CRYOGENIC STORAGE OF HELIUM FOR PROPELLANT TANK PRESSURIZATION

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SECTION I

INTRODUCTION

This paper has been prepared in response to many inquiries regarding the application of cryogenic helium storage to propellant tank pressurization systems. The high weight penalties associated with conventional helium storage systems have prompted the development of a Supercritical Helium Storage and Supply System for this application.

Storage of helium at extremely low temperatures and supercritical pressures yields fluid densities much greater than the liquid density. Figure 1 shows the range of densities possible with the supercritical helium concept. The fluid density selected for an optimized system is dependent upon the mission requirements. During both storage and fluid delivery, the helium is a single-phase fluid; thus, no two-phase problems are encountered during operation.

To date, missile and space vehicle propulsion systems have used either high-pressure gas storage or the more advantageous method of immersing the high-pressure storage vessel in a low-temperature propellant in order to increase densities and thus reduce weight penalties. This latter concept is limited, however, to applications in which a cryogenic fluid is utilized as the spacecraft propellant. Combining the advantages of both storage techniques, supercritical storage can provide the low tankage weight and volume possible with the cryogenic immersed-storage concept for applications where cryogenic propellants are not used or where immersion is not considered practical. Figure 2 provides a weight penalty comparison for the various helium storage systems.

Recent investigations have been made of helium storage requirements for cryogenic helium vessels leading to minimum weight and volume penalties. To maintain the required cryogenic helium temperatures, it is necessary to store the fluid in a vacuum-jacketed, superinsulated vessel that minimizes heat transfer from the ambient environment to the low-temperature helium. Recent state-of-the-art advances in flight-weight, high-performance cryogenic storage systems have made this achievement possible.

The concept described herein should thus be of interest to designers of propulsion systems for present and future spacecraft of extreme complexity for longer-duration missions, in which cryogenic propellants are not anticipated and the weight penalties associated with high-pressure gas storage are intolerable.

The balance of this report concerns a typical system and its operation. Emphasis is placed upon the fundamentals and techniques involved in design of a supercritical helium storage vessel.

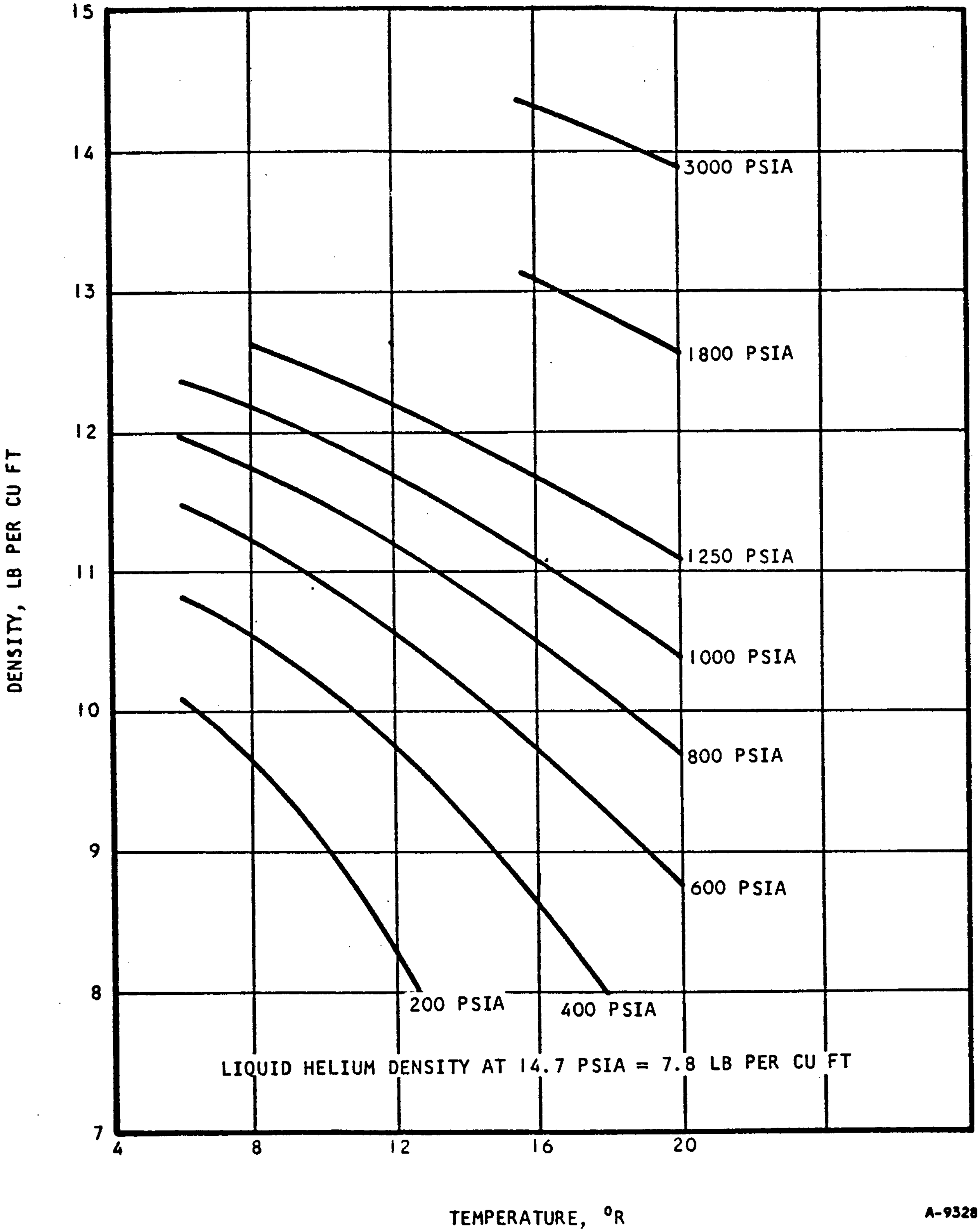

Figure 1. Helium Density

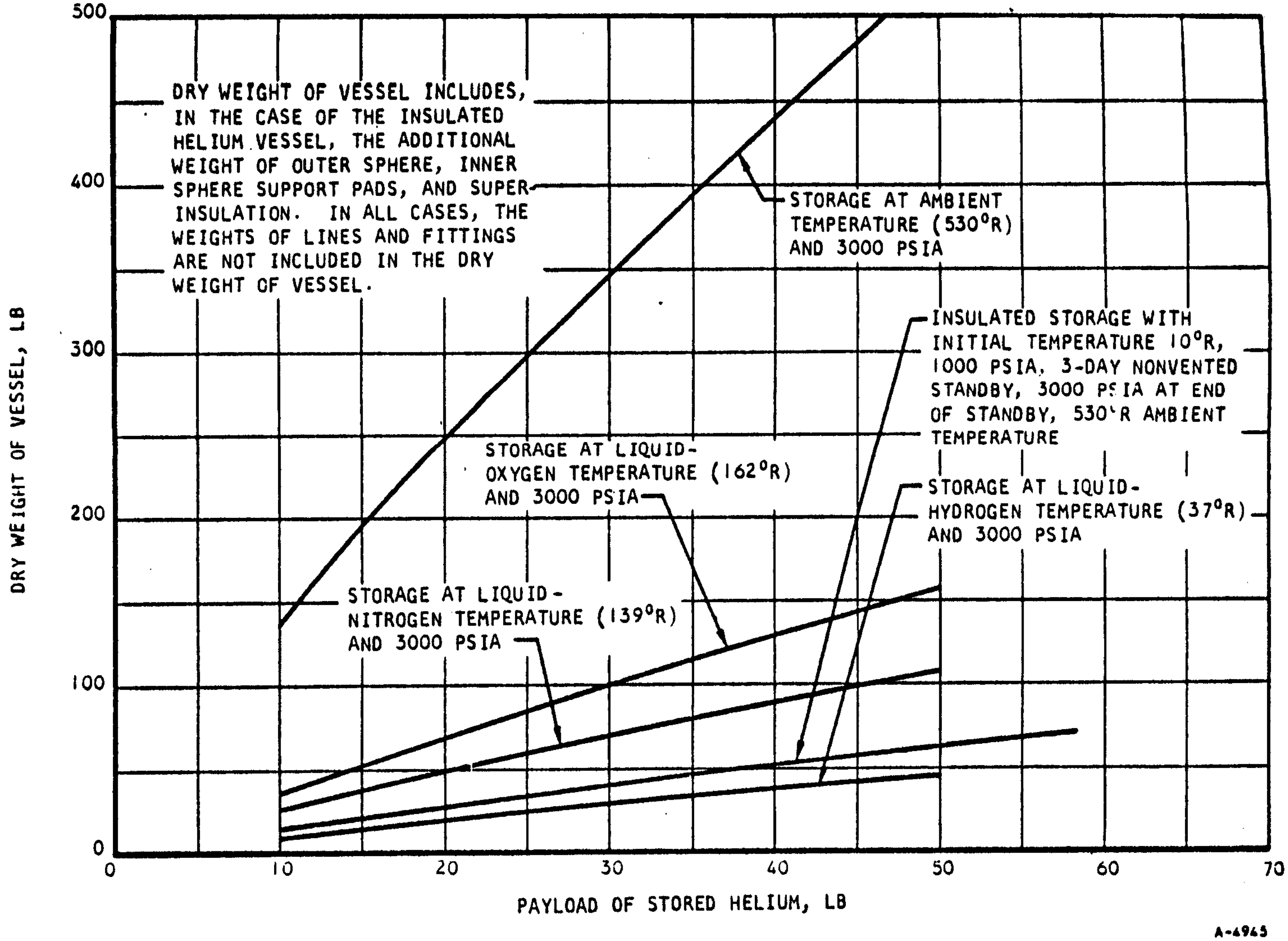

Figure 2. Dry Weight of Vessel vs Payload of Stored Helium for Various Storage Temperature Conditions

SECTION 2

DESCRIPTION AND OPERATION OF SUPERCRITICAL CRYOGENIC-HELIUM STORAGE VESSELS

DESCRIPTION

A simplified schematic of a typical insulated vessel for supercritical cryogenic helium storage is shown in Figure 3. The vessel consists of concentric, spherical inner and outer shells, each fabricated from titanium alloy hemispheres welded together in an inert gas atmosphere. Five semirigid fiberglass support pads serve to separate the inner from the outer sphere. The annulus between the inner and the outer spheres is filled with a superinsulating material such as aluminized Mylar and is evacuated to approximately 10^{-5} mm Hg. Although for simplification the connections between the inner and outer spheres are shown as straight lines in Figure 3, they would actually be routed through the insulation to give a long heat-leak path. The lines are fabricated from titanium, which has exceptionally low thermal conductivity. The internal heat exchanger, located inside the inner sphere, consists of a tube attached to a copper sphere, which provides the extended surface required for efficient heat transfer.

OPERATION

A specially designed ground fill system is required to fill the cryogenic storage vessel. The ground fill system is capable of supplying liquid helium and high-pressure gaseous helium at 10^{0}R or lower. Using the ground fill system, the flight vessel is first filled with liquid helium and then pressurized with 10^{0}R high-pressure helium to the predetermined design fill pressure; maximum density storage is obtained by this method. The temperature and pressure measurements will indicate when the fill density is attained.

When the fill is completed, the storage vessel enters the standby phase. Standby is defined as the time between vessel fill and when fluid delivery will first be required. During standby, ambient heat leak through the vessel insulation and connecting lines causes the fluid storage pressure to rise. When the maximum storage or vent pressure is attained i.e., 3200 psi, furthur pressure buildup is prevented by venting through the high-pressure relief valve.

Helium delivery is started by opening the solenoid-operated shutoff valve. As fluid is withdrawn from the storage vessel, it passes through the external heat exchanger (No. 1) and the helium temperature is raised to approximately 500^{0}R. The liquid propellant is an excellent heat source for this purpose. From the external heat exchanger, the warm helium enters the bypass control valve, where it is either diverted through the vessel internal heat exchanger or delivered to the supply line. The bypass control valve diverts the warm helium into the internal heat exchanger whenever the operating pressure drops below the operating pressure band. The warm helium gas flowing through the internal heat exchanger has the effect of warming the stored fluid in the

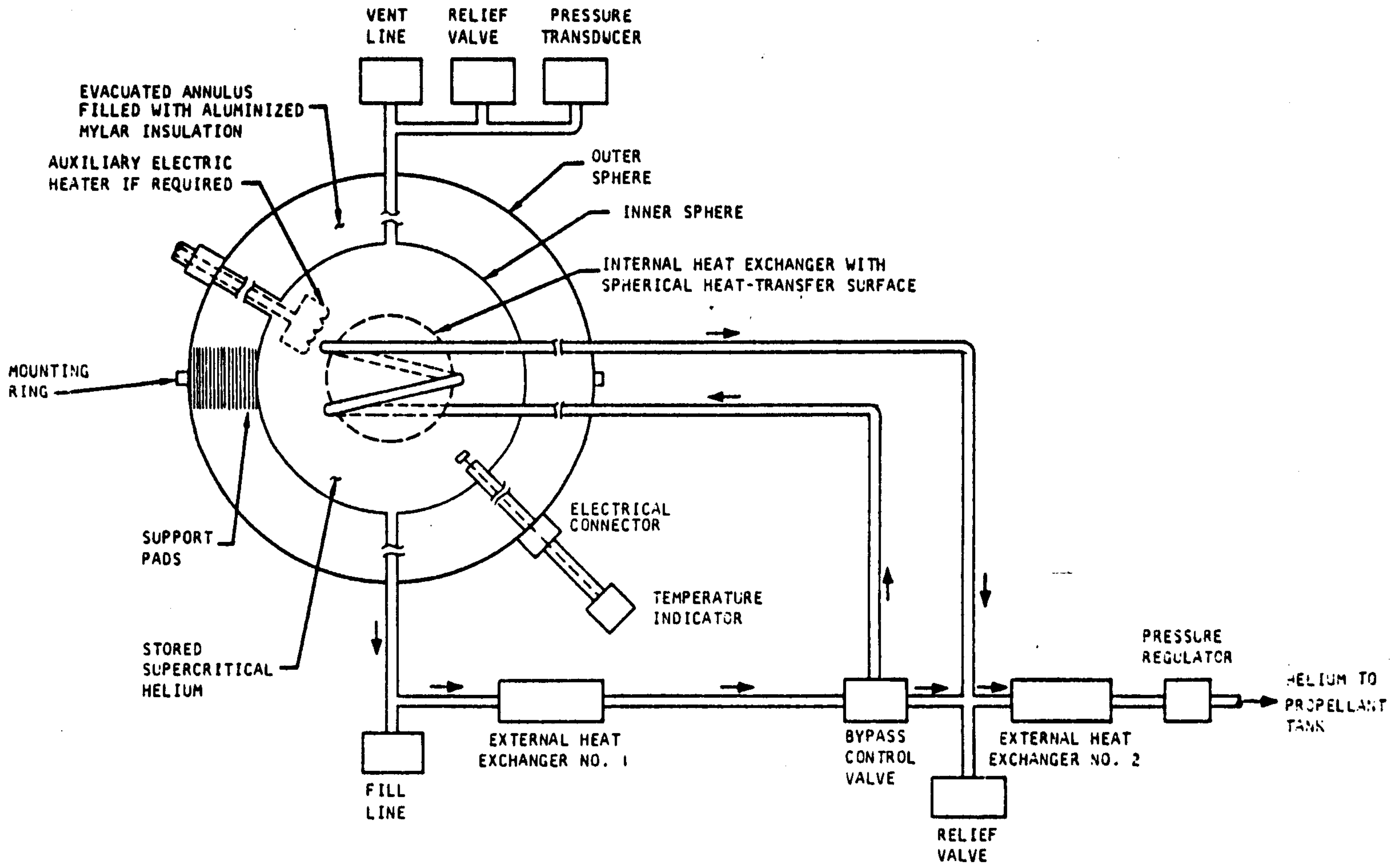

Figure 3. Simplified Schematic of Typical Supercritical-Helium Insulated Cryogenic Vessel

vessel and increasing the storage pressure. The cooled helium gas leaving the internal heat exchanger then passes through the external heat exchanger (No. 2), where it is again warmed to a suitable delivery temperature. Once the operating or storage pressure increases to the desired level, the bypass control valve directs the helium straight to the supply line, in certain designs, the bypass control valve can be eliminated, and all the delivery fluid is passed through the internal heat exchanger. The system pressure regulator controls the gas discharge pressure to the desired delivery pressure.

SECTION 3

TECHNICAL DISCUSSION

GENERAL

The design of a supercritical storage vessel falls into four general categories: thermodynamic design, heat-transfer considerations, structural and material considerations, and general design and weight optimization. Each of these areas must be carefully considered in order to design a minimum-weight, highly reliable pressurization system consistent with the most stringent design objectives.

This report considers only spherical vessels, because they represent the lightest configuration. The qualitative conclusions, however, are applicable to many other configurations.

THERMODYNAMIC DESIGN

The thermal design objective for a supercritical storage system is to provide an adequate insulation system that will enable the fluid to be stored for a specified standby period. Standby refers to the period after filling and capping, before delivery of the stored fluid is initiated. Two basic methods may be employed for accomplishing this objective.

The first method, nonvented standby, consists of filling the vessel with the amount of fluid required to accomplish the mission and designing the insulation so that the amount of heat transferred to the helium during the specified standby time does not exceed that amount which would cause the pressure to rise to the maximum allowable pressure before the pressurization system is operated.

In many instances where the standby time requirement is very long, it may be more feasible to fill the vessel with more fluid than would normally be necessary to meet mission requirements. The helium pressure is allowed to build up, and then the excess fluid is vented until the system is required for pressurization usage; this method is called vented standby. This concept should be used when the weight of the vented fluid combined with any resultant increase in tankage weight is less than the weight of the additional insulation that would be required to meet the nonvented standby requirement. The use of vented standby, with the associated possibility of using the vented fluid to remove heat that is being transmitted through the insulation by means of a vapor cooled shield, will not be discussed further in this report. This method should be considered, however, when it is desirable to provide cryogenic helium storage for long-duration standby.

Fundamental Thermodynamic Equations

The thermodynamic characteristics and the resultant heat input and insulation requirements are determined by application of the first law of thermodynamics. For this development, the thermodynamic system considered consists of the mass of stored fluid at any time. The first law as applied to this system can be written as:

$$d\underline{E} = \underline{q} - \underline{w} + E_a \, dm \tag{1}$$

where $\underline{E}$ = total internal energy of the fluid in the vessel

$\underline{q}$ = infinitesimal quantity of heat input to the system

$\underline{w}$ = infinitesimal quantity of work

E_a = internal energy of the exit stream

m = total stored mass

Differentiation of the relation between enthalpy and internal energy results in:

$$d\underline{E} = d\underline{H} - Pd\underline{V} - \underline{V}dP \tag{2}$$

where $\underline{H}$ = total enthalpy

$\underline{V}$ = total volume

P = vessel pressure

The work term in Equation (1) is given by:

$$\underline{w} = Pd\underline{V} - P\,V_a \, dm \tag{3}$$

where $Pd\underline{V}$ = the work performed by the stored fluid due to a volume change.

$PV_a \, dm$ = the work performed on the fluid exiting from the vessel

Subscript "a" = the fluid properties of the exit stream in the event they vary from the state of the stored fluid mass

Combining Equations (1) and (3) results in

$$\underline{q} = d\underline{H} - \underline{V}dP - H_a dm \tag{4}$$

This equation can be integrated for evaluation of finite changes. For the case of a constant-volume storage vessel, the integrated form is

$$\underline{Q} = \underline{H}_2 - \underline{H}_1 - \underline{V}(P_2 - P_1) - \int_{m_1}^{m_2} H_a \, dm \tag{5}$$

where $\underline{Q}$ = the quantity of total heat input over the finite time interval.

Equation (5) is the form of the first law that is most useful for design purposes.

Basic Considerations

For the case of nonvented standby, the mass of the stored fluid does not change, and Equation (5) becomes

$$\underline{Q} = \underline{H}_2 - \underline{H}_1 - \underline{V}(P_2 - P_1) \tag{6}$$

Due to the relationship between internal energy and enthalpy ($\underline{E} = \underline{H} - P\underline{V}$), this equation can be written as

$$\underline{Q} = \underline{E}_2 - \underline{E}_1 \tag{7}$$

When put on a differential rate basis, Equation (7), can be written as

$$d\theta = \frac{d\underline{E}}{\dot{\underline{Q}}} \tag{8}$$

which, upon integration, yields

$$\theta = \int_{\underline{E}_1}^{\underline{E}_2} \frac{d\underline{E}}{\dot{\underline{Q}}} \tag{9}$$

where θ = the standby time

$\dot{\underline{Q}}$ = the ambient heat leak rate

To perform the indicated integration, the rate of heat transfer to the cryogen $\dot{\underline{Q}}$ must be expressed as a function of the internal energy of the stored fluid. Thus, in principle, a definite tank geometry must be considered and a transient heat transfer analysis conducted to compute the standby time.

In practice, this transient analysis is unnecessary for two reasons. First, the temperature change of the stored fluid during pressure buildup is moderate. Second, due to the characteristics of the terms which constitute $\dot{\underline{Q}}$, the heat transfer rate is only slightly affected by these moderate fluid temperature changes. Taking $\dot{\underline{Q}}$ as an average value, the standby time is given by:

$$\theta = \frac{\underline{E}_2 - \underline{E}_1}{\dot{\underline{Q}}}$$

The amount of thermal protection or insulation required to meet a specified standby time is a function of the initial fill state, the final state at the end of the standby period, and the quantity of fluid contained in the vessel. The total weight of fluid stored differs from the usable weight by the weight of residual fluid remaining in the tank at the end of the delivery period. The determination of residual will be discussed later. In order to maintain a common basis of comparison, all systems discussed in this report, except where specifically stated, will be designated for an initial fluid density of 11.4 lb per cu ft and an initial fill pressure of 1000 psia. The maximum operating pressure will be considered to be 3000 psia. Equation (6) can then be used to determine the total amount of heat transfer allowed, so that the standby time requirement can be met.

The actual fluid-state path followed during withdrawal is dependent on the initial pressure, the energy input to the system, and the flow schedule required to pressurize the propellant. Equation (5) can be used and integrated over the flow rate schedule to solve for the pressure, internal temperature, and time history of the fluid. From this relationship, it can be seen that the residual fluid remaining in the vessel may be calculated as a function of energy input to the fluid. Figure 4 shows a typical withdrawal path for a specified condition. This curve is presented as a typical example and should be used only as an indication of the behavior that may be expected from a supercritical helium system during fluid delivery. It is only after detailed mission requirements are analyzed that an optimum system may be designed to accomplish a particular end application.

HEAT-TRANSFER CONSIDERATIONS

The various modes of heat transfer to the inner vessel and important design features are discussed below. The insulation used is part of a weight-optimized system and must have high thermal efficiency and low weight. The type of insulation that has been shown to be consistent with these requirements, and is often referred to as superinsulation, consists of multiple layers of highly reflective radiation shields separated by a low-conductivity material.

The two promising kinds of superinsulation are: (1) layers of aluminized polyester film and (2) layers of aluminum foil separated by fiberglass paper. The low-conductivity material is the polyester film in the first, and the fiberglass paper in the second. Both of these insulation systems must be used in conjunction with a vacuum level of at least 10^{-4} mm Hg in order to minimize residual gas conduction. Under these circumstances, apparent thermal conductivities of these materials have been found by measurement to have ideal values of approximately 4×10^{-5} Btu-ft per hr-ft^2-^{0}R. This apparent thermal conductivity includes heat-transfer effects due to radiation, conduction, and convection (residual gas conduction). With an assumed thermal conductivity of 7.5×10^{-5} Btu per hr-ft-^{0}R, Figure 5 shows the heat-transfer rate through the insulation as a function of the thickness for various vessel diameters.

Heat transfers through lines penetrating the annulus and the intravessel supports also contribute to heat transfer to the helium. These heat leaks can be calculated by Fourier's equation of heat conduction.

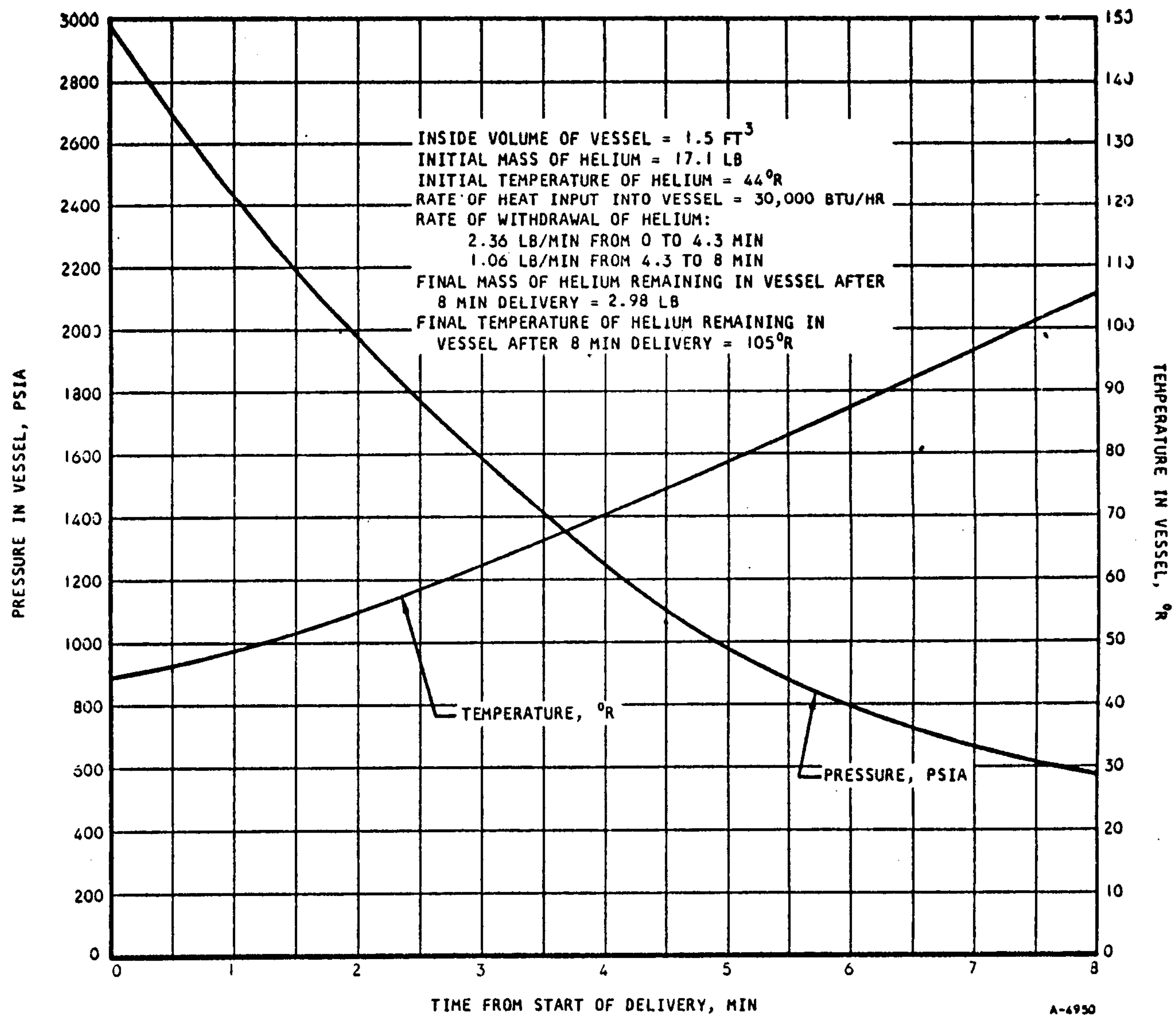

Figure 4. Typical Helium-Vessel Pressure and Temperature vs Time for Rapid Delivery Lasting 8 Min

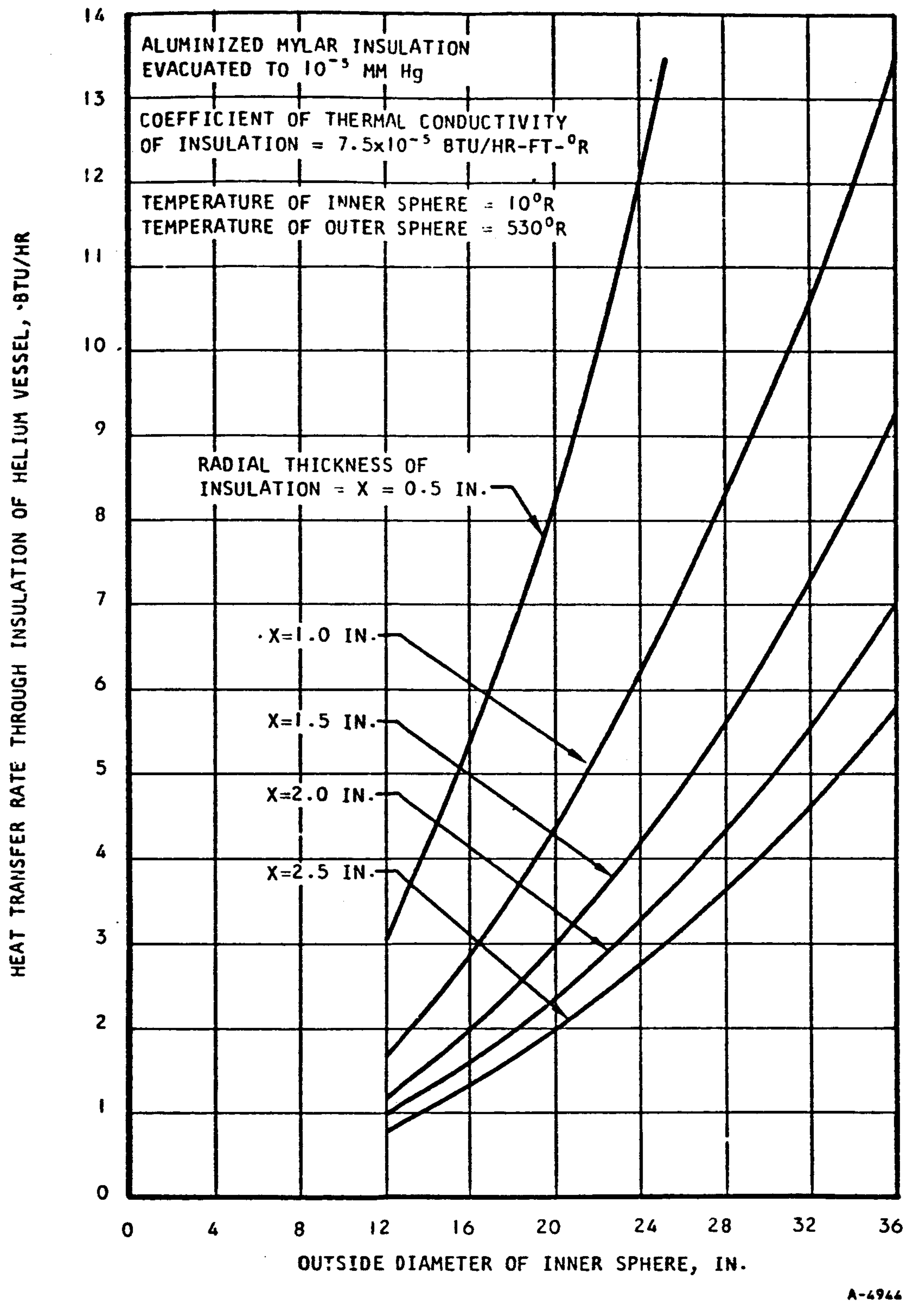

Figure 5. Heat-Transfer Rate Through Insulation of Helium Vessel vs Outside Diameter of Inner Sphere for Various Thicknesses of Evacuated Aluminized Mylar Insulation

As was stated in the previous section, fluid expulsion requires the simultaneous addition of energy. This addition of energy may be accomplished by electrical heat input, by simultaneous electrical heating combined with fluid-to-fluid heat transfer, or by fluid-to-fluid heat transfer alone.

The flow rates usually required for a pressurization system are generally so large that electrical power penalties preclude the use of electrical heating alone as the energy source. Fluid-to-fluid heat transfer is generally the most acceptable method for fluid expulsion. This method consists of the following steps: (1) fluid withdrawal is initiated and the fluid is diverted through an external heat exchanger, which transfers heat to the fluid; (2) the fluid is run back through a heat exchanger that is located inside the storage vessel; (3) the helium then leaves the storage vessel and flows into the propellant tank to initiate pressurization. Waste heat from the spacecraft or from some other heat source must be provided for the external warmup heat exchanger.

VESSEL DESIGN

Typical storage vessels for the supercritical helium consist of two concentric spherical shells separated by an evacuated insulation space. The inner vessel contains the cryogenic fluid at operational pressures and temperatures, and the outer vessel is exposed to ambient pressure and temperature.

An intershell support system is required to transmit loads between the shells. The design of the intershell mounts is extremely critical because a large percentage of the vessel heat leak is normally carried through these supports by direct conduction. The inherent difficulty is that the thermal insulation requirements demand minimal contact area between the tanks, yet large structural loads must be transmitted through the same supports. In addition, appreciable changes in radial clearance between the shells will occur because of the large temperature difference between the inner and outer tanks when cryogenic fluid is stored in the inner vessel. Despite the change in radial clearance, the intershell supports must continue to hold the inner shell rigidly in place without introducing appreciable local stresses. This recurring problem in the design of high-performance cryogenic tanks has led to the development of numerous support devices, ranging from stacked insulated washers to support rings on the inner and outer shells that are connected by rods, wires, or metal bands. These supports have the common disadvantages of intricate fabrication procedures, and transmittal and amplification of dynamic loads to the inner shells.

A novel method of support has already proved successful under severe vibration, shock, and thermal load conditions associated with the Project Gemini program. The annular space is largely filled with a non-load-carrying superinsulation. Support of the inner shell is accomplished by the use of equally spaced, compressed fiberglass pads. With the proper choice of pad density, size, and location, a suitable support can be designed to hold the inner shell for the various environmental shock and vibration loadings which may be encountered. The compressibility of the fiberglass support pads is sufficient to easily absorb the radial-clearance changes due to thermal effects without loss in holding effect. Precompression of the pads results in an energized (self-strained) structural system. For a given vibratory loading, less cyclic deflection and, consequently, less incremental force is experienced by the supports. The resonant frequency of the inner shell mounting can be tuned by varying the stiffness or the compressibility of the fiberglass support pads; the maximum amplification factor is 2.0 to 3.0 at resonance. The inner shell structure then is isolated from the high g-level vibratory inputs at the high frequencies. Conversely, loadings due to the support pads on the outer shell support structure are materially less than those associated with more rigid inner-shell support methods.

Reinforcement rings are unnecessary for the inner shells. The pad loading on the inner shell is much lower than internal pressure loading, and the influence of pad loading tends to reduce membrane stresses in the shell with little or no bending. Pad precompression loads are designed so that they are low enough to preclude inner shell buckling when the inner vessel is not pressurized.

MATERIAL CONSIDERATIONS

Besides the tensile strength-to-density ratio, the problem of low-temperature brittleness is of critical importance in the selection of materials for cryogenic applications. There are several methods for evaluation of material toughness, often referred to in terms of resistance to brittle fracture (or notch sensitivity); these include determinations of tensile elongation value, notched-to-unnotched tensile strength ratio, energy required to initiate and propagate a crack, charpy V-notch tests, etc. In general, a combination of tensile elongation and notched-to-unnotched tensile strength ratio provides the information for the selection of the most suitable material.

After extensive studies, the titanium alloys Ti-TAl-4V ELI and Ti-5Al-2.5 Sn ELI appear best suited for the inner shells, since they have the highest strength-to-density ratios at cryogenic temperatures. For the outer shell, either titanium or aluminum are considered the best metals. Titanium is generally selected because of better fatigue characteristics and manufacturing compatibility.

WEIGHT OPTIMIZATION

To arrive at the minimum weight design for a supercritical cryogenic tank, it is necessary to investigate all the environmental and performance requirements, the various combinations of design parameters which will meet these requirements, and the effect of these combinations of design parameters on the total tank weight. The requirements fall into the following categories:

a. Usable fluid weight

b. Standby time

c. Buildup time

d. Delivery pressures

e. Vibration levels

f. Acceleration levels

g. Shock levels

h. Temperature levels

i. Electrical power characteristics

In designing to meet these requirements, the following general parameters can be varied:

a. Fill pressure

b. Residual density

c. Fluid weight vented

d. Percent of initial fill

e. Vent pressure

f. Minimum operating pressure

g. Internal heater on and off pressure

h. Pressure decay path

Selection of a combination of these parameters will define the details of a tank design which includes the items which contribute to the weight. These details will include:

a. Fluid fill weight

b. Internal heater and quantity gauge configuration

c. Inner shell diameter and thickness

d. Interwall support configuration

e. Interwall line configurations

f. Insulation configuration

g. Outer shell diameter and thickness

h. External support configuration

There is a complex relationship between the tank requirements, the general design parameters, and the design details. Generally speaking, a change in one variable will result in changes in many of the design details, and a change in one of the design details will affect conformance to a number of the requirements.

Because these quantities involve such complex interrelationships, a meaningful optimization procedure must treat upon all of them simultaneously. If it were possible to establish an overall functional relationship between the filled tank weight and all the variables, then they could be treated simultaneously by analytical means with LaGrange multipliers. It has not been found possible to obtain such an expression without making unrealistic assumptions about its form, so that the analysis ceases to have practical value. An alternate method is available which permits these variable to be considered simultaneously without this disadvantage.

The method utilized is based on the use of digital computer techniques, rather than analytical techniques. A program is used that selects the combination of design parameters which result in the lowest total weight, without assuming any parametric relationships between the various quantities. This is accomplished by first determining the total weight for the tank corresponding to selected combinations of design parameters which satisfy the tank requirements and then selecting the combination which yields the lowest total weight. In arriving at these weights, the program considers the fluid thermodynamics, the system heat transfer, the stress analysis of the shells and supports, and the constraints which manufacturing considerations place upon the design. The combinations of design parameters are chosen to cover a range sufficient including all the combinations which might yield the minimum weight tank.

The sequence of operations involved in this program is as follows: A set of design parameters is chosen based on a system which increments the parameters one at a time in a manner to assure that all possible combinations will occur. With this set of parameters, the volume and weight of the fluid fill can be derived. The weight and volume of the capacitance gauge and internal heat transfer surface can then be found. The diameter thickness, and weight of the inner shell can then be determined. These items represent all the weight supported by the fiberglass pads; therefore, the required pad area can then be found.

The allowable heat transfer rate during standby can be obtained from the required standby time, fluid properties, and design parameters being used. With this information, plus the pad area and inner vessel dimension, it is possible to determine the insulation thickness needed to meet the standby requirement. The outer shell diameter is then fixed, and its thickness and weight can be calculated. At this point, the total weight supported by the external ring is known, and the ring size and weight can be found. This defines a tank which will satisfy the standby requirements. The predicted performance of this tank is then compared to the requirements imposed by the mission flow profile. If it does not satisfy these requirements, the program returns to the appropriate point in the previous calculation and redesigns the tank so as to meet these requirements. When a tank has been found that satisfies all of the requirements, its total weight is compared to the total weight of the tanks calculated for different combinations of design parameters. The design parameters for the lightest three tanks are stored, and the next set of design parameters is chosen; the whole process is then repeated. When all the combinations of design parameters have been considered, the characteristics of the lightest three tanks are printed out. The number of combinations considered can run into the thousands for a complex set of requirements.

Once this minimum weight configuration has been established, the effects of variations in the design parameters can be investigated to determine the weight penalty associated with these changes. This permits tradeoffs involving factors, other than tank weight, to be made in a quantitative manner.