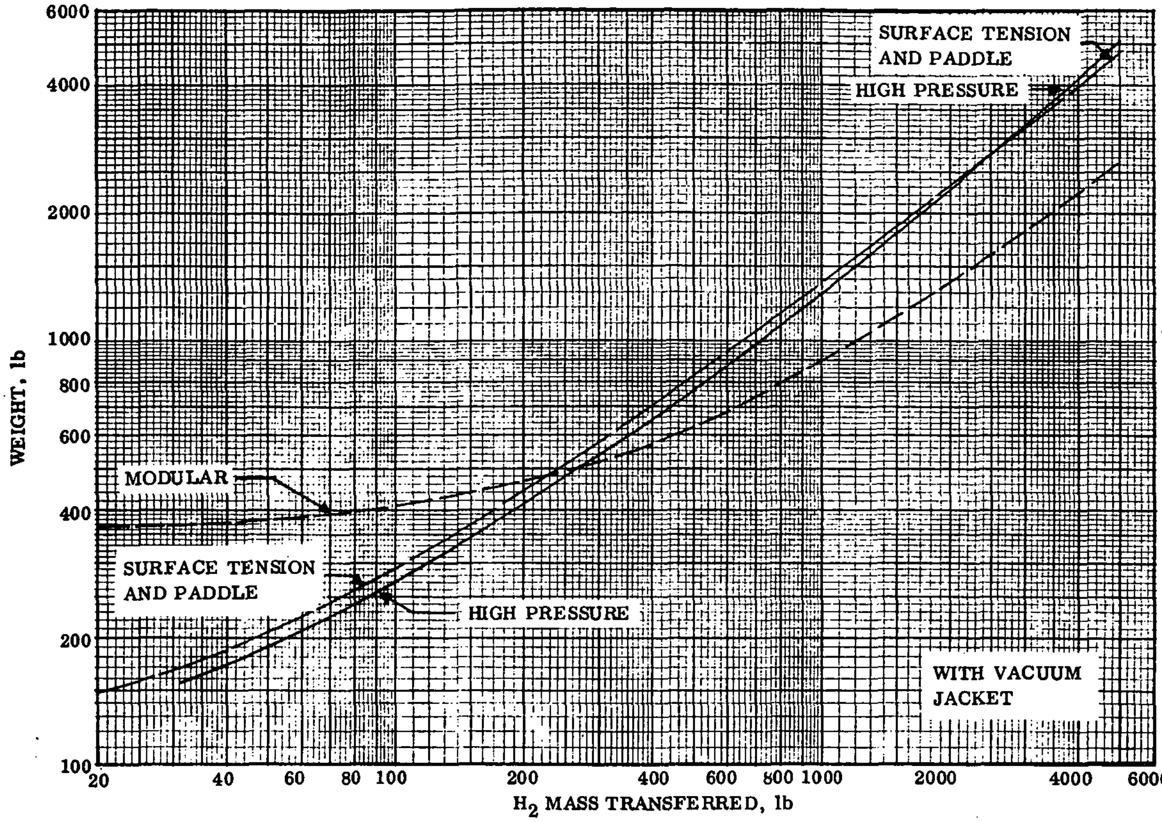


Figure 7-10. H_2 System Supply Weights Without Fluid (Dry)

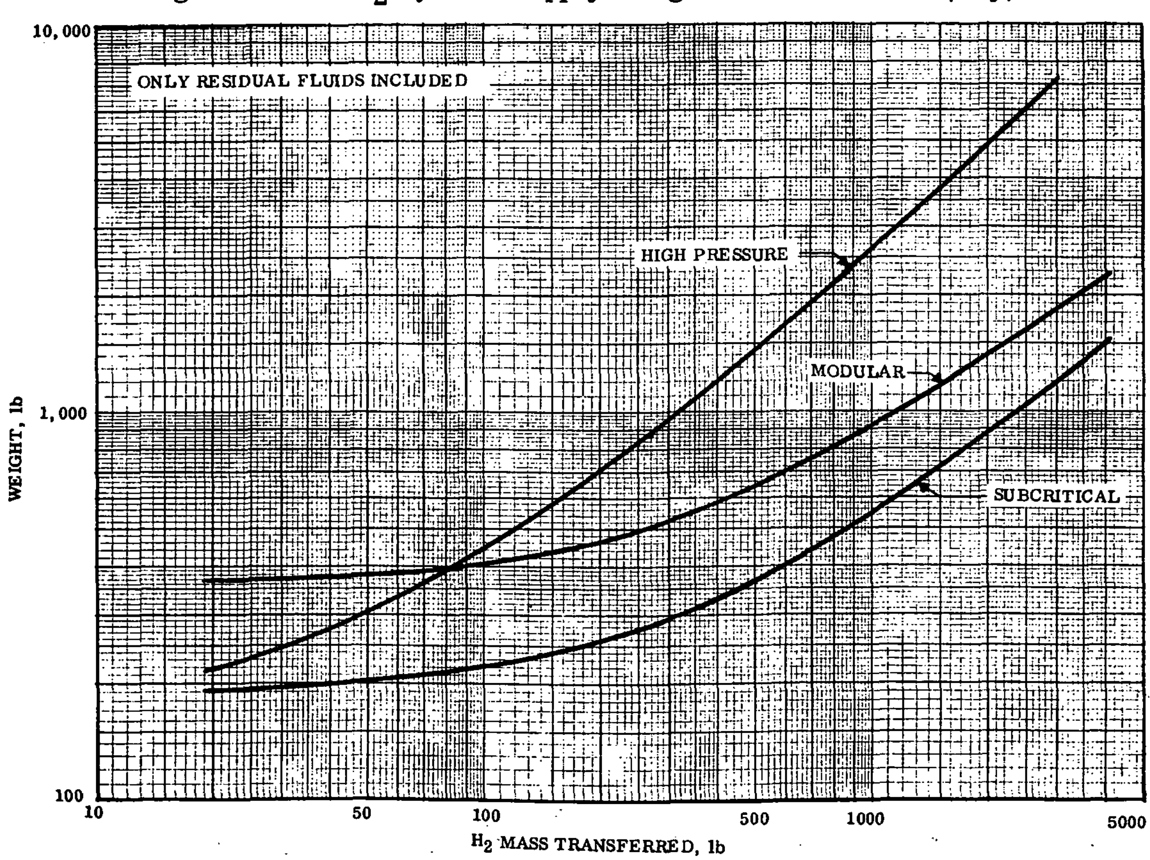


Figure 7-11. Receiver Weights for H_2 Transfer

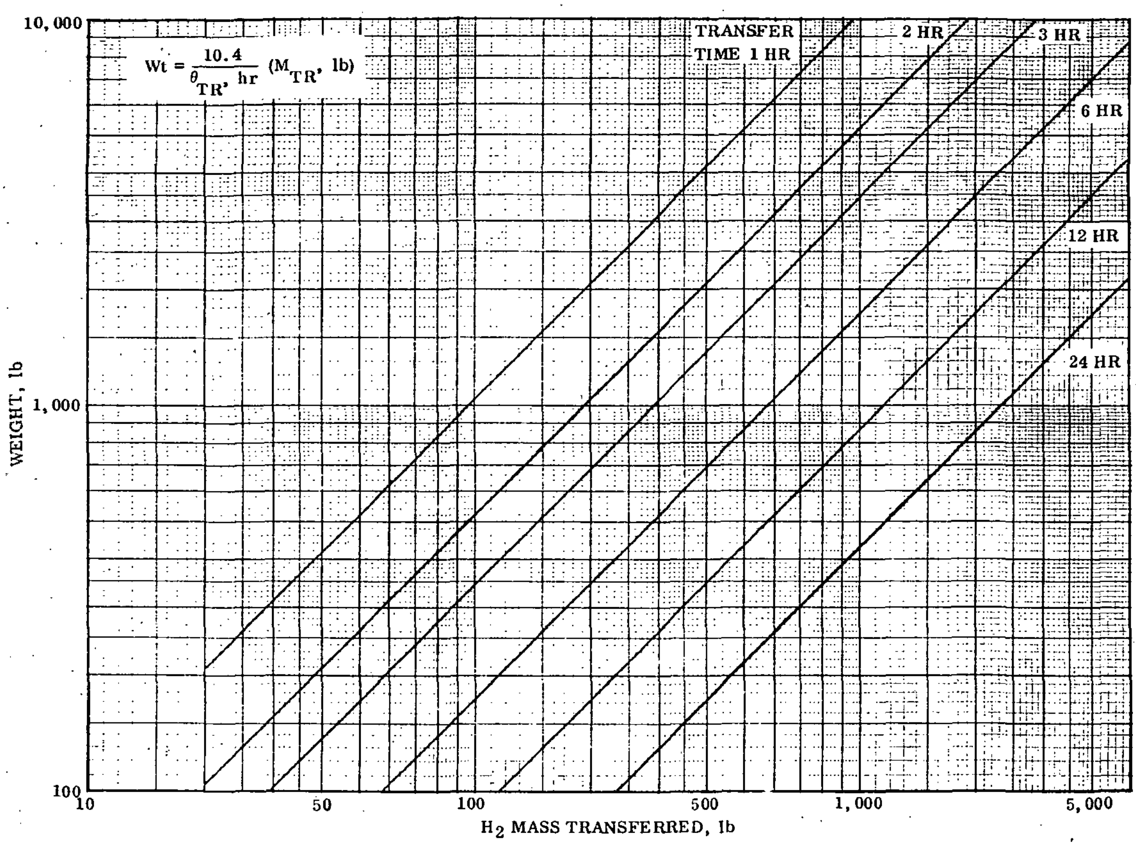


Figure 7-12. High Pressure System Power Supply Weights Based on Usage Rate

Modular system receiver weights are the same as for the modular supply system, except that docking hardware are obtained from Equation 6-4 rather than Equation 6-3.

The weights shown in Figure 7-11 include only the residual fluid. Residuals are based on Equation 7-5 for the high pressure system and 2 per cent of tank volume for the other systems. Vacuum jacketing is included for the modular system but not for the other systems.

Figure 7-11 shows that, except at low transfer quantities, the high pressure system is significantly heavier than the other systems, while the subcritical system has the lowest weight over the full range.

In order to obtain an overall comparison of the various systems the data from Figures 7-8, 7-11 and 7-12 are combined and presented in Figure 7-13. In this case 20 resupplies were assumed over the life of the station and hardware on the station was assessed at only one-twentieth of that on the shuttle.

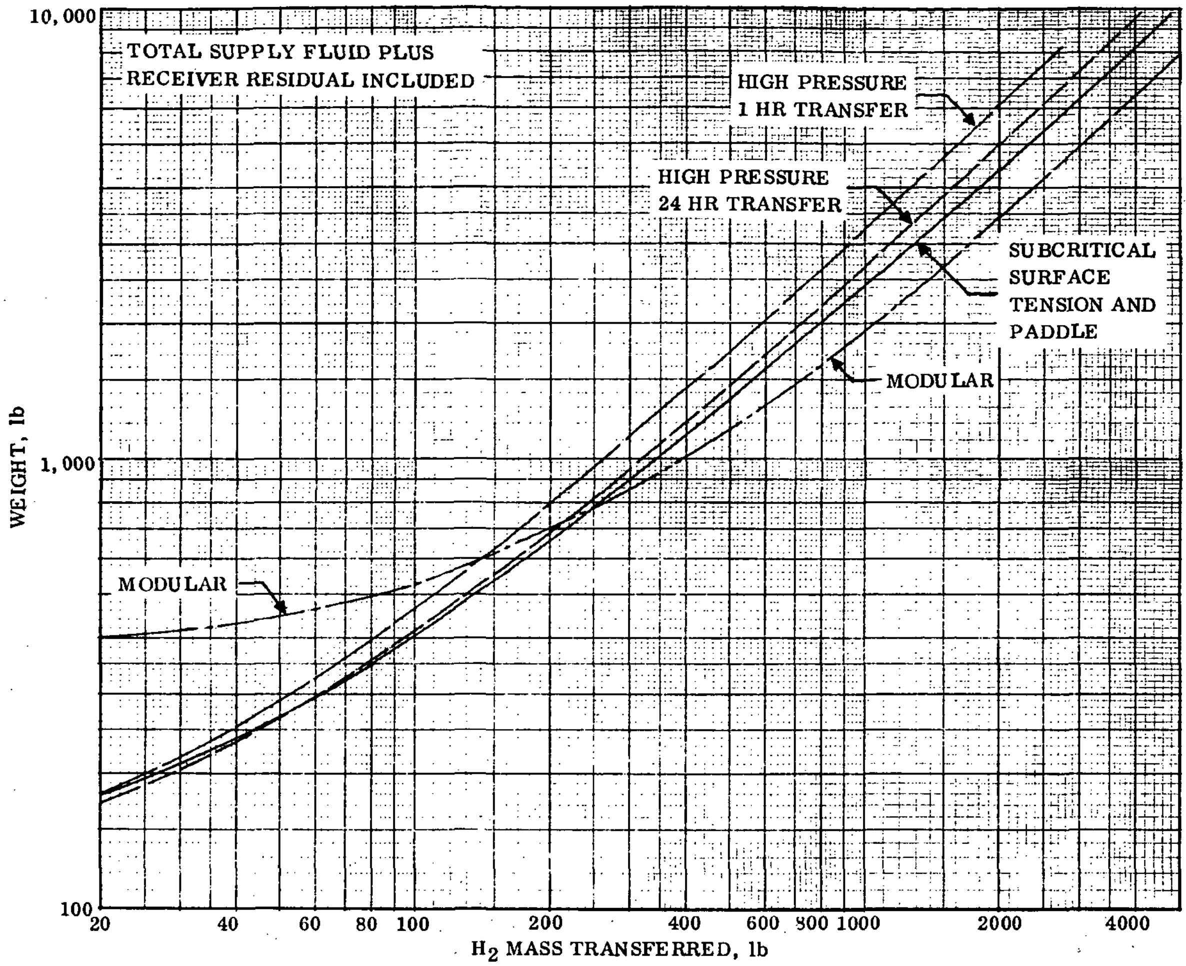


Figure 7-13. Total H_2 Transfer System Weight
(Supply + Receiver/20)

On this basis comparison of Figures 7-8 and 7-13 shows essentially the same relation between modular and subcritical system weights for the supply-only as for the overall system. In the case of the high pressure system, inclusion of the receiver and associated power supplies significantly increased the relative weights of this system. The effect of transfer time is also presented showing a significant increase in high pressure system weight for transfer times approaching those of the subcritical systems.

Supply weights for LO_2 transfer are presented in Figure 7-14. Weight data were obtained in a similar manner as described for the H_2 systems. The main differences in the weights from those for H_2 are due to increased power required for the paddle system and reduced insulation for all systems.

Nitrogen supply system data are presented in Figure 7-15. In comparison with the LO_2 data, the weights are affected by slightly reduced power requirements for the paddle system and less stringent insulation requirements for the modular system. Otherwise, the basic hardware weights are the same for LN_2 as for LO_2 for the same size tank.

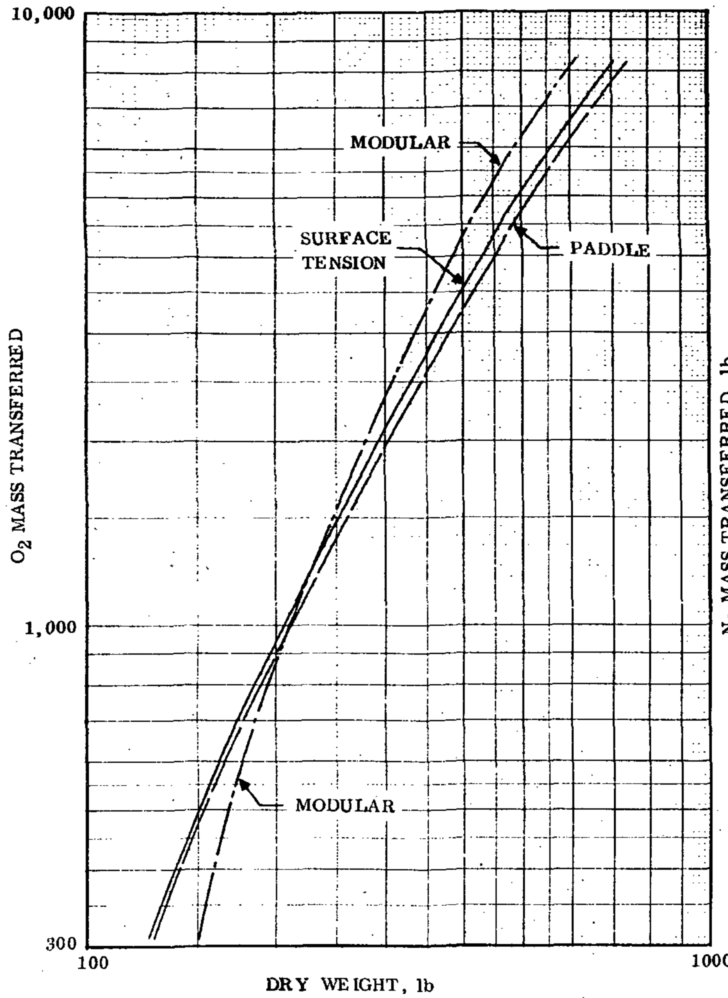


Figure 7-14. O_2 Supply System Weights Without Fluid (Dry)

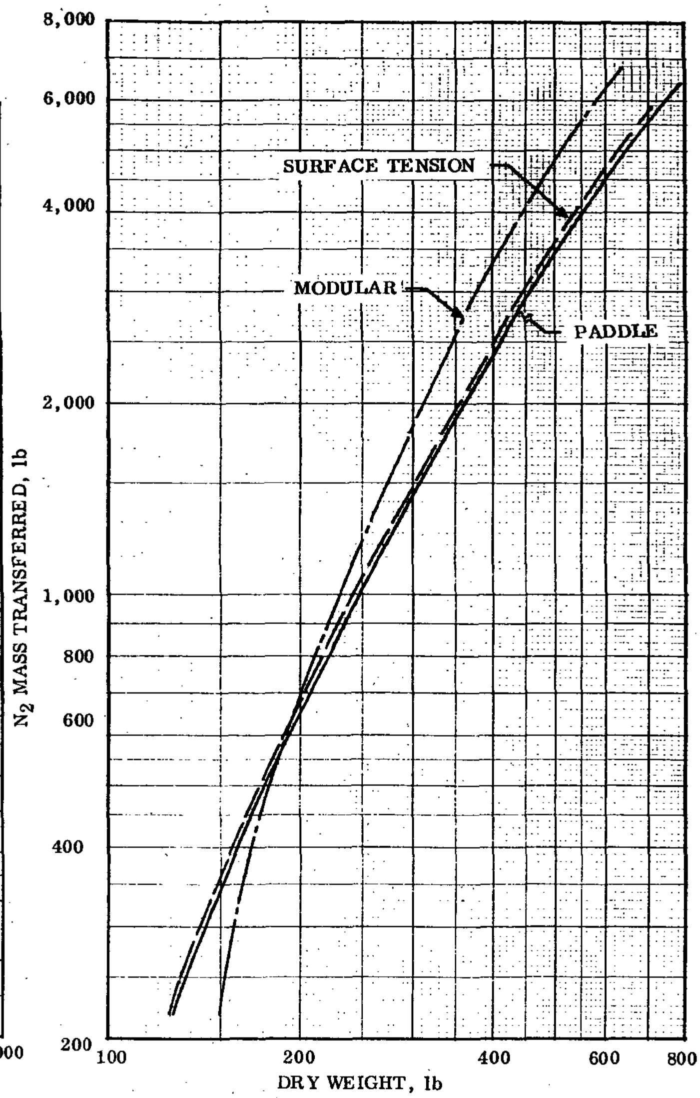


Figure 7-15. N_2 Supply System Weights Without Fluid (Dry)

Insulation requirements for the modular system are determined from Equations 6-1 and 6-2, except for the N_2 case, 100 percent rather than 50 percent boiloff is allowed over a 180-day period. On this basis required insulation thicknesses for N_2 are determined to be 80 percent of those for O_2.

Using the data from Figures 7-10, 7-14 and 7-15, comparisons were made between the surface tension, paddle and modular systems for the specific application where 1096 lb of H_2, 2480 lb of O_2 and 3150 lb of N_2 are transferred to eight H_2, two O_2 and two N_2 bottles located on a space station.

Results are presented in Table 7-1 for two supply cases: (1) individual supply tanks for each receiver tank and (2) a single supply tank for each fluid.

The Table 7-1 data show a significant savings by having a single supply bottle for each fluid rather than individual units. The modular system is significantly heavier than the subcritical systems when compared with single tanks. In all cases the surface tension and paddle systems have the lowest weights. Weight differences between these two subcritical systems are small.

Weight data, assuming the use of common tankage for the individual tank case to reduce the costs per tank, are presented in Table 7-2. A tank size of 42.5 ft^3 was used per the baseline system described in Section 2. It is seen that there is a 19 percent increase in weight for the subcritical systems and 9 percent for the modular system.

7.3 RELIABILITY ANALYSIS

Failure rate predictions were performed for each of the fluid transfer concepts described in Section 7.1. Component failure rates were multiplied by their respective stress times, number of tanks, if applicable, and system lifetime factors to obtain expected numbers of failures. The sum of component-expected failures for each concept is the total expected number of failures for that concept. Data are presented in Table 7-3 for system lifetimes of four missions per year over ten years for each transfer concept.

The boost and rendezvous and the fluid transfer phases of the system mission were considered in this analysis. Rocket burn times totalling a half-hour were assumed for boost and rendezvous, during which the fluid supply equipment aboard the orbiter would be under stress. Fluid receiving equipment aboard the orbiting station would not be affected by the boost and rendezvous phase. The fluid transfer phase is considered to begin with initiation of fluid system hookup or, in the case of the modular transfer concept, of de-mating the first expended tanks from the orbiting station. Component stress (operating) times for receiver hardware are based on a one-time boost.

Shutoff valves aboard the orbiter must operate only on initiation of fluid transfer and the success of this operation depends only on their survival during the boost and rendezvous phase. No likely failure mode in orbital environment would preclude successful fluid transfer, i.e., if a valve failed to close on completion of transfer, venting of the supply tank would still permit decoupling. These factors were accounted for in the analysis.

Component failure rates were extracted from the Data Collection for Nonelectronic Reliability Handbook (NEDCO), RADC-TR-68-114, Rome Air Development Center, June 1968. Failure rates of components used in similar applications were taken. Since the purpose of this analysis was only a basic comparison of concepts, no attempt was made to optimize system reliabilities nor to provide for redundancy. The prime concern was for consistency rather than accurate determination of what

Table 7-1. Transfer System Weights (Dry) to Supply H_2, O_2 and N_2 Aboard a Space Station

Supply Configuration	Supply System Weights, lb		
	Surface Tension	Paddle	Modular
Individual Tanks			
LH_2 (8 Tanks)	2800	2800	3448
LO_2 (2 Tanks)	456	474	456
LN_2 (2 Tanks)	620	630	560
Total	3876	3904	4464
Single Tank			
LH_2	1470	1470	-
LO_2	329	342	-
LN_2	463	474	-
Total	2262	2286	-

Table 7-2. Transfer System Weights (Dry) Using Common Tank Sizes)

Fluid	Supply System Weights, lb		
	Surface Tension	Paddle	Modular
LH_2 (8 Tanks)	3200	3200	3600
LO_2 (2 Tanks)	700	730	648
LN_2 (2 Tanks)	700	712	616
Total	4600	4642	4864

Table 7-3. Reliability Comparison Data (Lowest Number Best)

Transfer System	Single Tank to Tank Transfer	Tank to Tank for 12 Supply and 12 Receiver Tanks	Single Supply for Each Fluid (3 Supply and 12 Receivers)
Modular			
Transfer Mechanism Aboard Shuttle	0.181483	1.033796	-
Transfer Mechanism Aboard Station	0.123883	0.870598	-
High Pressure	0.101120	1.090240	0.341440
Subcritical			
Bellows	0.179704	1.998048	0.540408
Surface Tension	0.114704	1.218048	0.325248
Paddle Vortex	0.146598	1.600800	0.422880
Metallic Diaphragm	0.113440	1.202880	0.324480

is feasible. Also, items common to all schemes, such as fixed electrical connections, were omitted. Where possible, ground failure rates were used directly for the fluid transfer phase. If only an aircraft failure rate could be found for a similar application, it was divided by 6.5 to convert it to an orbital environment. Orbital rates were multiplied by 80 to convert them to rates during burn of rocket engines.

The numbers in the Table 7-3 column called "Single Tank to Tank Transfer" are the expected number of failures during the stated mission phases to replenish or replace just one tank, accumulated over the assumed system lifetime. The next column to the right contains the expected number of failures for replenishing or replacing all the assumed 12 tanks aboard the station on a one-for-one basis. For the modular transfer scheme this merely entails exchanging 12 full for 12 empty tanks. The data in the third column assumes that there is just one supply tank per fluid, or a total of three supply tanks. This condition is not applicable to the modular transfer scheme because of the one-for-one replacement requirement. Also, Table 7-3 modular data are shown for two cases of transfer mechanism locations: (1) installed on the orbiter, and (2) aboard the station. The difference in expected failures reflects the effects of repeated boost and rendezvous stresses on that item, if installed in the orbiter.

From examination of the Table 7-3 data, the modular transfer system is the most reliable if the system requires a tank-for-tank replenishment. The reliability advantage is due to not requiring a mechanism for moving fluid from one tank to another in orbit. For the systems with one tank for each fluid, the metallic diaphragm and capillary control schemes have the highest reliability due to their inherent simplicity. It is noted, however, that the metallic diaphragm system will not meet the life requirement without periodic replacement.

7.4 COST ANALYSIS

Based on information obtained from hardware vendors and Convair Aerospace inhouse estimates, relative costs were generated for the systems described in Section 7-1. Valving, pressurization and line data were included; however, the tankage, including insulation and vacuum jacketing, was found to be the highest cost item.

Costs of mechanisms required to transfer the modular bottles from the shuttle to the station were not included. The data are presented in Table 7-4 for H_2 systems only. All costs are related to that of the surface tension system where 8 supply and 8 receiver tanks are used. The basic tank size was 42.5 ft^3 except for the high pressure receivers which were 122 ft^3. For the single supply case, the supply tanks are 340 ft^3. Fluid costs were not included.

The diaphragm system is assumed to be replaced four times during the 20 missions. All other systems are used for the full 20 missions without replacement or refurbishment. Single pressurization and power supply systems are assumed for the supply tanks

Table 7-4. Relative Costs of H_2 Transfer Systems

Transfer System	Single Tank to Tank Transfer (1 mission)	Tank to Tank Transfer with 8 Supply and 8 Receivers (20 missions)	Single Supply and 8 Receivers (20 missions)	Single Supply and 2 Receivers (20 missions)
High Press.	0.181	1.013	.591	0.301
Surface Tension	0.141	1.0	.523	0.262
Bellows	0.193	1.421	.617	0.356
Diaphragm	0.139	1.655*	.671*	0.411*
Paddle	0.139	0.989	.472	0.242
Modular	0.195	1.559	---	0.486**

* Diaphragm assumed to be replaced 4 times over a period of 20 missions.

** Modular costs based on 2 supply and 2 receivers.

and are thus not significantly affected by the number of supply bottles. This is the primary reason that, except for the modular system, the cost of tank to tank transfer with 8 supply and 8 receivers is not eight times the cost of a single tank to tank transfer.

With a single tank to tank transfer for one mission the surface tension, diaphragm and paddle systems have the lowest costs. In the case of tank to tank transfer with 8 supply and 8 receivers for 20 missions the high pressure, surface tension and paddle systems are the lowest cost. The diaphragm system cost increases significantly in relation to the one mission case due to the requirement for periodic replacement over 20 missions. The relative cost of the high pressure system is reduced because of the high costs associated with the power supply which are independent of the number of supply bottles for a given transferred fluid quantity and overall transfer time.

In all cases costs were reduced significantly by using a single supply where possible. Additional cost savings are shown in Table 7-4 where a single supply tank and two receiver tanks are used. Two receiver tanks were chosen as the minimum to allow separation of space station storage for safety. In this case the receiver tanks are 167 ft^3, except for the high pressure system which is 500 ft^3.

7.5 CREW PERFORMANCE EVALUATION

This section discusses the effect that the low gravity propellant transfer systems described in Section 7.1 have on the crew in terms of potential hazards, time and energy requirements, and skill and training requirements. The impact of selected study ground rules and assumptions is presented first, followed by a description of the potential cargo transfer device for the space shuttle system and the crew's role in its operation. Then the differences in operational procedures of the three primary transfer modes are highlighted in functional flow diagram form, which leads into the final discussion and summary of the crew performance comparisons between the proposed systems.

Study Ground Rules and Assumptions. Two of the study ground rules from Section 2.0 are: (1) "Crew tasks will be minimized and crew safety is a prime consideration," and (2) "The space station bottles are in unpressurized areas. . ." The impact of these ground rules is to essentially rule out manual transfer of the propellant tanks for the proposed modular transfer system. The latter ground rule requires EVA pressure-suited operations which, in conjunction with a manual transfer mode, would maximize crew tasks, time and energy expenditures.

It is noted that even without the constraints of the above ground rules, man's capability to manually transfer the proposed bottles in the low gravity environment is unknown. Ground-based and aircraft studies using neutralized-gravity simulators report mass handling limits (under IVA conditions) varying from 100-150 pounds (Ref. 7-1) to in excess of 1600 pounds (Ref. 7-2). The actual limits await space environmental experimentation.

The effect of ruling out manual transfer is to assume that some type of device will be available for the modular transfer mode. The most likely candidate to fulfill the requirements of such a device is the remote manipulator, currently under study by NASA. Its significant characteristics of interest to this study are discussed in the following section.

Potential Remote Manipulator System. A typical candidate for the transfer device required by the modular transfer mode is the remote manipulator described in Reference 7-3. Its pertinent characteristics are identified in Table 7-5. The tip force of 10 pounds and the tip positional accuracy of 2 inches are sufficient to accomplish the docking required for tank transfer described in Section 6.0. Also its load carrying capability, although not firmly established at this time, would span that required for the transfer operation.

A typical timeline for unloading and deploying a payload is illustrated in Figure 7-16. Based on this timeline and assuming that the same time is required for transporting empty as well as full bottles, the total time to transfer the 12 baseline H_2, O_2 and N_2 bottles is 240 minutes (10 minutes per transfer operation) or 4 hours. Thus the total transfer could be accomplished well within the 24-hour ground rule limit.

Use of a remote manipulator has the effect of reducing any differences between the proposed transfer systems in terms of hazard potential. The crew for the modular transfer mode would be remotely located in a control center, such as the space shuttle cabin or on the space station just as they could be for the high pressure and subcritical transfer modes, and the hazard potential of one system is thus similar to the hazard potential of the other two.

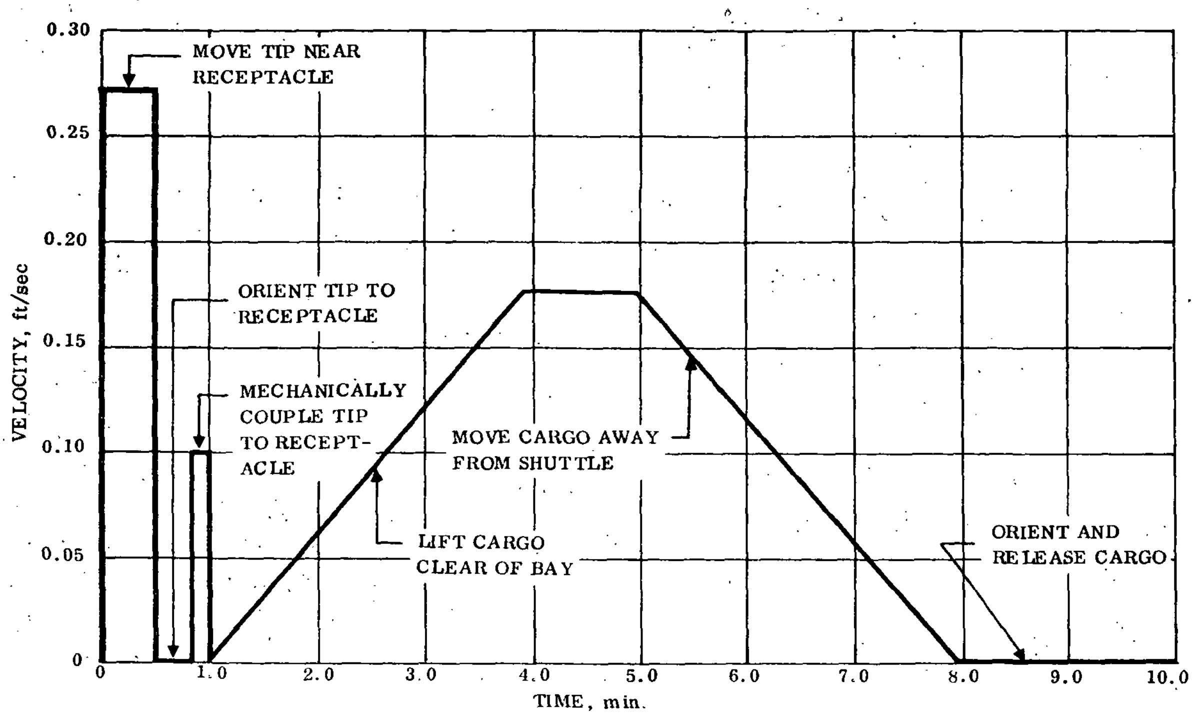


Figure 7-16. RMS Tip Velocity Time-line to Unload and Deploy Payload

Table 7-5. Remote Manipulator Characteristics

General Characteristics

- Two identical arms (most tasks only need one arm)
- Both direct and TV viewing from control station
- Operated by single crewman in shirtsleeves environment
- Control console area: 5 ft^2
- Total arm length: 50 ft.
- Total weight of an aluminum system: 2783 lb.

Performance Characteristics

- Tip force: 10 lb.
- Tip deflection: 1 inch
- Power consumption: 1008 watts
- Tip speed/no load: 1.5 ft/sec
- Tip speed/full load (20,000 - 65,000 lb): 0.174 ft/sec
- Tip deceleration/no load: Stop in 1.5 ft.
- Tip deceleration/full load: Stop in 15 ft.
- Tip positional accuracy: 2 inch
- Tip velocity accuracy: 0.05 ft/sec

Functional Flow Diagrams. The functions required by each transfer system are summarized in Figures 7-17, 7-18 and 7-19. In each case the overall functions are shown at the top with individual tasks expanded below. For the modular case, 4a represents transfer of the expended tank to the shuttle and 4b the transfer of the full tank to the station. Note that in terms of potential crew functions, high pressure and subcritical transfer are similar in that they primarily require the crew as a system monitor, whereas the modular transfer system requires more crew time and more specific skills as the crew serves as the remote manipulator operator. This advantage of minimal crew time and skill requirements that high pressure and subcritical transfer systems have over modular transfer systems would be only slightly reduced if multiple bottles were involved since additional remote valve actuation and deactuation to accomplish filling of individual bottles would need to be accomplished by the crew.

Summary of Crew Performance Comparisons. Table 7-6 summarizes the crew performance considerations. The subdivisions of subcritical transfer (metallic bellows, paddle vortex, etc.) are relatively non-impacting on crew performance and have been considered collectively. The operational functions of the flight crews have been discussed above. The modular transfer system has been rated higher in the category of flight crew maintenance, because the receiver tanks are replaced with each resupply flight, thus allowing any required maintenance on them to be performed on the ground. The other two systems use permanent space receiver bottles which would require periodic maintenance by the flight crew.

Ground crew functions appear to be similar regardless of the transfer system employed. Their number and the crew time to install, connect harnesses, etc., however, increases with multiple bottle techniques.

Crew support equipment such as manipulators, controls, displays, etc., increases with the crew's involvement in the transfer process.

Hazards are greater for high pressure systems or where the potential exists for loss of control over a module.

7.6 RECOMMENDATIONS AND SUMMARY OF COMPARATIVE DATA

This section presents recommendations and a discussion of the overall comparative data obtained for each of the transfer systems considered.

In general it was found that the fewer the number of bottles for a given fluid transfer requirement, the lower the weight, cost and crew requirements and the higher the reliability. Safety, with respect to separation of station fluids, and redundancy requirements would determine the minimum number of bottles which could actually be used. Information applicable to individual systems is presented below.

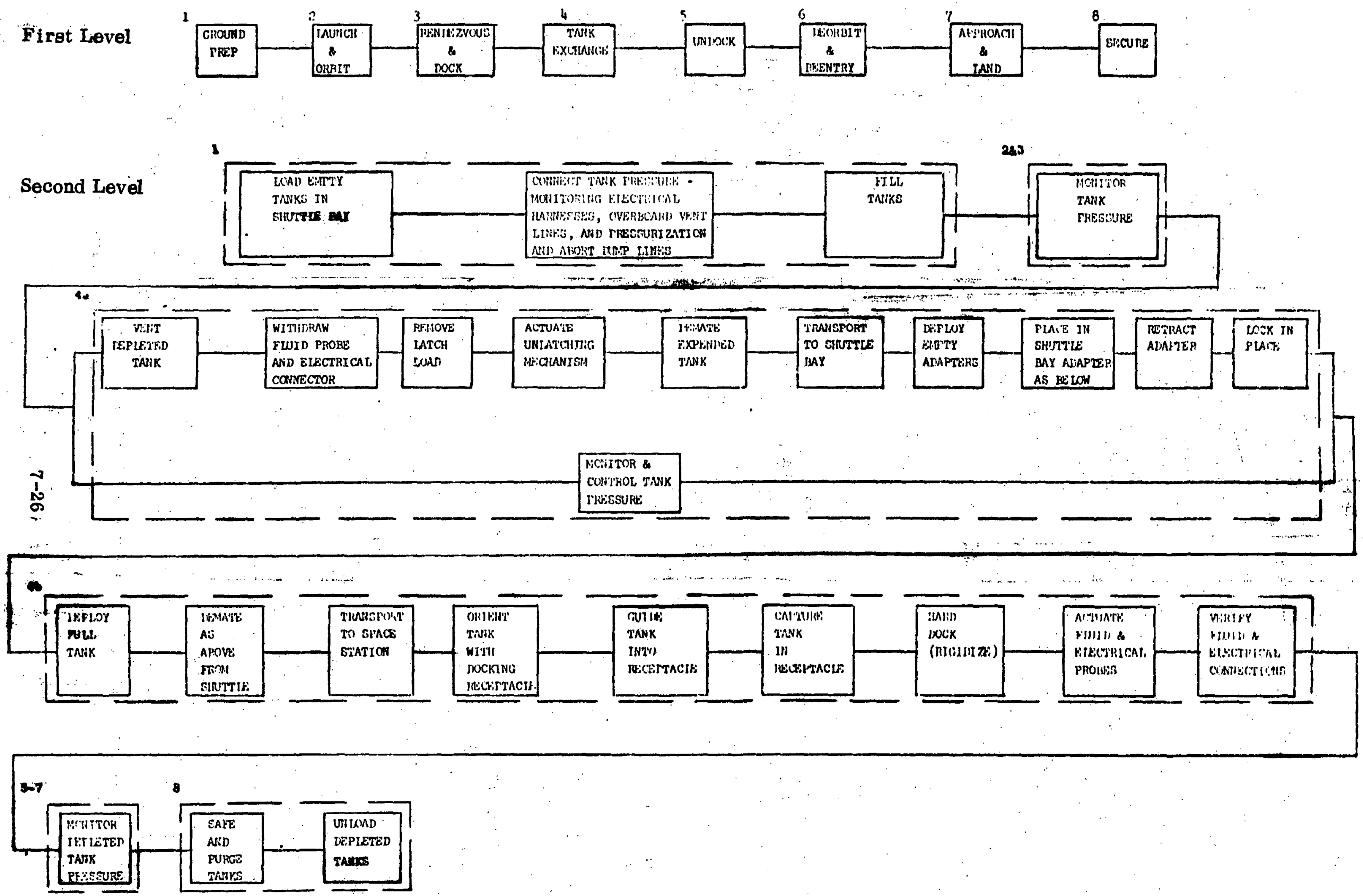


Figure 7-17. Modular Transfer System - Functional Flow Diagram

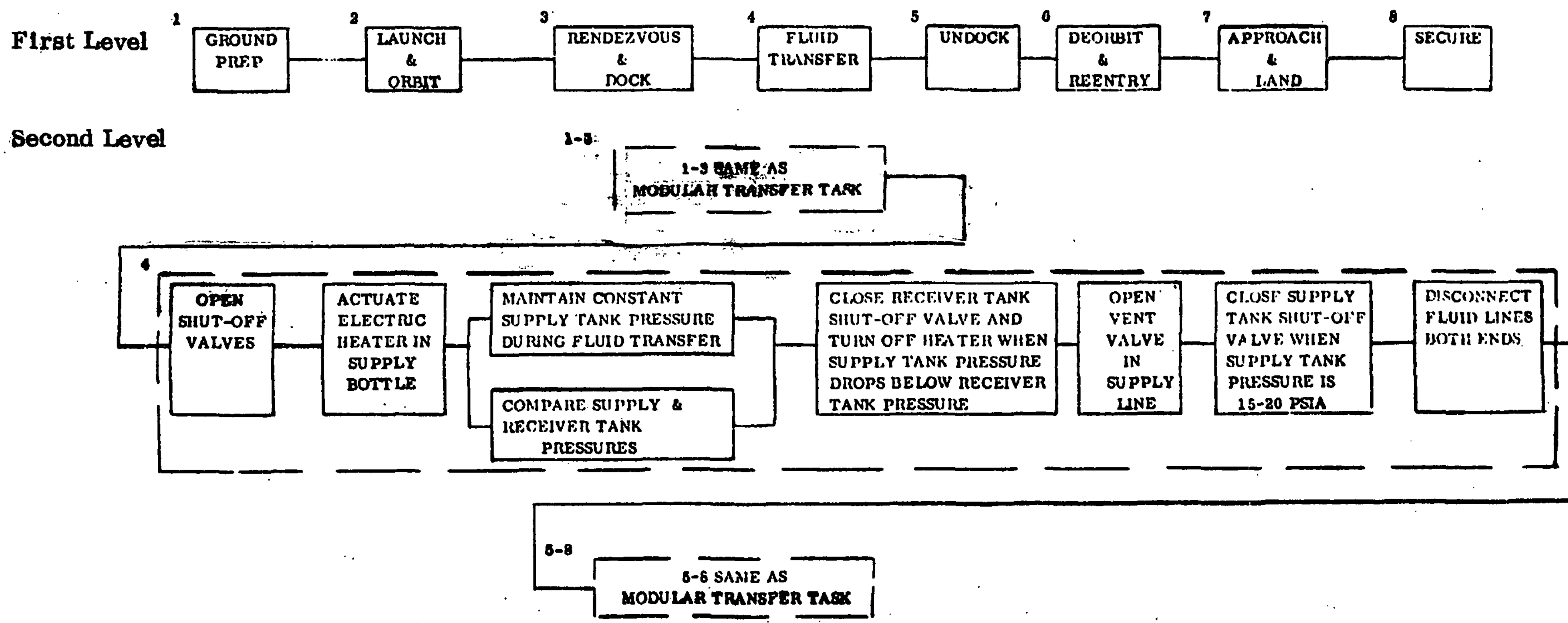


Figure 7-18. High Pressure Transfer System - Functional Flow Diagram

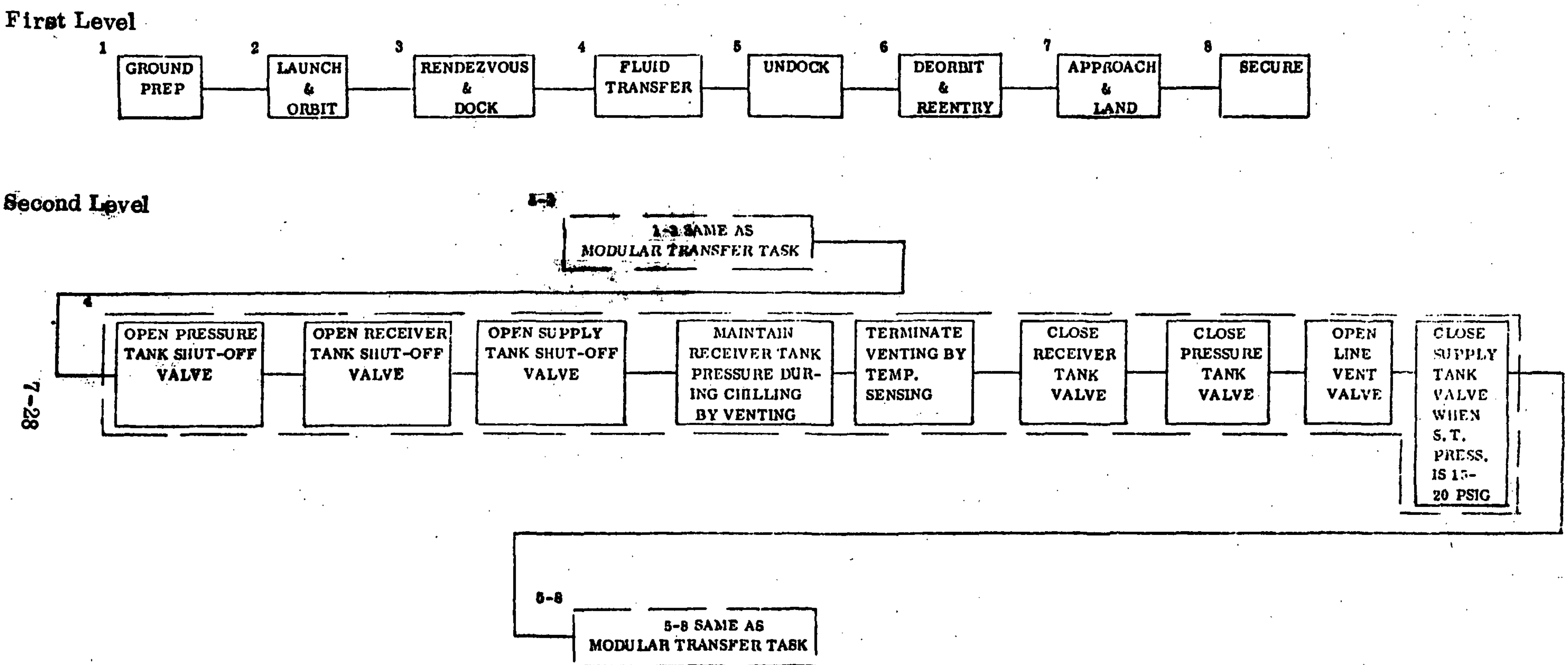


Figure 7-19. Subcritical Transfer System - Functional Flow Diagram

Table 7-6. Summary of Crew Performance Comparisons Between Candidate Systems

	Modular Transfer		High Pressure Transfer		Subcritical* Transfer	
	Single Bottle	Multiple Bottles	Single Supply And Receiver	Mult. Receiver Single Supply	Single Supply And Receiver	Mult. Receiver Single Supply
Flight Crew Functions						
Operational						
Number	High	Max	Low	Mod	Low	Mod
Crew Time Requirements	High	Max	Low	Mod	Low	Mod
Crew Training Rqmts (Skill Level Required)	High	High	Low	High	Low	High
Maintenance						
Number	Min	Low	Mod	High	Mod	High
Crew Time Requirements	Min	Low	Mod	High	Mod	High
Crew Training Rqmts (Skill Level Required)	Min	Low	Mod	High	Mod	High
Ground Crew Functions						
Number	Low	Mod	Low	Low	Low	Low
Crew Time Requirements	Low	Mod	Low	Low	Low	Low
Crew Training Rqmts (Skill Level Required)	Low	Mod	Low	Low	Low	Low
Crew Equipment						
Support Equipment Requirements	High	High	Low	Mod	Low	Mod
Hazards	Low	Mod	Mod	Mod	Min	Min

*The subdivisions of subcritical transfer (metallic bellows, paddle vortex, etc.) are relatively non-impacting on crew performance and have been considered collectively in this analysis.

7.6.1 HIGH PRESSURE. The primary advantage of this system, as compared to subcritical transfer, is that low-g liquid orientation is not required and thus the hardware for accomplishing high pressure transfer can be developed within the present state-of-the-art.

With respect to reliability, cost and crew requirements the system is comparable to the best subcritical concepts with only a small added hazard due to the higher pressures involved.

The main disadvantage is that energy rates required to accomplish transfer in a reasonable time are high. This is especially true for oxygen and nitrogen transfer where energy requirements were considered excessive for the present application. This conclusion could only change if an extremely high rate, low cost, low weight source of power were available on the station.

In the case of hydrogen systems, supply bottle weights were competitive, but receiver weights and volumes were significantly higher than for the other systems. Even assessing weights of hardware and power supplies located on the station at one-twentieth the supply weights, the high pressure system was found to have the highest overall weight of all fluid transfer schemes.

Another disadvantage of this system is that only the supply of supercritical receivers is practical. Condensation of fluid in the receiver to allow storage and subsequent use of a liquid was found to result in unreasonable power and hardware requirements.

It is recommended that the high pressure system not be considered further for the space station application unless station configurations with highly efficient power systems become available and long transfer times can be tolerated. It is noted that additional analyses would need to be accomplished, where long transfer times are involved, to account for environmental heating of the receiver tank.

7.6.2 SUBCRITICAL. The primary disadvantage of this method of transfer is that low-g liquid orientation and/or collection is required. Also, potential requirements for line and receiver tank chilldown, pressure control and supply tank pressurization adds to the system complexity.

Advantages are that both subcritical and supercritical receivers can be filled and, in comparison with modular transfer, weight, reliability and cost advantages result from using a single supply for each fluid rather than individual supply bottles for each receiver. Also, the design of the supply is not dependent on receiver requirements and vice-versa, and operation does not require significant crew participation or complicated bottle handling mechanisms.

Specific data for individual subcritical transfer systems in terms of the methods used for liquid orientation is presented below.

Surface Tension

Along with the paddle vortex system, the surface tension system represents the lowest weight of all the subcritical systems and is equally applicable to LH_2, LO_2 and LN_2 transfer. It also has the potential to be fabricated in all sizes studied. Next to the diaphragm, reliability is projected to be the highest of the subcritical systems, relative costs are low. Overall, this system is desirable for the present application and should be developed further, as discussed in Section 8.

Metallic Bellows

This system has the highest weight and unit cost and lowest potential reliability of all the subcritical systems. The main advantages of this system are that a positive orientation of liquid is accomplished and liquid quantity can be determined simply and with good accuracy at low-g.

Based on the weight and cost disadvantages of this system and the difficulties involved with fabricating large sizes (above 48 inches dia), it is recommended that such systems not be a primary development target for the present application.

Metallic Diaphragm

This system has only a slightly higher weight than the surface tension and paddle systems and has the highest potential reliability of the subcritical systems. The primary disadvantage for the present application is high cost, due to the requirement for replacement following every five flights. For the present reusable application, this system should be considered only as a backup, in the event the more complicated surface tension or paddle systems cannot be satisfactorily developed.

Paddle Vortex

In the case of hydrogen, this system, along with the surface tension system, has the lowest weight. Analysis showed a slight weight increase over the surface tension system for LO_2 and LN_2 use. Costs are estimated to be lower than the surface tension system, but with lower reliability due to required motor and drive elements.

The paddle system has good potential for the current application and unknowns with respect to fluid residuals and power requirements should be investigated further as discussed in Section 8.

7.6.3 MODULAR. The main weight penalty for individual modules is the requirement to insulate the supply for long term storage in the space station. This makes the use of small bottles unattractive where the boiloff must be restricted to 50% in 180 days. This penalty is significantly reduced where LO_2 and LN_2 are involved.

Even with this insulation penalty the modular systems have the lowest weight and also the best reliability where fairly large quantities of propellant are to be transferred on a single tank to tank basis. This is due to elimination of supply tank pressurization and receiver tank vent systems and transfer lines. This, of course, neglects the weight of remote manipulation systems required for modular transfer. Another important advantage of this system is that system maintenance would be much simpler than other systems since the bottles would be periodically returned to earth as a normal part of the operation.

A disadvantage of the system is that individual supply tanks are required for each receiver, resulting in high weight and cost when compared with subcritical systems using a single supply for a numberof receivers. Reducing the number of receivers for a fixed fluid requirement improves the relative cost and weight of the modular system. Use of a number of modules, however, does increase the ease with which redundancy can be incorporated into the transfer operation.

The primary overall consideration is the development of a suitable cargo handling system and the crew participation involved. Satisfactory development of this cargo handling system would be a controlling factor in the final choice of modular versus subcritical means of transfer.

8

DEFINITION OF FUTURE INVESTIGATIONS

As a result of the work described in the previous sections several promising systems were defined for the propellant transfer application. There are; however, uncertainties associated with these systems which require further theoretical and experimental investigations before flight qualification and operational use can be planned with a high level of confidence.

In the case of modular transfer schemes, the mechanics of docking and the engagement of fluid and electrical couplings are important. Further investigations for these systems should involve the design and construction of a prototype unit representing both the module and related space station components. Tests would be conducted to establish the level of crew efforts, system response to maximum tolerance bands, emergency or backup capabilities, life expectancy, and permissible leakage rates.

Fluid couplings (involving dynamic seals) and electrical connections usually become problem areas when cycled. Therefore, tests should include repeated docking and undocking at maximum pitch and yaw tolerances combined with random roll positions. Results of impact loading would be included. Investigations are recommended to determine what overall affects the ground service, boost, and transfer phases would have upon the module assembly. The primary hardware development associated with the modular concept is the shuttle cargo handling systems discussed in Section 7.5, a detail study of which was not within the scope of the present propellant transfer study.

In the case of subcritical transfer, two systems are considered to be worthy of future investigation. These are the surface tension and paddle systems described in Section 7.0. In both cases the overall transfer system is divided into supply, transfer and receiver elements.

A representative system for the space station application, employing surface tension orientation, is illustrated in Figure 8-1. As shown, the method of liquid acquisition and collection was chosen to be a double screen liner. Calculations (Section 5.0) showed that with the use of a reasonable amount of high performance insulation HPI; such a system could be operated through boost and orbital hold without venting. This assumes that the tank is designed for a maximum operating pressure of 100 psia. The primary problem with such a system is the prevention of liquid vaporization within the screen liner due to heat leakage through fluid lines and structural supports. The double liner is designed to allow wicking to maintain the screen wetted at all times. In the system shown, an optical screen baffle is used in the outlet line to prevent radiation heat transfer from direct vaporization of liquid within the liner. Where lines and supports attach directly to the screen, special wicking paths must be provided

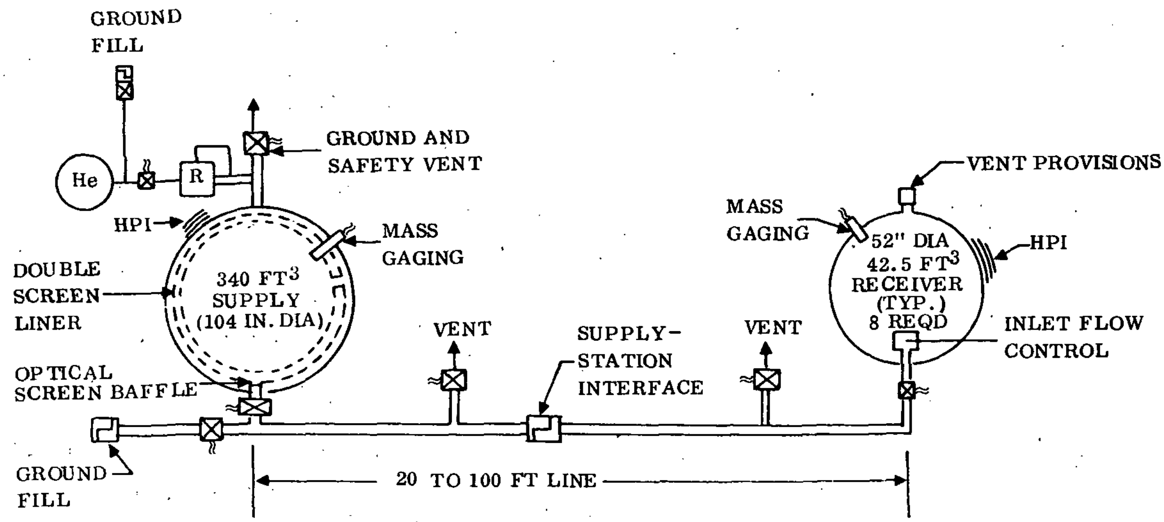


Figure 8-1. Representative H_2 Space Station Resupply System

to maintain these areas wet and at temperatures not exceeding saturation. In the system shown, helium stored at ambient conditions is used as the pressurant. The helium serves to prevent liquid flashing within the liner during transfer as well as to expel liquid from the tank.

It is noted that the transfer line length is not fixed and will depend on the location of the receiver in the station and the supply in the shuttle payload. The analysis performed on line and receiver tank chilldown (Section 5.3) indicated that the line length could have a significant effect on attempts to operate the receiver tank in a non-vent condition during chilldown. In this regard it is questionable whether or not the receiver tank can be chilled and/or filled without venting. Therefore details of the vent system and corresponding inlet flow control provisions are not shown in Figure 8-1. Three potential receiver configurations are presented in Figure 8-2.

For the case where a non-vent chilldown is to be accomplished, a receiver system such as shown in Figure 8-2a would be used. Analysis has shown that the minimum tank pressure rise occurs when the incoming liquid is completely dispersed with the ullage and maintains minimum contact with the walls. This allows the most liquid to finally absorb the total energy from the tank walls resulting in a minimum total pressure rise. The system shown in Figure 8-2a employs a screen inlet diffuser to spray liquid into the ullage and minimize heat transfer at the walls.

In the case of long transfer lines where non-vent transfer is unfeasible and venting is required, potential systems are illustrated in Figures 8-2b and 8-2c. In the Figure 8-2b system, the inlet flow is directed at the walls of the tank but away from the vent, the opening of which extends some distance into the tank. The flow impinging on the wall is designed to promote a maximum of liquid evaporation and a minimum of liquid available at the vent. The flow rate is also chosen such that any lqiuid not immediately vaporized will flow around the wall and not out the vent. This configuration is a result

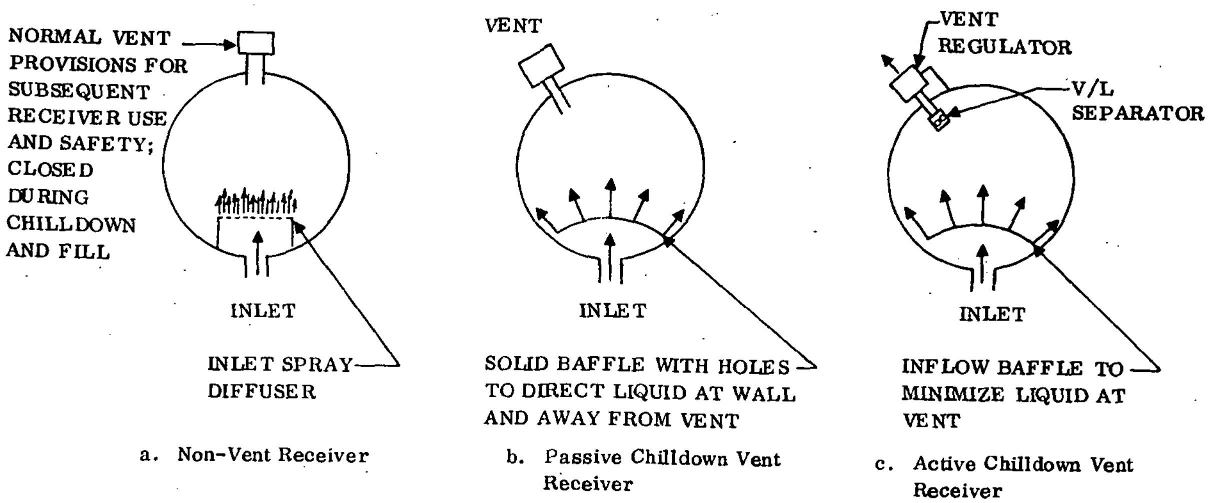


Figure 8-2. Typical Receiver Configurations

of drop tower testing performed at LeRC using non-cryogenic fluids (Ref. 8-1 and 8-2). Testing is presently being set up at LeRC using LN_2 flowing into a 2 ft diameter aluminum tank during a 5 sec drop to verify the efficiency of this type of concept (Ref. 8-1).

It is possible, however, that under boiling conditions a significant amount of liquid may be entrained in the vent gas and lost overboard unless active liquid/vapor separation is accomplished using a system such as shown in Figure 8-2c. Such systems were studied under contract NAS8-20146 (Ref. 5-5).

Based on a review of the state-of-the-art of the transfer system illustrated in Figure 8-1, it is recommended that the following areas of performance be defined and/or demonstrated through testing and/or analysis.

I. Supply Associated Items

A. Acquisition System

1. Ability to fill at 1-g.

2. Fabrication and maintaining cleanliness over system life.

3. Thermal performance (prevention of vapor formation within double liner due to heating and/or loss of pressure) during boost, coast and fluid transfer. Important considerations in this regard are:

a. Wicking characteristics of screen.

b. Ability of screens to prevent radiant heat transfer.

c. Structural attachment of lines and supports to allow wicking and prevent temperatures above saturation at the screen.

d. Effects of warm pressurant gas on the formation of vapor within the liner.

Liquid Outflow Characteristics at Low-g

a. Residuals as a function of flow rate

b. Effects of vehicle disturbances

B. Mass Gaging System Operating at Low-G

1. Accuracy at all levels of fill mass.

2. Effect of screens.

C. Tank Pressurization During Outflow With Respect to Prediction of Pressurant Requirements for Expulsion.

II. Transfer Line

A. Prediction of Pressure, Temperature and Quality of Fluid at Line Outlet During Chilldown, Including Effects of Potential Pressure and Flow Surging.

B. Development of Reliable Automatic Connections Between Supply and Station.

III. Receiver Tank

A. Mass Gaging. Type of System Developed for Supply Tank Should be Adequate.

B. Tank Pressure Control During Chilldown With;

1. Initially warm line and cold tank

2. Initially warm line and tank.

C. Tank Pressure Control During Fill Following Any Required Chilldown

NASA directed work, such as NAS8-18039, is presently underway on the solution of the problems associated with low-g mass gaging. Tank pressurization and line chilldown are problems which have previously been analyzed and a significant amount of data is available. With respect to pressurization, where the liquid/vapor interface may not be fixed, further work must be accomplished to determine pressurant cooling under conditions of liquid motion as a function of pressurant flow rate or expulsion time. This testing could primarily be done at 1-g with induced sloshing. Conservative assumptions about the actual degree of sloshing would then need to be made in any final analysis. In the case of the transfer line, at reasonably high flows, the operational line conditions can be satisfactorily simulated by 1-g testing.

A discussion is presented in the following paragraphs of the most pertinent technology programs required to demonstrate performance of the various liquid orientation and receiver pressure control systems.

8.1 SURFACE TENSION SCREENS

Screen fabrication and performance test programs are described below.

8.1.1 FABRICATION. The primary structural considerations in designing the capillary device are providing the ability to resist fluid impingement loads, screen pressure drops, and deflections which could cause structural failure of the configuration or alteration of the micron rating. An important fluid performance criteria is to retain the bubble point of the screen within a narrow band consistent with hydrostatic head requirements. This implies seals and connections which do not detract from capillary device bubble point. Another important performance parameter is screen pressure drop. Retaining the screen pressure drop within narrow limits requires consideration of the filtration problems.

Capillary devices have been fabricated and have performed successfully under orbital design conditions. Fabrication has, to date, been limited to small sizes of below 5 feet and to designs which are used only once and not reused. A program is thus required to demonstrate fabrication technology for large capillary devices and to develop techniques for insuring reusability.

In order to promote reuse, the primary area of concern is cleanliness. Procedures must be developed for eliminating contamination in the tank, in the fluid, and on the capillary device surfaces. This should be done by providing initially clean fluid and surfaces and by minimizing handling through development of subassembly procedures for installing the capillary device in the tank. In order to minimize turn around time, repair techniques must be developed so that the large capillary devices can be repaired without removing the entire device from the tank. In order to determine whether the capillary device will perform successfully upon reuse, a procedure for in-tank check-out by measuring capillary device bubble point and screen pressure drops needs to be developed.

A large scale device should be fabricated, installed in a tank and subjected to a simulated acceleration, vibration, and outflow environment to provide a life-cycle test of the capillary device structural capability.

8.1.2 PERFORMANCE TESTING. The major performance problem with the screen device is keeping the screen wetted during standby and while transferring in order to prevent vapor from being blown into the receiver tank. Prevention of screen drying in the present case is by screen wicking. Optical baffling to prevent direct heating of the contained liquid by radiation from transfer line surfaces is also proposed. Testing required is divided into basic wicking tests and overall system tests which can be performed at one-g. Details are presented below.

Wicking Tests

These tests are designed to obtain the basic wicking data required to define a complete operational screen acquisition system. As an example, the ability to fabricate seams, corners, supports and outlets such that wicking is satisfactorily accomplished in these areas would be demonstrated, as well as the wicking characteristics of the basic screen material. Testing can be accomplished using both calibrated heating and volatile fluids (fluids which boil at atemperatures near ambient) and visual inspection and non-volatile fluids. The basic screen wicking test data will be used to provide input constants for wicking equations developed under NAS8-21465 (Ref. 3-1).

Seams, corners, supports and liquid outlets are designed to be representative of a full scale operational screen system. It is anticipated that two configurations or types of screen material would be chosen to represent each of the above areas. This results in a total of 10 specimens including two simple flat screen samples used as a reference.

For one of the screen material tests and one of the fabricated specimens, several fluids should be used to confirm the fluid scaling properties determined during contract NAS8-21465 (Ref. 3-1). For some of the samples, both a non-volatile and volatile fluid would be used and compared to verify wicking capacity predictions. Several different sample orientations with respect to the 1-g gravity vector would be tested in each case. A typical test set-up as taken from Reference 3-1 is presented in Figure 8-3. A total of approximately 60 tests are proposed. In each case wicking rates and liquid covering capabilities would be measured visually by eye and by camera. In selected cases, for the volatile fluid, heating rates would also be measured with a wattmeter, and a wattmeter and thermocouple would be attached to the sample guard to obtain an accounting of losses at the edge of the sample.

Coupled with a knowledge of the fluid heat of vaporization, these data can be used to determine wicking rates of a wetted screen operating under steady state or equilibrium conditions. This data would be correlated with the visual data to verify the accuracy and applicability of the overall test series. Testing similar to that described above was performed on a limited scale and is described in Reference 3-1. The above data would be reduced and correlated with an analytical model designed to calculate wicking rates (Ref. 3-1).

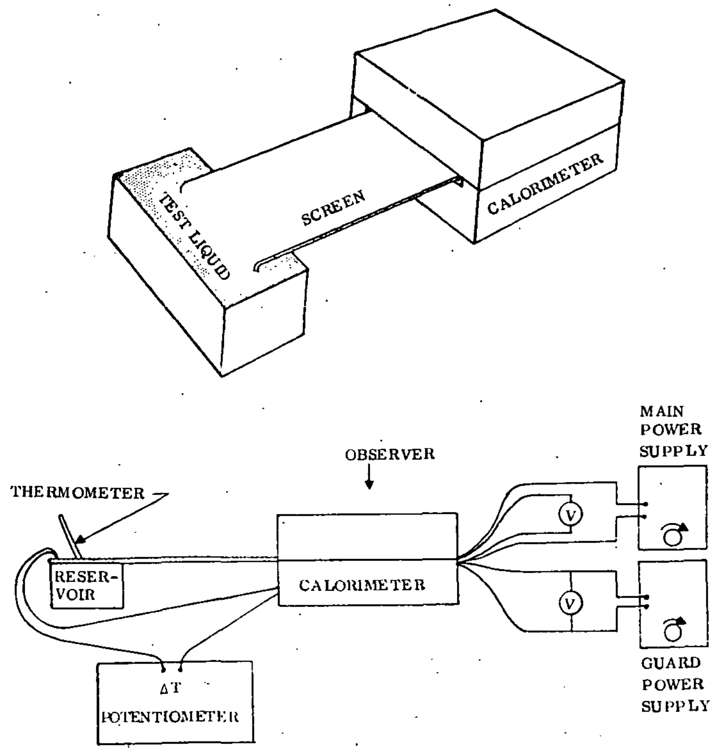


Figure 8-3. Wicking Test Set-Up

Integrated Surface Tension System Test

Filling, draining and liquid retention testing should be performed to demonstrate satisfactory operation of the overall surface tension concept. Due to the low temperatures involved hydrogen is the most critical fluid from the standpoint of maintaining a wetted screen outflow under expected environmental heating conditions. It is therefore proposed to accomplish the system demonstration testing with hydrogen. The system operating parameters and characteristics to be determined from the test program would be:

1. 1-g filling
2. structural integrity
3. overall bubble point with hydrogen
4. pressure drop and residuals during draining
5. reusability
6. effect of pressurant addition during outflow using both hot and cold pressurant gas
7. effectiveness of wicking and optical baffling to prevent vapor formation within the screen liner.

It is assumed that a test tank on the order of 16-inch diameter would be used. The 16-inch tank size represents a maximum which is reasonable to accommodate 1-g testing using screen mesh sizes similar to those anticipated for an operational system. A schematic of a representative test setup is presented in Figure 8-4. The capillary system design would be made to simulate support, screen joining, penetration transitions and corner sections to be expected in a full size system. This would allow a reasonable assessment of actual pressure drops, bubble points and fabrication difficulties which may be encountered in an operational system. Also, an opaque screen at the outlet, as shown in Figure 8-4, would be provided to prevent boil-off from outlet line radiation. Fabrication and installation of the capillary device would be accomplished in a manner similar to that expected for an operational system.

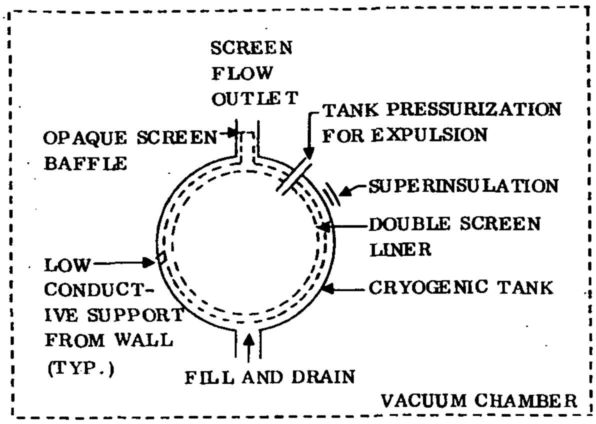


Figure 8-4. Surface Tension Verification Test

The basic test sequence would be to chill-down the tank system, fill the capillary device with liquid, allow the system to remain in a standby condition (no outflow) for a period of time, and then perform an outflow to empty the tank. Capillary device filling should be demonstrated at a minimum of three different fill rates. Liquid outflow would be accomplished at four different flow rates. A minimum of two of these should be repeated in the course of the testing in order to verify the data. Standby times must be sufficient to reach equilibrium and to demonstrate steady state performance.

In all cases the primary test evaluation parameter should be the location of liquid or the absence of vapor within the capillary device, particularly at the outlet. Carbon resistors would be spaced throughout the tank and within the capillary device with a concentration at the outlet in order to evaluate the effectiveness of cooling and mixing to prevent internal vapor formation. Carbon resistors would also be closely spaced at the bottom of the tank in order to evaluate the quantity of residuals following outflow. Temperatures would be measured throughout the tank using platinum resistance probes. Inlet and outlet flow rates would be measured using turbine flow meters. Temperature, pressure and liquid sensing measurements would also be made in the outlet feed line.

A correlation should be made with the data obtained from wicking tests as discussed in the previous section and the outflow screen pressure drop and residual fluid computer program used in the analyses of Section 5.4 and described in Reference 4-1.

8.2 PADDLE VORTEX SYSTEM

Primary difficulties with this system are the prediction of fluid residuals and power requirements. As with all the subcritical systems,meaningful evaluation of system operation at low-g through testing at one-g can be a problem.

In examining the low-g to one-g scaling the following equations from Section 5.7 are used.

$$\text{Required Paddle Rotation } (\omega_p) = \sqrt{\frac{a}{R_i}} \tag{8-1}$$

where, a = disturbing acceleration trying to position liquid away from the outlet

R_i = inner radius from which liquid must be pumped against the acceleration a.

$$\text{Power to drive paddle } (\dot{P}_p) = \text{Constant } (\omega_p)^3 \ \rho_L \ R_p^5 \qquad (8\text{-}2)$$

Using m as a subscript to designate the test model; then from Equations 8-1 and 8-2,

$$\frac{(\omega_p)_m}{\omega_p} \quad \sqrt{\frac{a_m/a}{(R_i)_m/R_i}} \qquad (8\text{-}3)$$

and

$$\frac{(\dot{P}_p)_m}{\dot{P}_p} = \frac{(\omega_p)_m^3 \ (\rho_L)_m \ (R_p)_m^5}{\omega_p^3 \ \rho_L \ R_p^5} \qquad (8\text{-}4)$$

Taking $a_m = 1$ g and $a = 10^{-4}$ g's,

$$(\omega_p)_m = 100 \ \omega_p \ \sqrt{\frac{R_i}{(R_i)_m}} \qquad (8\text{-}5)$$

Assuming a full scale model using the same fluid as the prototype,

$$(\omega_p)_m = 100 \ \omega_p$$

and

$$(\dot{P}_p)_m = 1 \times 10^6 \ \dot{P}_p$$

For the case of a 52-inch diameter tank with H_2; from Section 5.7, $\omega_p = 6$ rpm and $P_p = 0.0175$ Hp. The model would then need to operate at 600 rpm with a predicted power input of 17,500 Hp. In the case of LO_2 this power would be 148,000 Hp. Referring to Equation 8-4, the use of a lower density fluid in the model than in the prototype would have some effect, but not sufficient to make testing at 1-g feasible under the above acceleration and size conditions.

To consider the use of subscale models, Equations 8-3 and 8-4 are combined to give,

$$(\dot{P}_p)_m = 1 \times 10^6 \left[\frac{(\rho_L)_m}{\rho_L}\right] \left[\frac{R_i}{(R_i)_m}\right]^{3/2} \left[\frac{(R_p)_m}{R_p}\right]^5 \qquad (8\text{-}6)$$

Taking R_i = Constant × R_p for both model and prototype,

$$\left(\frac{R_i}{R_p}\right)_m = \frac{R_i}{R_p}$$

and

$$(\dot{P}_p)_m = 1 \times 10^6 \ \dot{P}_p \left[\frac{(\rho_L)_m}{\rho_L}\right] \left[\frac{(R_p)_m}{R_p}\right]^{3.5} \qquad (8-7)$$

Assuming a 1/10 scale model with the same fluid as the prototype, then from Equation 8-7,

$$(\dot{P}_p)_m = 314 \ \dot{P}_p$$

For the hydrogen case,

$$(\dot{P}_p)_m = 5.5 \ H_p$$

and from Equation 8-5,

$$(\omega_p)_m = 1880 \text{ rpm}$$

The above analysis indicates that subscale testing at one-g could be used to evaluate hydrogen system performance with some confidence.

The above analysis was repeated for a 104-inch diameter operational tank resulting in similar power and speed requirements for the same size model (1/20 scale).

Due to the extreme size scaling involved the final verification of system performance would need to be accomplished by orbital testing under operational conditions.

8.3 RECEIVER TANK

A significant problem with respect to the prediction of overall system operation during low-g transfer is associated with tank pressure control during chilldown and fill, with the chilldown portion being the most critical. There are two basic methods of filling a receiver tank. One is to maintain the tank in a locked-up (no-vent) condition and design the tank to withstand any resultant pressure rise. The other is to maintain a specified maximum pressure by venting.

The problem with venting, however, is that some means must be provided to prevent an excessive loss of liquid. Both passive and active means of preventing liquid loss have been proposed, as illustrated in Figures 8-2b and 8-2c.

The primary problem with a locked-up tank is that for small tank sizes and long inlet lines it is sometimes not feasible to design the tank to take the pressure which can occur during chilldown. That is, the higher the design pressure the heavier the tank and resultant chilldown pressures. Even where it is feasible, certain fill and heat transfer conditions must be met.

Fluid inflow dynamics and heat transfer at low-g during transfer must be known in order to define the optimum fill method and its performance as to maximum pressure and/or quality of fluid vented. Suggested analytical models and required testing and test limitations, with respect to determining the performance of the systems illustrated in Figure 8-2, are discussed in the following paragraphs.

8.3.1 ANALYTICAL MODEL. A numerical technique has been developed at Los Alamos Scientific Laboratory which can serve as a basis for solving the cryogenic liquid transfer problems discussed above. This method, called the Simplified Marker and Cell (SMAC), (Ref. 8-3) is an improvement of the Marker-and-Cell (MAC) method (Ref. 8-4, 8-5 and 8-6) previously developed by Los Alamos. Both methods consider the time-dependent, viscous flow of an incompressible fluid, however SMAC is significantly simpler to use, particularly for arbitrarily shaped boundaries. Under Contract NAS8-21291 (Ref. 8-7 and 8-8) Convair Aerospace developed a computer code using the MAC finite difference technique. This code could handle liquid in a container with rectangular boundaries and considered not only the fluid motion but also heat transfer, binary gaseous diffusion, inflow and outflow. It could also handle variable cell mesh in both plane rectangular and axisymmetric problems. Its main limitation was that the boundaries had to be rectangular. Under Contract NAS3-14361 (Ref. 8-9) Convair developed a computer code using the SMAC technique which could handle arbitrarily varying boundaries. However, this model is limited by a fixed cell mesh and the inability to handle inflow, outflow and heat transfer. What is needed is a single program which has all the features of each computer program.

While presently no existing program has all the options of the new proposed code all the options required have been successfully accomplished in separate programs. Therefore this task mainly involves combining existing information together into one program.

To accomplish this objective the computer code developed under Contract NAS3-14361 could be modified to handle inflow, outflow, and heat transfer. So that the problems can be handled efficiently, the program should also be modified to include a variable cell mesh network.

8.3.2 TESTING. Test limitations and requirements associated with the Figure 8-2 receiver systems are discussed below.

Non-Vent Childown (Figure 8-2a)

Based on tank chilldown calculations described in Section 5.3 the following fill conditions were estimated to be required in order to allow a non-vent chilldown and fill of a 52-inch diameter tank with hydrogen.

Fill time =	30 sec
Fill mass =	144 lb

Fill Rate	=	4.8 lb/sec
Inlet Velocity	=	90 fps
Inlet Line Dia.	=	1.5 inches for 20 ft transfer line length

Work performed in-house at LeRC (Ref. 8-10, 8-11 and 8-12) indicated that the inlet Weber No. defined as

$$We = \frac{R_{in} (V_e)_{in}^2}{2\ (\sigma_L/\rho_L)} \tag{8-8}$$

was the controlling parameter in determining fluid conditions in a tank at low-g. Work by LMSC (Ref. 8-13) at 1-g showed that the Froude Number

$$Fr_N = \frac{(V_e)_{in}^2}{2a\ R_{in}} \tag{8-9}$$

was important for fluid inflow and that the dimensionless mass flow parameter

$$\dot{m}^* = \frac{\dot{m}_{fill}\ C_{pL}}{k_L\ D_t} \tag{8-10}$$

must be maintained constant to simulate the thermodynamic chilldown aspects of the problem.

Calculations for the present H_2 inflow conditions resulted in

$$We = 2.49 \times 10^5$$

and

$$Fr_N = 2.01 \times 10^7$$

where $a = 10^{-4}$ g's

Assuming that 1g testing is accomplished with a full size (52-inch dia) tank using the same fluid, inlet line and flow rate, then the Weber Number and dimensionless mass flow would remain constant and the Froude Number would be 2.01×10^3. It is noted that even though the Fr_N has changed considerably, it is still well into the inertia dominated regime. Therefore, in this case, it is possible that testing could satisfactorily be accomplished at 1-g in a full size tank. The effect of the change in Froude Number would, however, need to be investigated. Testing should be accomplished in several different size tanks in order to determine the effect of changes in the various dimensionless parameters. As an example, reducing the tank and inlet size by one-half and adjusting the mass flow rate to maintain $\dot{m}^*$

constant results in an increase of 2 in the Weber Number and 8 in the Froude Number. Therefore, fairly large variations in the Froude Number can be investigated with relatively smaller changes in the other parameters. In the case of testing in a drop tower, the primary limitation is time and size. Assuming the use of the operational fluid as the test fluid, in order for the three dimensionless numbers (Equations 8-8, 8-9 and 8-10) to be maintained constant between model (m) and prototype, the following relations must be maintained.

Dimensionless mass flow constant; $D_m V_{e_m} = D V$

Weber Number constant; $D_m V_{e_m}^2 = D V_e^2$

Froude Number constant; $V_{e_m}^2 / a_m D_m = V_e^2 / a D$

It can be seen from the above that the Weber number and dimensionless mass flow number could never be maintained constant unless both tank size and inlet velocity are also maintained constant. However, by proper control of the applied acceleration in a drop tower, the Froude Number and either the Weber Number or dimensionless mass flow can be maintained constant. As an example, using a one-half size tank the acceleration would need to be respectively four times and eight times that of the prototype to maintain the Weber Number and dimensionless mass flow constant when the Froude Number is also maintained constant. Effects of changes in the various parameters can be determined by performing tests with different size tanks and inlets and at different flow rates.

For a system where a non-vent chilldown is proved to be marginal, final demonstration testing must be in a full scale system in orbit.

Passive Chilldown Vent (Figure 8-2b)

Recent drop tower work at NASA/LeRC (Ref. 8-1 and 8-2) has indicated that a relatively passive system, as shown in Figure 8-2b, could be used to allow receiver tank chilldown with a minimum of liquid loss. The testing indicates that at inflow Weber Numbers of 600 to 700 and even up to 11,000 that smooth flow patterns of liquid around the tank walls existed such that liquid venting could be prevented. The testing was accomplished with line inlet diameters approximately 1/25 that of the tank. Assuming this condition for the 52-inch diameter tank of the present operational case, then from Equation 8-9 for hydrogen the inlet velocity to provide a Weber Number of 600 would be 3.83 fps. Considering testing at one-g with this inflow velocity and Weber Number, in a full size tank and using the same fluid (H_2), the Froude Number would be from Equation 8-9 equal to 2.74. This would be much less than the operational case and would be too close to one (where inertial and gravity forces are comparable) to expect good correlation.

If it could be shown that higher Weber Numbers, such as 11,000, could be used for the operational system then 1-g testing would have a greater chance of providing some useful data. The use of small scale models would have the effect of allowing an increase in the Froude number while maintaining the Weber Number. However, in any case where the tank size is changed there would be considerable difficulty in scaling the thermal performance of the system. This would also be the main difficulty with drop tower testing where test sizes and times are limited. Where venting is being accomplished, the heat-transfer-time relation and fluid dynamics are the most important parameters to model rather than absolute pressure change rates and therefore thermal equations other than Equation 8-10 will likely be applicable. This should be further defined following cryogenic drop testing presently scheduled at NASA/LeRC (Ref. 8-1).

Active Chilldown Vent (Figure 8-2c)

Assuming that passive means do not prove highly efficient under all required conditions, the use of a mechanical vapor/liquid separator may be optimum. Such systems should be able to vent high quality vapor when liquid fractions up to 90 percent exist at the vent inlet. Knowledge of flow dynamics in the tank, as discussed in connection with the passive vent, must be obtained to the extent required to determine that the liquid fraction at the vent is below a specified upper limit. This information should be available from the (Reference 8-1) drop tower work. Thermal performance testing can then be accomplished at 1-g with worst case fluid conditions being imposed directly at the separator inlet. This can be done by using a controlled spray at the separator inlet in conjunction with liquid vent measurements. This would be performed while the separator is both rotating and not rotating at various orientations with respect to gravity.

If the separator is operated at zero rpm, the difference between the mass flow rate of liquid being sprayed on the unit and that being vented will be due to boil-off losses and the separation effect of gravity. Thus these effects can be separated out when evaluating the performance under normal rotating conditions.

Conclusions

Overall testing with respect to receiver tank chilldown and fill should include the following.

a. One-g testing in various size tanks with non-cryogenic fluids to determine the fluid dynamic scaling effects of variations in Weber and Froude Numbers at high inflow rates.

b. One-g testing with cryogenic fluids,directed to obtaining adequate combination fluid dynamic and thermal models and scaling criteria for definition of full scale system operation. Transfer line chilldown and flow characteristics effect receiver tank conditions and should be included as part of the overall testing and evaluation.

c. Drop tower testing is needed to investigate flow patterns occurring at fairly low Weber Numbers and to fill in gaps in the data obtained at one-g. In this regard, times required to simulate thermal effects in small drop tower tanks should be investigated and the limitations of such testing defined.

d. Long term orbital experiments in large size tanks should be planned to verify 1-g and drop tower data obtained and to extend the range of this data to the performance demonstration of full scale systems.

Cryogenic Test Program

Following is an outline of a test program to accomplish the objectives of b. above.

The proposed program is designed to determine transfer line and receiver tank fluid conditions as to liquid dispersion, fluid temperatures and pressures and vent efficiency during chilldown and fill at several different flow rates and with several different tank inlet configurations. Hydrogen is considered to be the most critical fluid and is, therefore, proposed as the test medium. A representative test set-up is illustrated in Figure 8-5.

The system will include inlet baffling and vent and instrumentation provisions. Inlet baffling will be designed to promote mixing of the fluid in the tank and/or to minimize liquid existing at the vent during chilldown. A subscale system is anticipated in order to simulate the fluid dynamic conditions to be expected at low gravity.

Temperature and liquid sensing probes will be strategically placed to measure both receiver tank and transfer line conditions. The presence of any liquid in the vent stream will be measured, as well as tank pressure at all times. Line pressure transients will also be monitored. Chilldown and fill testing will be conducted under simulated space environment heating conditions with the receiver tank being super-insulated in a high vacuum. Testing should be accomplished at a minimum of four different flow rates and two different inlet baffle configurations. In each case testing should be accomplished under both venting and locked-up conditions.

Comparisons and correlations would be made between the experimental data and analytical models such as described in Section 8.3.1.

P = PRESSURE

T = TEMPERATURE

× = PLATINUM TEMPERATURE PROBES

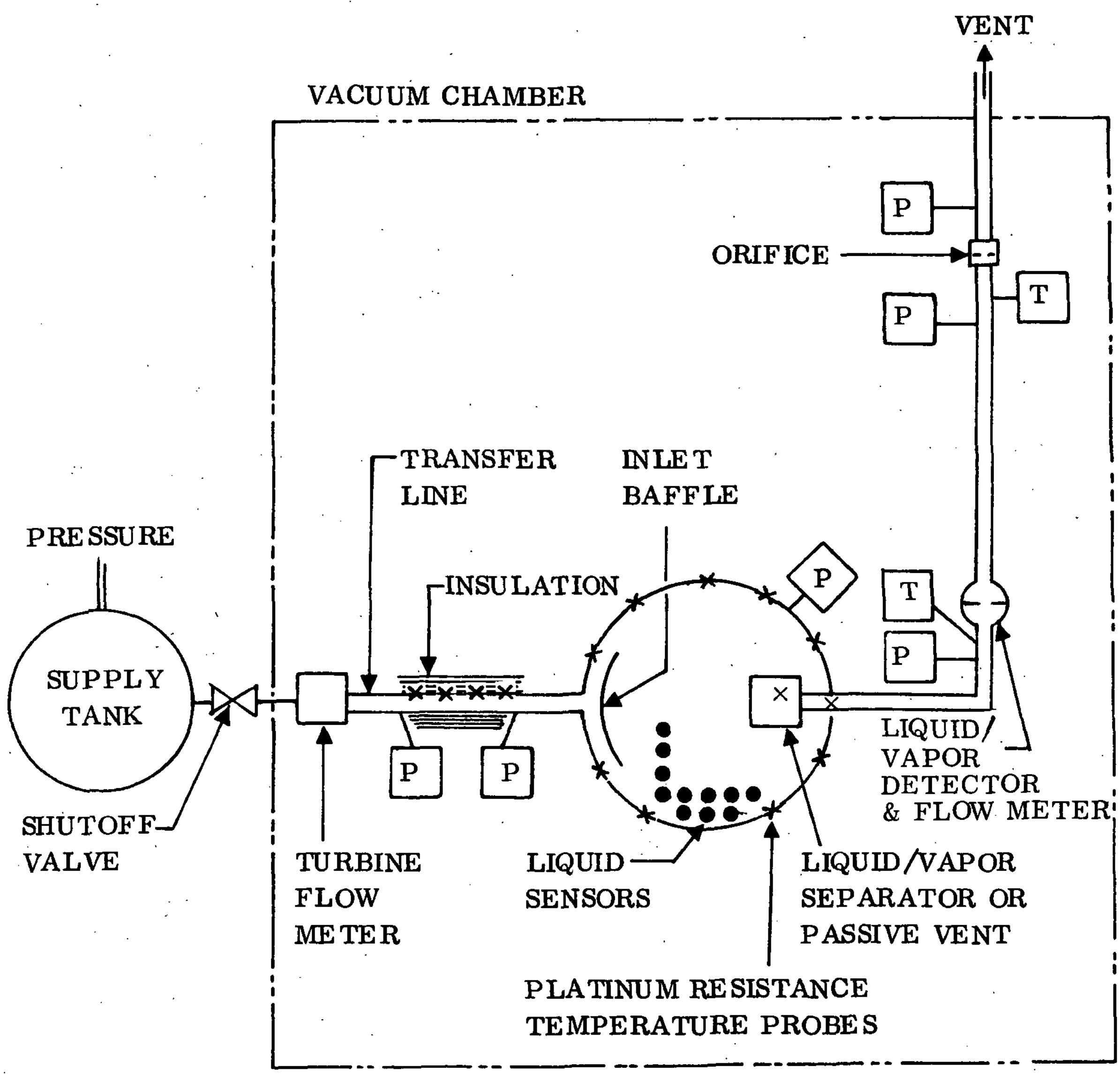


Figure 8-5. Typical Receiver Tank and Transfer Line Test Schematic

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CONCLUSIONS

The significant conclusions resulting from this study are presented below.

a. In the case of high pressure transfer, the only concept found to be worthy of detailed analysis was one employing simple supply bottle heating. The main disadvantage with high pressure transfer is that energy rates required to accomplish transfer in a reasonable time are high. This is especially true for oxygen and nitrogen transfer. Also, it was found that only the supply of supercritical receivers is practical. Condensation of fluid in the receiver tank to allow storage and subsequent use of a liquid results in unreasonable power and hardware requirements. The high pressure system should not be considered further for the space station application unless station configurations with highly efficient power systems become available and long transfer times approaching 24 hours are desirable. It is noted that additional analyses would need to be accomplished where transfer times greater than 2 hours per bottle are involved, to account for environmental heating of the receiver tank.

b. In the case of subcritical systems, a screening analysis on the basis of safety, weight and development potential resulted in the selection of surface tension, metallic bellows, metallic diaphragm and paddle vortex methods of low-g liquid orientation for detailed definition and analysis. Of these systems the surface tension and paddle systems were determined, on the basis of low weight and cost and high reliability and reusability, to have the best potential for the present application. Also, for a subcritical supply bottle designed for 100 psia, thermal analysis showed that use of a locked-up (non-vented) tank resulted in the simplest storage, with a minimum weight penalty, for the seven day period prior to transfer.

c. For the present application, a surface tension system having a double screen liner was chosen for simplicity. In this system the liner is isolated from the tank wall and wicking is relied upon to maintain the screens wetted at all times, to prevent vapor from entering the liquid outlet. Uncertainties with this system requiring further demonstration are associated with structural integrity of large size screens, cleanliness over repeated flow cycles, maintenance of proper wicking at seams, corners, supports, and outlets and overall cryogenic flow performance.

d. The paddle vortex system operates by creating a centrifugal acceleration on the supply liquid to maintain it at the tank outlet for transfer. For the present case an electric motor operating through a hermetically sealed flex spline was chosen to drive the paddle.

Fluid residuals, however, can be quite high for this system unless special sump designs are incorporated. Also the power requirements are uncertain and subscale model testing is needed to demonstrate further confidence in the system.

e. A significant problem with respect to the prediction of overall subcritical system operation during low-g transfer is associated with receiver tank pressure control during chilldown and fill, with the chilldown portion being the most critical. There are two basic methods of filling a receiver tank. One is to maintain the tank in a locked-up (no-vent) condition and design the tank to withstand any resultant pressure rise. The other is to maintain a specified maximum pressure by venting. In the case of a locked-up tank, high inflow rates and/or correspondingly low heat transfer rates are required to minimize receiver tank pressure. Even under optimum conditions it was shown that for certain tank sizes and design pressures, H_2 chilldown of a locked-up tank was not feasible. That is, the maximum pressure reached cannot be maintained below the tank design pressure. The larger the tank the more likely a non-vent transfer would be feasible. Also the use of CRES lines and tanks results in reduced pressure rise. In order to provide high inflow rates, line sizes need to be large with relatively high mass and a resultant increase in the final receiver tank pressure. Calculations for the 42.5 ft^3 station receivers showed in this case a non-vent fill to be questionable and use of a vent is recommended. Where receiver tank venting is accomplished during tank chilldown, it was found that the condition of the fluid actually being vented overboard had a significant effect on the vent quantity required. Also, this mass could be a significant proportion of the loaded mass (up to 50% for the conditions considered).

In any case, fluid inflow dynamics and heat transfer at low-g must be known in order to define the optimum fill method and its performance as to maximum pressure and/or quality of fluid vented. Development of a numerical technique based on the Marker-and-Cell (MAC) method is recommended for solving the cryogenic receiver liquid inflow problem. Subscale cryogenic 1-g and drop tower testing should also be performed and the data correlated with analytical models.

f. In the case of modular transfer, the main weight penalty for the individual modules is the requirement to insulate the supply for long term storage in the space station. This makes the use of small bottles unattractive where the boiloff must be restricted to 50% in 180 days. This penalty is significantly reduced where LO_2 and LN_2 are involved. Even with this insulation penalty the modular systems are the lowest weight and have the highest reliability where large quantities of propellant are to be transferred on a single tank to tank basis. This neglects the weight of remote manipulation systems required for modular transfer. Another important advantage of this system is that system maintenance would be simpler than other systems since the bottles would be periodically returned to earth. A disadvantage is that individual supply tanks are required for each receiver, resulting in high weight and cost when compared with subcritical systems using a single supply for a number of receivers. Use of a number of modules, however, does increase the ease with which redundancy can be incorporated into the transfer operation. The primary overall consideration is the development of a suitable cargo handling system and the crew participation involved. Satisfactory development of this cargo handling system would be a controlling factor in the final choice of modular versus subcritical means of transfer.

g. In all cases it was found that the fewer the number of bottles for a given fluid transfer requirement, the lower the weight, cost and crew requirements and the higher the reliability. Safety, with respect to separation of station fluids, and redundancy requirements would determine the minimum number of bottles which could actually be used.

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APPENDIX A

The data presented in this section were developed under the Convair Aerospace Independent Research and Development (IRAD) program and are reported herein for reference use in the current propellant transfer study.

General hardware data used in the overall analysis and screening of various transfer concepts are presented.

Propellant Pressure Vessel Weights

Data for 2219 aluminum and 347 stainless spheres are presented in Figures A-1 and A-2 respectively. The design allowable stresses used were 27,100 and 82,700 psi and densities are 0.102 and 0.286 lb/in^3. The above allowables are based on yields of 36,000 and 110,000 psi and a safety factor of 1.33. Weight estimates of weld lands, mounting pads and inflow and outflow provisions are based on the data presented in Ref. 5-9. Also, for convenience in determining spherical bottle volumes and surface areas as a function of diameter, the Figure A-3 and A-4 curves are included.

Vacuum Jacket

The wall thickness of a homogeneous spherical shell under external pressure, P_e psi is

$$t_j = C(P_e/E)^{1/2} D_j$$

$$W_j = t_j A_s \rho_j = \rho_j C_1 (14.7/E)^{1/2} D_j^3$$

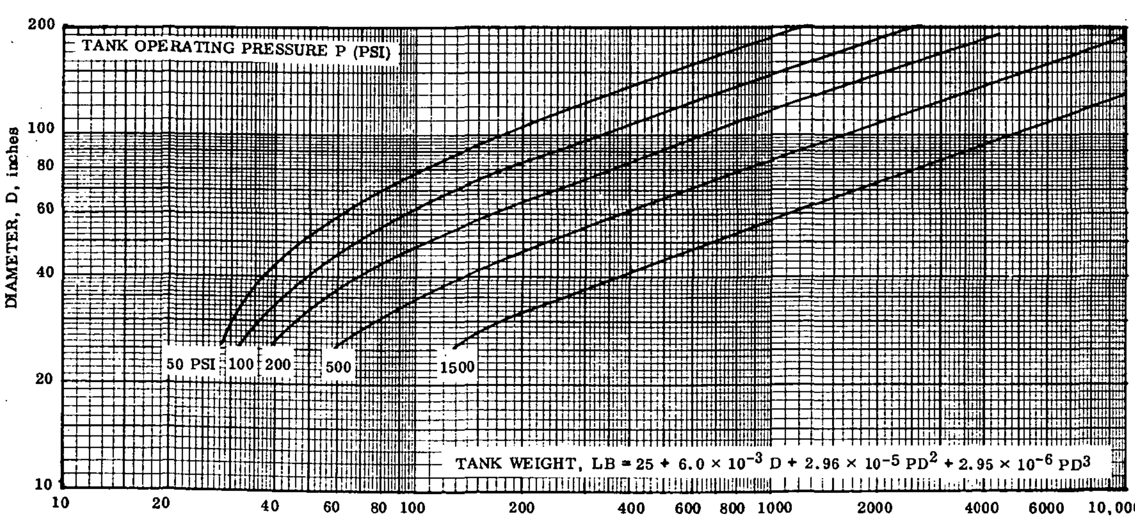

Figure A-1. Typical Pressure Vessel Weights for 2219 Aluminum Spheres

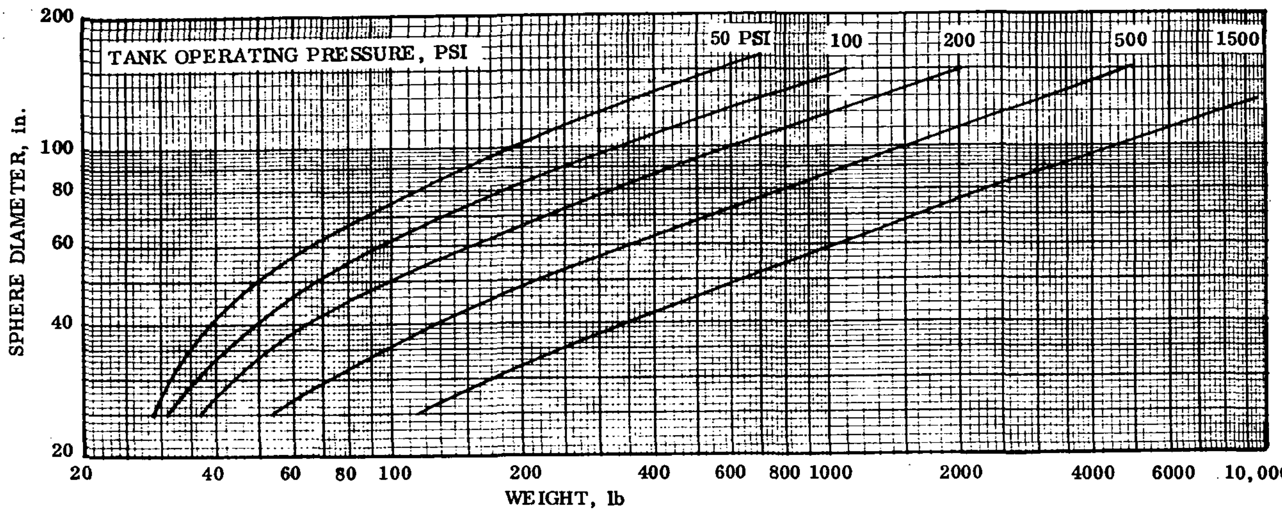

Figure A-2. Typical Pressure Vessel Weights for 347 CRES Spheres

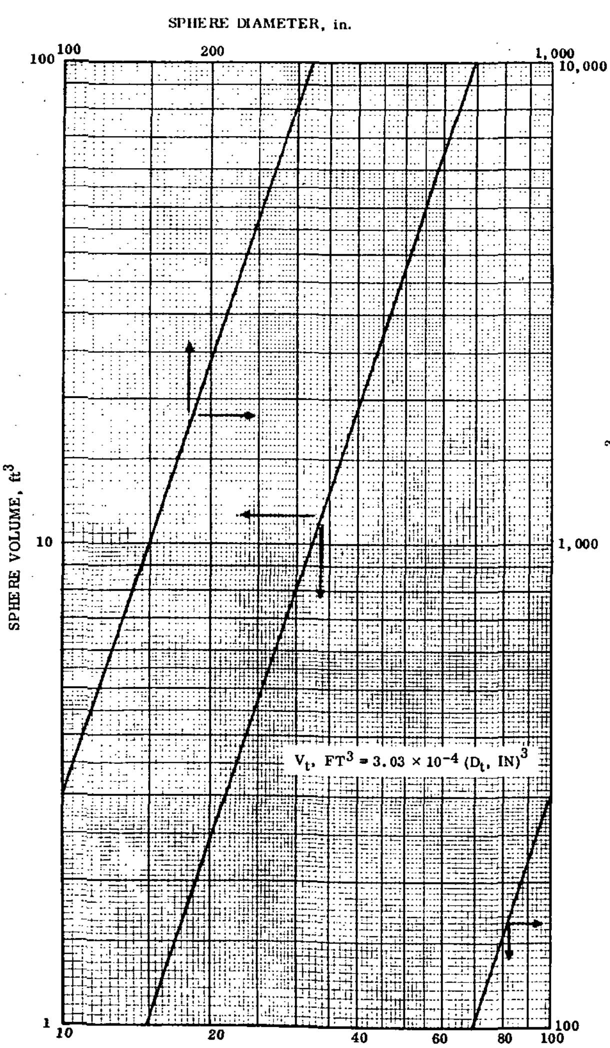

Figure A-3. Sphere Volume Vs Dia

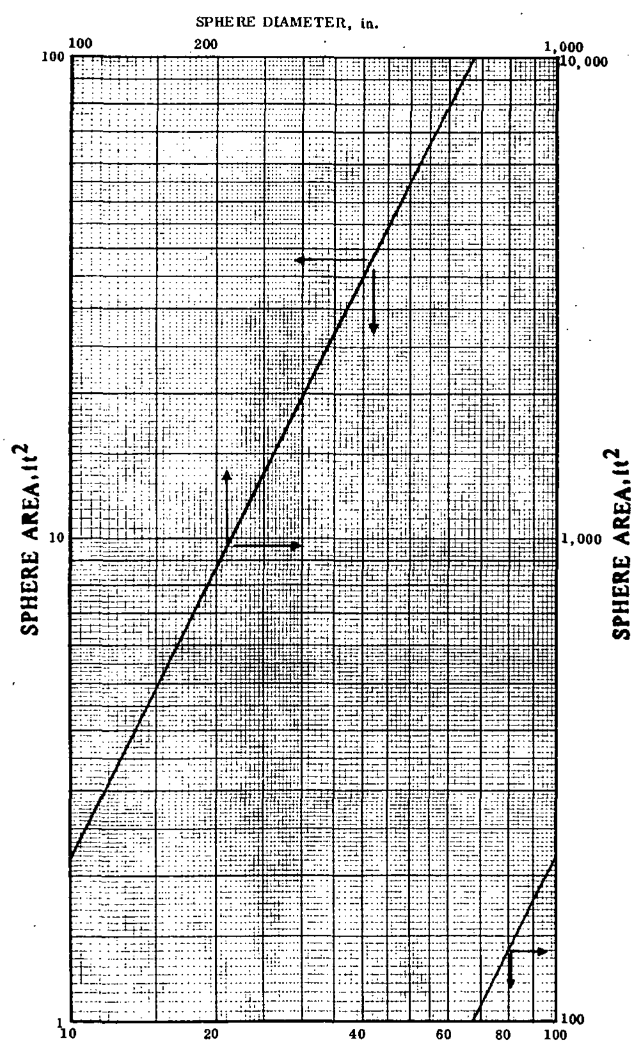

Figure A-4. Sphere Area Vs Dia

or

$$W_j = C_2 D_j^3$$

where

E = Young's modulus

C = constant

D_j = jacket diameter, in

W_j = jacket weight, lb

ρ_j = jacket material density, lb/in^3

However, for any but the smallest vacuum vessels, a homogeneous jacket is not weight-economical. Optimum design for bending rigidity, results in a composite structure of skin and stiffeners, or honeycomb sandwich, of a comparatively thick section composed of minimum gage materials. In this case the above equation reduces to

$$W_j = C_3 D_j^{C_4} \tag{A-1}$$

The empirical constants C_3 and C_4 differ widely from the values anticipated by any simplified theoretical approach. Individual weight data for the jacket are presented in Figure A-5. The spherical case is based on Equation A-1 with coefficients determined from the Reference 5-9 data such that $W_j = .35 D_j^{1.5}$.

Cylindrical data are ratioed to that of the sphere on the basis of surface area and a comparison of cylinder to sphere shell design criteria.

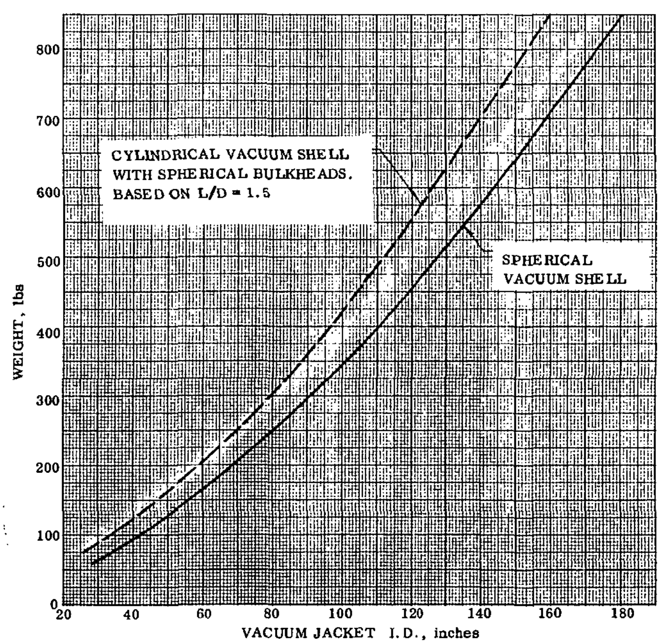

Figure A-5. Parametric Vacuum Jacket Weights

Helium Bottle

Weight data are given in Figure A-6 for Ti-6Al-4V high pressure bottles designed for room temperature operation with a design allowable stress of 95,700 psi and density of 0.160 lb/in^3. A 10% addition to the theoretical sphere weight was made for weldments and line and mounting bosses. The design allowable is based on a yield of 127,000 psi with a safety factor of 1.33.

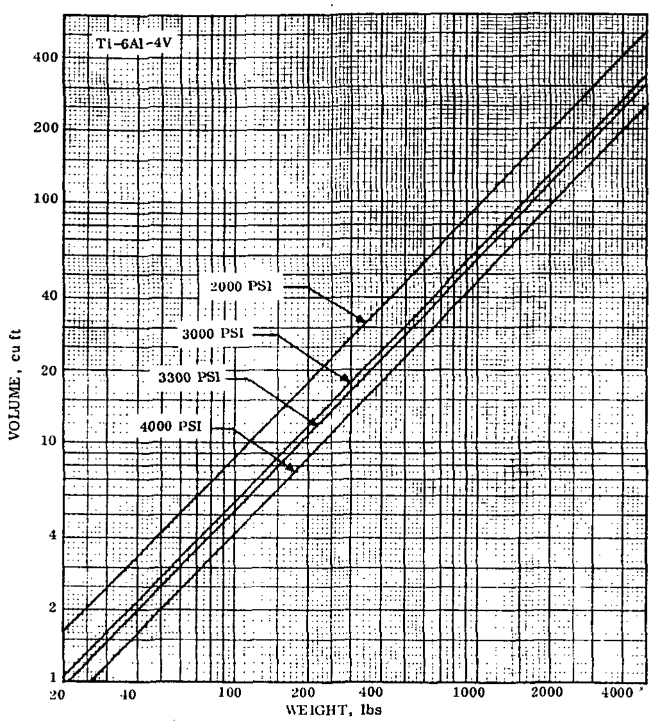

Figure A-6. Helium Pressure Bottle Weight as a Function of Volume for Titanium Bottles

Line Weights

Parametric line weights were defined for both Al Aly and CRES lines over a range of diameters from 0.5 to 3 inches and for operating pressures up to 2,000 psi. The total weight is taken to consist of the basic line plus mounting brackets and expansion loops or joints. Figures A-7 and A-8 present basic line weights for CRES and Al Aly ducting and Figure A-9 gives corresponding weights for attachments. Both weights are per foot of line length. Line expansion and contraction are assumed to be taken care of by providing loops in the line or gimbal type flex joints. Expansion loops will be used for line diameters of 1.0 inches and less and gimbal joints for diameters greater than 1.0 inch. The weights for each loop or joint are presented in Figure A-10 as a function of the basic line weight. The number of loops or joints required as a function of line length is presented in Figure A-11. With respect to the number and weight of gimbal joints, a range of values was designated and the actual quantity needed would depend on the specific routing configuration under consideration. The curve drawn represents an average of the best available data.

An example of the determination of the total line weight for an 0.5 inch by 100 ft Al Aly line operating with 160 psi is presented below.

From Figure A-8, $Wt_{line} = 100 \times 0.036 =$	3.60 lb
From Figure A-9, $Wt_{attach} = 100 \times 0.0027 =$	0.27 lb
From Figure A-11, Number of Loops = 4	
From Figure A-10, $Wt_{exp\ loops} = 4 \times 0.05 =$	0.20 lb
Total	4.07 lb

Valve Weights

Data are presented in Figures A-12 through A-14 based on a perusal of available vendor weight information. Actual data points are shown on the figures.

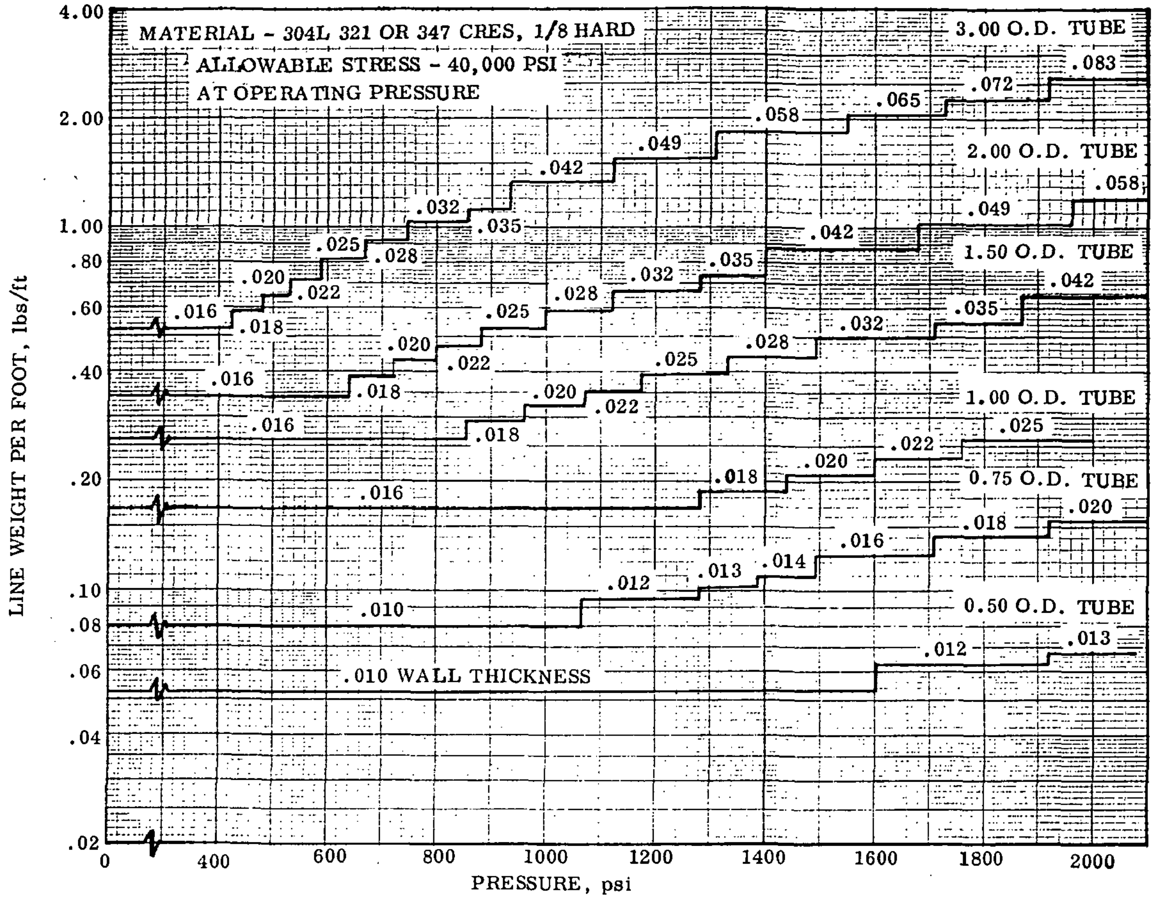

Figure A-7. Line Weight Versus Pressure for CRES Tubing

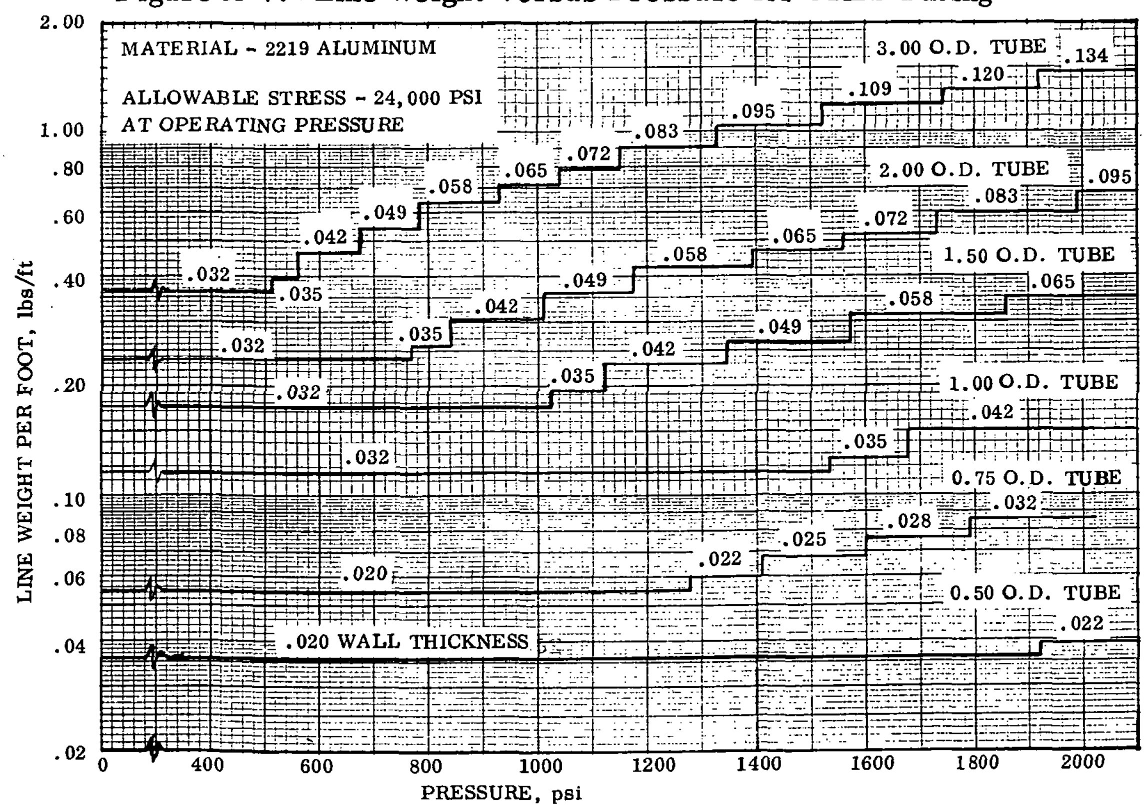

Figure A-8. Line Weight Versus Pressure for Aluminum Tubing

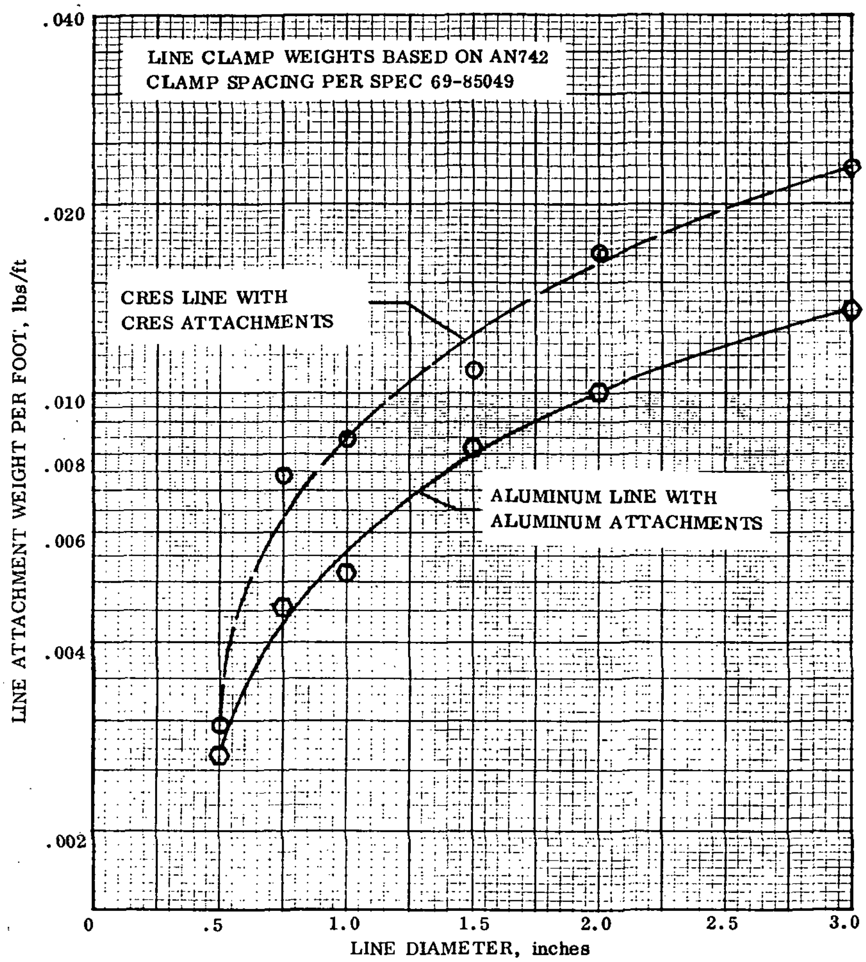

Figure A-9. Attachment Weight Vs Line Dia for CRES and Aluminum Tubine

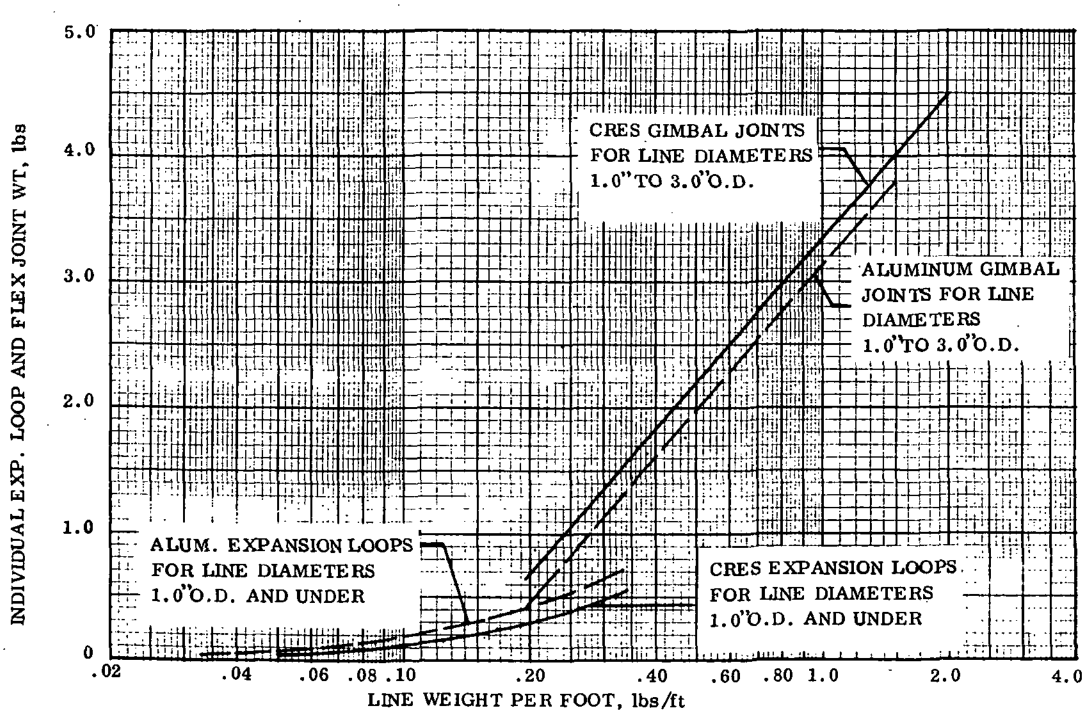

Figure A-10. Line Weight Versus Flex Joint Weight

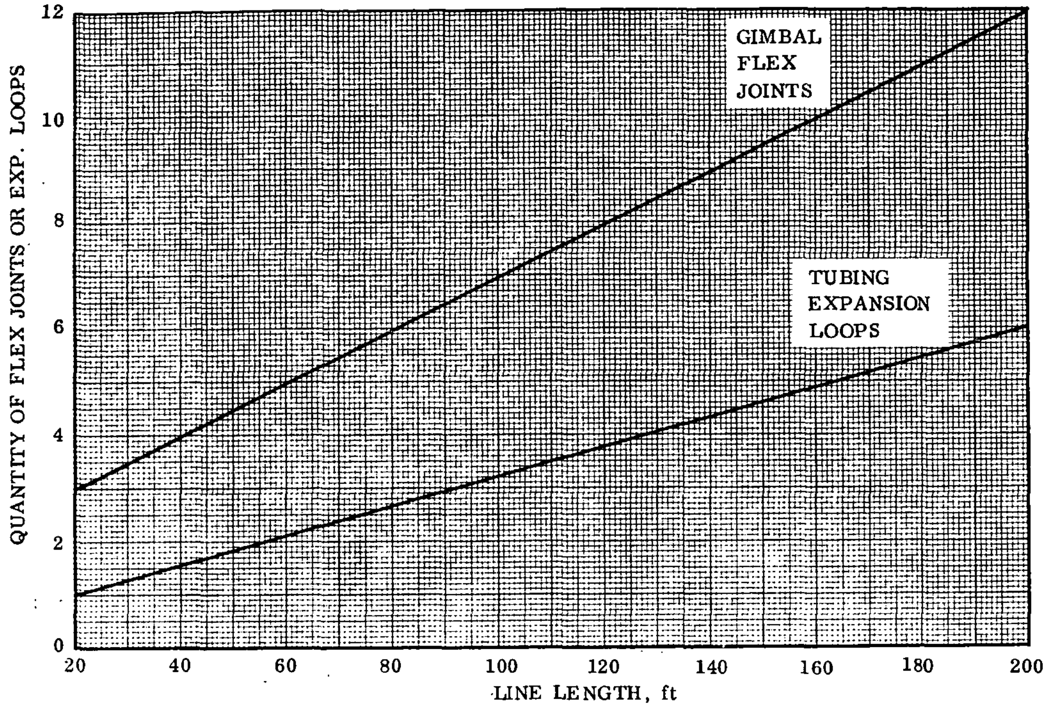

Figure A-11. Flex Joint Quantity Vs Line Length

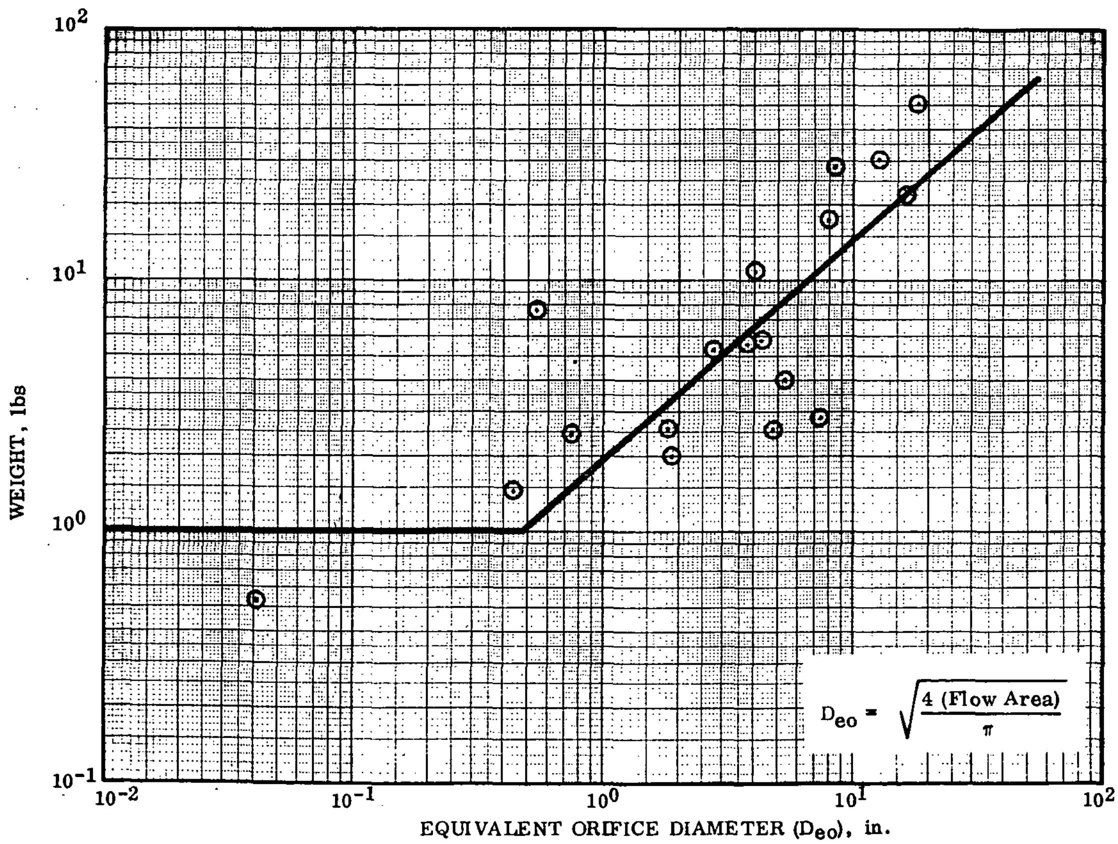

Figure A-12. Shutoff Valve Weight Data

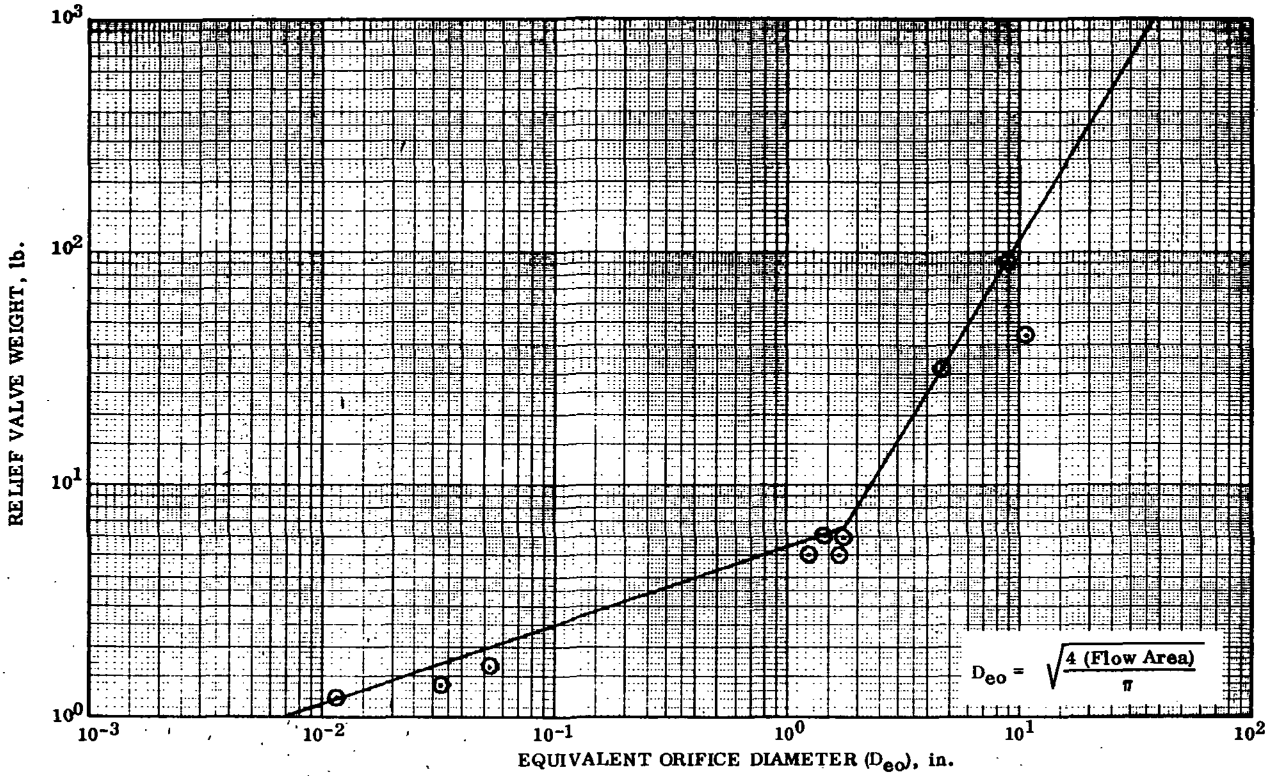

Figure A-13. Relief Valve Weight Data

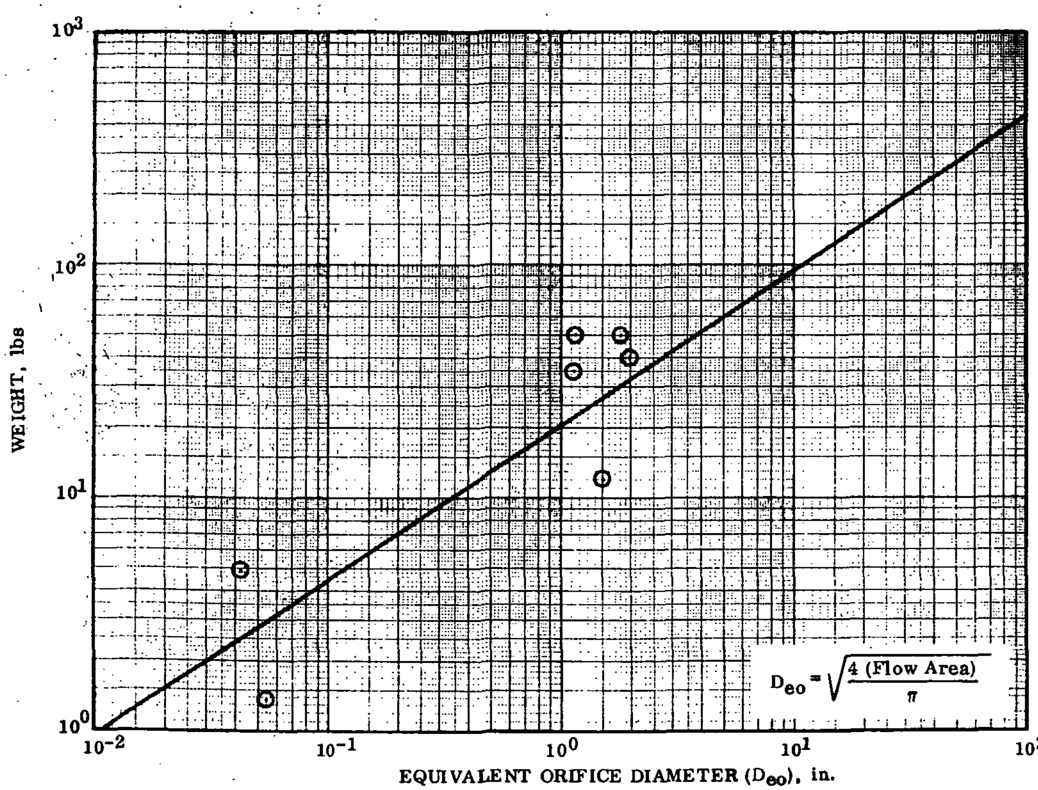

Figure A-14. Pressure Regulator Weight Data

APPENDIX B

HIGH PRESSURE TRANSFER ANALYTICAL DEVELOPMENT

The work presented in this section was performed under the Convair 1970 Independent Research and Development (IRAD) program and is reported herein for reference as it relates to the pertinent subject of cryogenic propellant transfer in space.

To replenish depleted supercritical cryogen tanks under low-g conditions one proposed method is to transfer the fluid under high pressure (supercritical) conditions. The collection and orientation problems peculiar to low-gravity liquid transfer are thus avoided. In addition, since the fluid is always single phase, analyses of the transfer process can be performed as if the fluid were gaseous even though its density will often be closer to the liquid state. This section describes the computer code PLUMBER which has been written to simulate high pressure systems for the purpose of evaluating this mode of low-g fluid transfer. Nomenclature used is presented in Paragraph B.6.

B.1 TANK THERMODYNAMICS

Considering the tank as a control volume the unsteady energy equation can be written

$$dU = (h + Ve^2/2g)\, dm + dQ \tag{B-1}$$

If the kinetic energy is negligible Equation B-1 can be written

$$dU = h_i\, dm_i - h_o\, dm_o + dQ \tag{B-2}$$

where the first two terms on the right-hand side of (B-2) represent the energy carried into and out of the control volume. A rearrangement of variables gives

$$m_t du = (u_i + p_i v_i - u_i)\, dm_i - (u_o + p_o v_o - u_t)\, dm_o + dQ \tag{B-3}$$

Since $u_i \neq u_t$, $u_o = u_t$, and

$$du = (\partial u/\partial p)_\rho\, dp + (\partial u/\partial \rho)_p\, d\rho \tag{B-4}$$

we can write (9-3), after dividing by dt, as

$$m_t \left(\frac{\partial u}{\partial p}\right)_\rho \frac{dp}{dt} + m_t \left(\frac{\partial u}{\partial \rho}\right)_p \frac{d\rho}{dt} = \left(u_i - u_t + p_i\, v_i\right) \dot{m}_i - p_o v_o \dot{m}_o + \dot{Q} \tag{B-5}$$

It is assumed that $a_t = a_o$ where a is any fluid property. Using this and the relation $\rho V = m$ we can collect terms to get

$$\rho_t V \left(\frac{\partial u}{\partial p}\right)_\rho \frac{dp}{dt} + \rho_t (\dot{m}_i - \dot{m}_o)\left(\frac{\partial u}{\partial \rho}\right)_p = (h_i - h_t)\, \dot{m}_i + p_t v_t (\dot{m}_i - \dot{m}_o) + \dot{Q} \qquad \text{(B-6)}$$

Now

$$p_t v_t = -\rho_t \left[\frac{\partial p_t v_t}{\partial \rho_t}\right]_{p_t}$$

so that the next to last term in(B-6) can be combined with the last term on the left-hand side to form a partial derivative of enthalpy. Since this derivative can be written

$$\left(\frac{\partial h}{\partial p}\right)_\rho = \left(\frac{\partial u}{\partial p}\right)_\rho + \left(\frac{\partial pv}{\partial p}\right)_\rho = \left(\frac{\partial u}{\partial p}\right)_\rho + \frac{1}{\rho} \qquad \text{(B-7)}$$

we have

$$\left[\rho_t \left(\frac{\partial h}{\partial p}\right)_\rho - 1\right] V \frac{\partial p}{\partial t} = \rho_t \left(\dot{m}_o - \dot{m}_i\right)\left(\frac{\partial h}{\partial \rho}\right)_p + \left(h_i - h_t\right) \dot{m}_i + \dot{Q} \qquad \text{(B-8)}$$

With the definitions

$$A = J\left(\frac{\partial h}{\partial p}\right)_\rho \qquad \text{(B-9)}$$

$$B = J\left(\frac{\partial h}{\partial \rho}\right)_p \qquad \text{(B-10)}$$

the general expression for pressure change in a tank with inflow and outflow is thus

$$\frac{\partial p}{\partial t} = \frac{\rho_t (\dot{m}_o - \dot{m}_i)\, B + J\,(h_i - h_t)\, \dot{m}_i + J\dot{Q}}{(\rho_t A - 1)\, V} \qquad \text{(B-11)}$$

Equivalently, the heat required to maintain constant pressure is

$$\dot{Q} = \rho_t (\dot{m}_i - \dot{m}_o)\, \frac{B}{J} + (h_t - h_i)\, \dot{m}_i \qquad \text{(B-12)}$$

B.2 PIPE FLOW EQUATIONS

The governing equations for one dimensional steady flow of a real, compressible gas in a constant area duct are listed below. (In this section u denotes the velocity and not internal energy)

$$dh + udu = dQ \qquad \text{(energy)} \qquad \text{(B-13)}$$

$$\rho udu + dp + \frac{1}{2} u^2 \rho \frac{fdx}{D} = 0 \qquad \text{(momentum)} \qquad \text{(B-14)}$$

$$ud\rho + \rho du \qquad \text{(continuity)} \qquad \text{(B-15)}$$

Choosing ρ and p as the independent thermodynamic variables we can write

$$dh = \left(\frac{\partial h}{\partial p}\right)_\rho dp + \left(\frac{\partial h}{\partial \rho}\right)_p d\rho \qquad \text{(B-16)}$$

or

$$dh = Adp + Bd\rho$$

Using this and dividing (B-13) by $p\rho u^2$, dividing (B-14) by pu^2, and dividing (B-15) by ρu^2 the three governing equations are

$$\left.\begin{aligned}
\frac{A}{\rho u^2}\frac{dp}{p} + \frac{B}{pu^2}\frac{d\rho}{\rho} + \frac{1}{2p\rho}\frac{du^2}{u^2} &= \frac{dQ}{p\rho u^2} \\
\frac{1}{u^2}\frac{dp}{p} &= \frac{\rho}{2p}\frac{fdx}{D} \\
\frac{d\rho}{\rho} + \frac{1}{2}\frac{du^2}{u^2} &= 0
\end{aligned}\right\} \qquad \text{(B-17)}$$

The system (B-17) can be solved by substitution for the variables dp/p, $d\rho/\rho$, and du^2/u^2 (Reference B-1).

If we define the dimensionless groups

$$\alpha \equiv \frac{\rho u^2}{2g_c p}$$

$$\psi^{-1} \equiv 1 - \rho A - \frac{g_c \rho B}{u^2}$$

$$F \equiv \frac{f\ell}{D}$$

$$\beta \equiv \psi \left[\rho A \frac{f\ell}{D} + \frac{2\,Jg_c dQ}{u^2}\right]$$

The formulas for the change of the three dependent variables are

$$\frac{dp}{p} = -\alpha(\beta + F) \tag{B-18}$$

$$\frac{d\rho}{\rho} = -\frac{1}{2}\beta \tag{B-19}$$

$$\frac{du^2}{u^2} = \beta \tag{B-20}$$

These relationships differ from those in Reference B-1 due to the fact that we cannot consider the working fluid as an ideal gas.

Knowing dQ and $f\ell/D$ and the fluid conditions at the entrance of the tube, the exit conditions can be computed by the finite-difference versions of equations B-18, B-19, and B-20, listed below.

$$p_{j+1} = p_j - \bar{\alpha}_j (\bar{\beta}_j + \bar{F}_j) \bar{p}_j \tag{B-21}$$

$$\rho_{j+1} = \rho_j - \frac{1}{2} \bar{\beta}_j \bar{\rho}_j \tag{B-22}$$

$$u_{j+1}^2 = u_j^2 + \bar{\beta}_j \bar{u}_j^2 \tag{B-23}$$

The bar superscript indicates an average between node j and j+1.

B.3 THERMODYNAMIC DATA

Supercritical pressure fluid transfer will occur at pressures and temperatures where the gas compressibility factor differs very seriously from 1.0. Thus use of the ideal gas relationships are most inappropriate. This is why the development in Sections B-1 and B-2 included the functions A and B instead of their relatively simple ideal gas counterparts.

Data tables of T, h, $(\partial h/\partial p)_v$, $(\partial h/\partial v)_p$, and c, with pressure and specific volume as the independent variables, were constructed for use in the program. Specific volume was used in place of density in development of the tables. This allows greater accuracy in linear interpolation between table values since the various properties such as temperature (T) and enthalpy (h) are more directly proportional to specific volume than to density. Conversion to values of ρ, A and B for actual program calculations is made using the relation of density to specific volume ($\rho = 1/v$).

The data tables range in pressure for H_2 from 10 psia to 5000 psia, for N_2 from 10 psia to 3000 psia, and for O_2 from 1 to 330 atmospheres. References B-2, B-3 and

B-4 supplied the basic data for the first four properties, and Reference B-4 was used for the speed of sound, c in O_2. Reference B-5 supplied the remaining basic data.

$(\partial h/\partial p)_v$ and $(\partial h/\partial v)_p$ values were computed by taking finite differences from the basic enthalpy data and the resulting data points did not always form smooth curves. The data were thus smoothed to correct this deficiency (caused by insufficient raw data from which to work) by a routine using a least squares fit to a parabola.

B.4 THE PLUMBER CODE

PLUMBER consists of a main program which controls the calculations, and several subroutines, each of which simulates a particular operation or piece of hardware. The subroutines are constructed so that the main program can be easily written to simulate a wide variety of tank and connecting line arrangements. A simplified flow diagram is shown in Figure B-1.

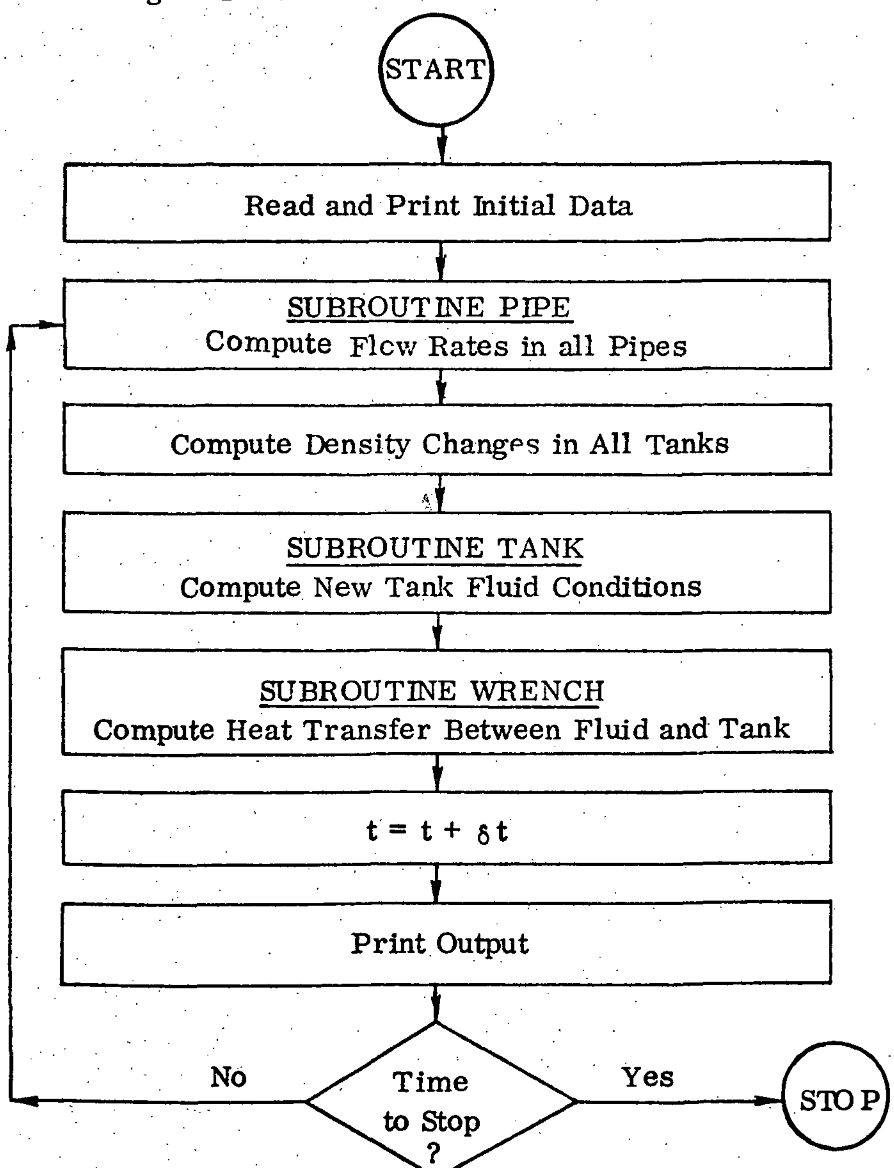

Figure B-1. Basic PLUMBER Flow Chart

B.4.1 SUBROUTINE TANK. This routine uses equations B-11 and B-12 to compute tank fluid properties. A code variable for each tank called KP is tested. If KP = 1 the routine uses an input heat rate and solves for a new bottle pressure using a finite difference version of equation B-11. If KP = 0 the routine calculates the amount of heat, per time step, required to maintain constant tank pressure using equation B-12. TANK is called separately for each tank, and any combination of KP codes can be used. Values of KP can be changed in the main program (PLUMBER) if desired. Properties used in these calculations are assumed to be constant over the time step.

B.4.2 SUBROUTINE PIPE. Given the inlet and exit pressures and inlet fluid conditions PIPE finds the mass flow rate by iteration. The user specifies the diameter and length of the tube and the number of finite difference sections desired; accuracy increases with the number of flow sections. Equations B-21, B-22 and B-23 are used to determine the fluid properties at each section. Newton's method is used to satisfy the equation

$$DP\,(\dot{m}) = 0 \tag{B-24}$$

in the usual manner

$$\dot{m}^{i+1} = \dot{m}^{i} - (DP/DP') \tag{B-25}$$

except that DP' is approximated by finite differences as

$$DP' \approx \frac{DP\,(\dot{m} + \delta\,\dot{m}) - DP\,(\dot{m})}{\delta\,\dot{m}} \tag{B-26}$$

where DP is defined as the error in pressure at the end of the pipe, and i is the iteration number. In reality more than one iteration is required for each i because convergence of $\overline{\rho}_i$ to $1/2\ (\rho_i + \rho_{i+1})$, for example, is not instantaneous, especially when the fluid velocities approach the speed of sound.

This procedure is for steady flow and the assumptions regarding its use in a time dependent calculation must be specified. In general, acceptable accuracy will result if the time steps are small compared with the time scale of tank property changes and if the time steps are large compared with the time scale of heat transfer into the pipe. Investigation of the effect of time step on program accuracy was accomplished for typical cases to aid in the overall analysis.

B.4.3 INPUT DESCRIPTION.

Card Type #1 (8A10)

80 column alphanumeric title used for problem identification.

Card Type #2 (2I10, E10.0)

ITUB = number of pipes in problem (10 maximum).
JBOT = number of tanks in problem (10 maximum).
DT = time step (seconds).

Card Type #3 (2I5, 7E10.0)

J = tank number. All entries on this card apply to tank J.
KP(J) = KP code:
 1 = pressures calculated in tanks for constant heat rate input
 0 = heat calculated for maintaining constant tank pressure
PB(J) = initial tank pressure (lb_f/ft^2 abs)
DB(J) = initial density (lb_m/ft^3)
VB(J) = tank volume (ft^3)
HEAT (J) = heat rate to tank fluid (BTU/sec)
HB(J) = heat transfer coefficient between tank wall and fluid (BTU/ sec-ft^2-°R)
AB(J) = inside surface area of tank (ft^2)
WB(J) = tank wall mass (lb_m)

NOTE: There will be JBOT cards of this type.

Card Type #4 (2I10, 4E10.0, 2I10)

I = pipe number. All entires on this card apply to pipe I.
NS (I) = number of flow sections (59 maximum)
MDØT(I) = iniital guess for mass flow rate (lb_m/sec)
L(I) = length of flow section (ft)
D(I) = diameter of pipe (ft)
DQ(I) = heat flow per section (Btu/sec-section)
JBIN(I) = number of tank supplying fluid to pipe I
JBØUT(I) = number of tank receiving fluid from pipe I

NOTE: There will be ITUB cards of this type.

B.4.4 OUTPUT DESCRIPTION. For each time step the following is printed:

Time (Seconds)

For each tank:

Pressure - lb_f/ft^2 abs

Density - lb_m/ft^3

Temperature of the fluid - °R

Heat - Btu (heat added is positive)

Tank metal temperature - °R

Heat transferred to fluid from tank metal - Btu (heat leaving metal is positive)

Mass of fluid in tank - lb_m

SUMH, integrated heat supplied to or removed from fluid - Btu

For each pipe:

Number of iterations

Mass flow rate - lb_m/sec

At each section:

Pressure - lb_f/ft^2 abs

Density - lb_m/ft^3

Velocity - ft/sec

Mach No. - dimensionless

Temperature - °R

B.5 SAMPLE PROBLEM

Use of the PLUMBER code is illustrated in the following problem: Two tanks, each with a volume of 42.5 ft^3, are connected by a 100 ft long 0.1 ft diameter pipe. The transfer line receives heat at 0.05 Btu/sec for each 1 ft long section. Pressure and density of hydrogen in the two tanks are 400 psia at 4 lb/ft^3 and 300 psia at 1 lb/ft^3.

Surface area and mass of both tanks are 59 ft^2 and 225 lbs. Heat between the tank wall and the bottle flows with a coefficient of 2 Btu/hr-ft^2-°R. Tank number 1 discharges fluid into transfer line number 1 and the tank pressure is to remain constant. Transfer line 1 empties into tank 2 where the heat rejected is to be zero and the pressure rise is to be computed.

The above conditions are supplied on data cards as described in Section B.4.3. To specify the working fluid the BLOCK DATA deck and subroutine PROP for the particular fluid must be used. Local problem controls are specified by coding in the main program. For this problem a request to stop execution when density for the fluid in tank no. 1 has fallen below 0.005 has been coded. In addition, it is desired that the pressure in tank no. 2 not rise above 300 psia.

When flow begins, cold gas from tank 1 is delivered to tank 2 and the pressure in the receiving tank drops. This causes a flow rate increase. At about 4 seconds the pressure in tank 2 starts to rise from a minimum of 270 psia and at 10 seconds has reached 300 psia. During this time about 35 lbs of fluid have been transferred and

7050 Btu have been added to tank no. 1. No heat has been removed from tank no. 2. At 58 seconds where the test case was arbitrarily stopped, 136 lbs of fluid have been transferred and 23,546 Btu have been supplied to the first tank to keep its pressure constant. To keep the second tank pressure from rising above 300 psia a total of 25,795 Btu have been removed. For this run the heat absorbed by the tank walls is small. Selected portions of the PLUMBER program output are shown in Figure B-2.

These results agree well with the data presented in Reference 4-2 and show that considerable heating and refrigeration is required to accomplish the transfer. With the high rate of transfer in this problem the heating rates needed are prohibitively high. An actual fluid transfer would thus be designed to take place more slowly, using smaller transfer lines, throttling valves or a smaller pressure difference between the tanks.

B.6 NOMENCLATURE

A	thermodynamic function - ft^3/lb_m
B	thermodynamic function - $ft^4\ lb_f/lb_m^2$
c	sound speed - ft/sec
D	pipe diameter - ft
dQ	heat - Btu/lb_m
dx	flow length - ft
f	friction factor - dimensionless
g_c	conversion factor from lb_m to lb_f ($= \frac{32.2\ lb_m\ ft}{lb_f\ sec}$)
h	enthalpy - Btu/lb_m
J	mechanical equivalent of heat (= 778 ft-lb_f/Btu)
m	mass - lb_m
$\dot{m}$	mass flow rate - lb_m/sec
p	pressure - lb_f/ft^2
Q	heat - Btu
$\dot{Q}$	heat rate - Btu/sec

T	temperature - °R
t	time - sec
U	internal energy - Btu
u	specific internal energy - Btu/lb
u	velocity - ft/sec
V	volume - ft^3
Ve	velocity - ft/sec
v	specific volume - ft^3/lb
μ	dynamic viscosity - lb_m/ft - sec
ρ	density - lb_m/ft^3
δt	time step - seconds

Subscripts

i	in
o	out
t	tank
j	denotes beginning of j^{th} pipe node

Figure B-2. Hydrogen Test Case, Supercritical Flow Between Two Tanks

ITUB = 1
JBOT = 2
DT = 2.0000E+00

BOT.	KP	PRESSURE	DENSITY	VOLUME	HEAT	HB	AB	WB
1	0	5.7600E+04	4.0000E+00	4.2500E+01	0.	5.5600E-04	5.9000E+01	2.2500E+02
2	1	4.3200E+04	1.0000E+00	4.2500E+01	0.	5.5600E-04	5.9000E+01	2.2500E+02

TUBE NO.	NS	MDOT	LENGTH	DIAM.	HEAT	JBIN	JBOUT
1	10	3.3500E+00	1.0000E+01	1.0000E-01	5.0000E-02	1	2

TIME = 0.0000 *************************

BOT.	KP	PRESSURE	DENSITY	TEMP	HEAT	TMET	QMET
1	0	5.7600E+04	4.0000E+00	5.1548E+01	0.	5.1548E+01	0.
2	1	4.3200E+04	1.0000E+00	7.8281E+01	0.	7.8281E+01	0.

BOTTLE NO.		MASS	SUMH
1		1.7000E+02	0.
2		4.2500E+01	0.
	1	3.350000E+00	2.147574E+02

TIME = 58.0000 *************************
MDOT = 1.383106E+00 ITER = 3

J	PRES	DENS	VEL	MACH	TEMP
1	5.760000E+04	8.370684E-01	2.103796E+02	9.291815E-02	1.043404E+02
2	5.641248E+04	8.203070E-01	2.146786E+02	9.519369E-02	1.039687E+02
3	5.519905E+04	8.030712E-01	2.192853E+02	9.765343E-02	1.035350E+02
4	5.395855E+04	7.855151E-01	2.241821E+02	1.001641E-01	1.034877E+02
5	5.269065E+04	7.679886E-01	2.292871E+02	1.028041E-01	1.034038E+02
6	5.139464E+04	7.503685E-01	2.346472E+02	1.056295E-01	1.031891E+02
7	5.006975E+04	7.325457E-01	2.403106E+02	1.086794E-01	1.028241E+02
8	4.871536E+04	7.144930E-01	2.463044E+02	1.119995E-01	1.022544E+02
9	4.733085E+04	6.961615E-01	2.526547E+02	1.155475E-01	1.016777E+02
10	4.591564E+04	6.774879E-01	2.594363E+02	1.192640E-01	1.014417E+02
11	4.446912E+04	6.583941E-01	2.666737E+02	1.232016E-01	1.010385E+02

BOT.	KP	PRESSURE	DENSITY	TEMP	HEAT	TMET	QMET
1	0	5.7600E+04	8.0452E-01	1.0656E+02	3.2557E+02	6.5074E+01	-1.3013E+00
2	0	4.4375E+04	4.1955E+00	4.5540E+01	-7.4658E+02	7.2464E+01	8.6535E-01

BOTTLE NO.	MASS	SUMH
1	3.4192E+01	2.3546E+04
2	1.7831E+02	-2.5795E+04

TUBE NO.	MDOT
1	1.383106E+00

GENERAL DYNAMICS

Convair Aerospace Division