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ENERGY DEPOT CRYOGENIC FUEL
STORAGE AND DISTRIBUTION
SYSTEM

PHASE I - CONCEPTUAL DESIGN STUDY

FINAL TECHNICAL REPORT

by

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August 1964

MASTER

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RESEARCH AND DEVELOPMENT DEPARTMENT
Allentown, Pennsylvania

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FOREWORD

This report is the Final Technical Report for the work performed by Air Products and Chemicals, Inc. under USAEC Contract AT(30-1)-3129, Energy Depot Cryogenic Fuel Storage and Distribution System: Phase I - Conceptual Design Study. The report, supplemented by three previously issued Quarterly Progress Reports, covers the reporting period March 15, 1963 to May 15, 1964.

Responsibility for the performance of this contract has rested with Dr. C. McKinley, Director of Research and Development Department, and was delegated to Dr. A. Lapin, Manager, Cryogenic Engineering Section. Mr. R. G. Clark, as Project Manager, was responsible for performing the work statement of the contract. The work was administered under the direction of the United States Atomic Energy Commission New York Operations Office with Mr. Gilbert Silverman as Project Engineer.

The authors wish to thank their many colleagues who contributed to the performance of this contract and to the preparation of this report. The authors are especially grateful to the following people for their contributions:

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ABSTRACT

This Final Technical Report summarizes the studies performed by Air Products and Chemicals, Inc. from March 15, 1963, to May 15, 1964, the duration of USAEC Contract No. AT(30-1)-3129, Energy Depot Cryogenic Fuel Storage and Distribution System: Phase I - Conceptual Design Study. The conceptual design of the system for distributing liquid hydrogen produced by the Nuclear Powered Energy Depot to fuel users (e.g. vehicles) is presented. Also reported are studies for feasibility demonstration of an advanced design concept for lightweight, compact, low heat leak, and rugged 50-gallon liquid hydrogen vessels for the Energy Depot System.

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I. INTRODUCTION

The Energy Depot is a concept for increasing the mobility of highly mechanized armies of the future. The present concept involves: (1) a readily transportable source of power; (2) readily transportable equipment for transforming feed materials (preferably air and water) into fuels and oxidizers; (3) equipment for storing and distributing the fuels and oxidizers to points of use; and (4) means for utilizing the fuels and oxidizers for vehicular propulsion, cooking, space heating, and the generation of electric power. The development of such a concept is motivated by a desire to eliminate the presently complex and vulnerable logistic operations by which fuels are supplied to a foreign military theater from the continental United States. With Energy Depot, highly mechanized military forces can become more maneuverable because the fuel source is itself readily transportable.

In 1962 both Allis-Chalmers Manufacturing Co. (with Air Products and Chemicals, Inc. as its subcontractor) and the Allison Division of General Motors Company completed Energy Depot technical feasibility studies(1,2) during which present and anticipated technology of a wide variety of power sources, fuel-oxidant combinations, and fuel utilization schemes were surveyed and analyzed. One of the most promising system concepts evolved during these studies consists of:

1. The production of electrical power by a nuclear fission reactor, e.g. the MCR-type (Military Compact Reactor);
2. The production of hydrogen by the electrolysis of water;
3. The liquefaction of the hydrogen;
4. The bulk storage and distribution of the hydrogen to points of use; and
5. The utilization of the hydrogen in a hydrogen/air fuel cell to power electric motors for vehicle propulsion or to operate other electrical equipment.

The present contract is considered to be the first phase of a continuing program for the development of the lightweight, rugged, efficient, reliable, and highly automatic system for the storage and distribution of liquid hydrogen by military personnel under field conditions. The present effort was initiated in recognition of the fact that the presently available technology of containing, storing, and transferring liquid hydrogen is not sufficiently advanced for the Energy Depot application. The object of

this effort has been (1) to advance the technology of lightweight, low heat leak, compact, and rugged 50-gallon liquid hydrogen vessels, a size representative of that required for small vehicles (e.g. jeeps); and (2) to achieve a conceptual design of the over-all hydrogen storage and distribution system for the Energy Depot. The activities were performed as the following five technical work packages(3):

Work Package I - 50-Gallon Vessel Conceptual Design Study

This effort consisted of (1) a survey and analysis of criteria for the Energy Depot in general and the Cryogenic Fuel Storage and Distribution System in particular; (2) a thermodynamic and structural analysis of the containment of liquid hydrogen; (3) the collection and evaluation of the properties of various vessel, support, and insulation materials of interest for this application; and (4) the selection of the most promising vessel, support, and insulation materials and vessel design concepts for further study.

Work Package II - Materials Study

During this work package, materials information and fabrication techniques which were not readily available in the literature and which are required for the effective utilization of the selected materials were obtained. The studies were to establish one of the candidate inner vessel materials as most promising, to identify a suitable welding transition piece, to establish fabrication techniques for the outer shell, and to select the application technique for the insulation.

Work Package III - Boiler-Plate 50-Gallon Vessel Test

The effort under this work package was to fabricate and test a 50-gallon vessel to establish the validity of the thermal protection system which was conceptually designed as Work Package I. No attempt was made to minimize the weight of the shells of the test vessel, which was considered to be a full-scale thermodynamic mock-up of the lightweight 50-gallon test vessel studied as Work Package IV.

Work Package IV - Lightweight 50-Gallon Vessel Test

This effort involved the fabrication and testing of a 50-gallon liquid hydrogen vessel designed for both low weight and low liquid boiloff. This vessel was considered to be the lightweight counterpart of the engineering model tested as Work Package III and was used to demonstrate the feasibility of the design concepts selected earlier in the program for combining the features of low weight, compactness, low heat leak, and ruggedness into a single vessel.

Work Package V - Conceptual Design of the Liquid Hydrogen Storage and Distribution System

This effort consisted of studies of the over-all fuel storage and distribution system for Energy Depot. System concepts were evaluated and the most promising approach was selected. Safe operating and maintenance procedures were outlined.

Section II, which follows this introduction, consists of a summary of the pertinent results of the fuel storage and distribution system study and the 50-gallon vessel study. General recommendations for future studies to culminate in the field acceptance testing of a prototype Energy Depot Fuel Storage and Distribution System are presented as Section III. In Section IV, the conceptual design of the liquid hydrogen storage and distribution system is presented. The conceptual design of an advanced 50-gallon vessel is described in Section V, while experimental studies of an advanced 50-gallon vessel are described in Section VI. The various appendices contain detailed supporting information. Information presented in the three previously submitted Quarterly Technical Reports^(4,5,6) has been used freely throughout this document.

II. SUMMARY OF RESULTS

A. LIQUID HYDROGEN STORAGE AND DISTRIBUTION SYSTEM CONCEPTUAL DESIGN

The purpose of the Energy Depot storage and distribution system is to distribute the fuel from the production depot to the device which converts this fuel into useful energy. For the hydrogen-fueled Energy Depot, the storage and distribution system will begin at the product manifold of the hydrogen liquefier and will extend to the inlet manifold of the energy utilization device (fuel cell, internal combustion engine, space heater, etc.). For the purpose of this study, the utilization device has been assumed to be a bank of fuel cells on board a vehicle. The energy obtained from the hydrogen/air fuel cells will operate electric motors for vehicular propulsion.

The fuel system has been designed to provide: (1) both long-term and short-term storage of the hydrogen delivered from the liquefier; and (2) fuel distribution to and storage at points of use. The system will be composed of two major components identified as the primary and the secondary units. The primary unit is to provide bulk storage at the fuel depot and bulk fuel distribution to secondary units, e.g. user vehicles. The secondary unit will involve the fuel storage vessels and handling equipment required for operation of the unit. The secondary storage capacity will be at least an order of magnitude less than that of the primary unit.

Fuel transfer operations will be highly automated and will require a minimum of operator activity or attention. Because much of the system efficiency and safety have been designed into the control and transfer equipment, all fuel transfers can be performed by a single operator with only limited specialized training. The main valves of the primary and secondary systems will be highly reliable pneumatic control valves. For the primary unit, manual by-pass valves will parallel the two main control valves. Provisions will be made for interconnecting the air and electrical systems of adjacent vehicles. These features will provide sufficient system redundancy to allow: (1) the delivery of the full fuel load from a primary unit which is without air or electrical power or which has damaged automatic control valves; or (2) the refueling of a powerless secondary vehicle.

For the purpose of this study, the primary unit was considered to be a modified 8-ton GOER, an all-purpose transport vehicle being developed by the Army. Figure 1 is a layout of the primary unit. The GOER has been considered to be a flat-bed truck which will accommodate the 3200-gallon primary storage vessel and the associated equipment.

REVISIONS							
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FIGURE 1

REQ'D	PART NO	NOMENCLATURE		MAT'L SPEC.	UNIT	FINISH	ZONE
UNLESS OTHERWISE SPECIFIED DIM S ARE IN INCHES FRACTION DECIMAL ANGLES ± ± ±		DRAFTSMAN MCKLOIBER CHECKED J. Schumacher ENGINEER J. H. H. g. w.	DATE 2-6-64 3/5/64 3/5/64	TITLE CONCEPTUAL LAYOUT PRIMARY LIQUID HYDROGEN STORAGE & DISTRIBUTION UNIT: 8-TON GOER		AIR PRODUCTS AND CHEMICALS, INC. ALLENTOWN, PENNSYLVANIA U S. A.	
FIRST USED ON		APPROVED A. J. Kahl 3-5-64		SCALE		WT.	DWG SIZE 67589 D

The control console from which the automatic transfer operations can be initiated will be located at the left rear of the vehicle. Access to the system controller for servicing will be through the small door in the left side of the vehicle behind the rear wheels. The vaporization coil will be located under the small diameter section of the vessel on the bed of the GOER. The hose compartment is shown at the right rear of the vehicle. The transfer piping and valving will be located within the enclosed space behind and below the storage vessel. The two square plates represented between the control panel and the hose chamber can be removed by loosening wing-nuts to yield access to the manual by-pass valves in case of failure of the valves or automatic control system. The vent is represented on the left side of the vessel ahead of the wheels.

The M-113 Armored Personnel Carrier has served as the representative model of a secondary unit during this study. The conceptual layout of that vehicle is presented as Figure 2. The general location and arrangement of the fuel storage vessels within the vehicle was established under another contract and has been previously reported(7). According to this arrangement, 240-gallons of liquid hydrogen will be stored in two locations, one along each side of the vehicle inside the armor. Again, for space economy, two 60-gallon vessels, one on top of the other, are to be located along each side. Vent piping will be located above the vessels. The opening of the vent is not shown in Figure 2 but will be essentially flush with the top of the vehicle at a location above the safety and relief devices. During filling of the secondary unit, this vent pipe will be extended to about 15 feet from the ground to increase the safety for personnel in and around the vehicle. The secondary vessels will be filled through a connection located behind a small door in the armor at the left rear of the vehicle. The piping interconnecting the right-hand and left-hand tanks, the heater and its piping, and the safety and relief devices are to be located forward of the vessels. The fuel system controller, occupying about 1.5 ft³ is not shown in the figure but will be located conveniently. The control console will be located on the dashboard in front to the vehicle operator.

Table I presents the estimated weight distribution for the fuel handling components on board the primary and secondary units. The weight of the fuel system required for operation of the GOER is not included in the tabulation for the primary system. The weights presented for the secondary vessels, piping and valves are based on the multiple, long, slender vessel storage arrangement shown in Figure 2. Considerable weight reductions are possible if a single, more-nearly spherical storage vessel is utilized.

SECTION A-A

FIGURE 2

WKL/DER DRAFTSMAN <i>J. J. H. H. H.</i> CHECKED <i>J. J. H. H. H.</i> ENGINEER <i>J. J. H. H. H.</i> APPROVED	DATE 2-11-64 3/9/64 3/6/64 3-5-64	TITLE CONCEPTUAL LAYOUT TYPICAL SECONDARY LIQUID HYDROGEN USER VEHICLE: M-113 ARMORED PERSONNEL CARRIER SCALE 1" = 1'-0" WT.	AIR PRODUCTS & CHEMICALS INC. ALLENTOWN, PENNSYLVANIA U. S. A. DWG. No. 67588 D
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TABLE I

ESTIMATED WEIGHT DISTRIBUTION FOR
THE PRIMARY AND SECONDARY UNITS

<u>Component</u>	<u>Weight (lb)</u>	
	<u>Primary</u>	<u>Secondary</u>
Vessel(s) and Supports	1640	425
Piping	250	525
Valves	180	200
System Controller and Control Console	70	40
Electric Vaporizer		75
Liquid Hydrogen	<u>1950</u>	<u>142</u>
TOTAL WEIGHT	4090	1407

The "normal" operation mode of the primary unit will involve short-term fuel storage, i.e. the primary unit will be filled and will proceed to rendezvous with fuel users. However, a single manual adjustment of one of the valves in the primary vent system will prepare the system for efficient long-term storage. The fuel system storage and transfer efficiency, defined as the fraction of the fuel produced which is deliverable to the fuel utilization device, will be relatively high. For short-term storage in which the hydrogen is used by a secondary vehicle about one day after production, the estimated transfer efficiency is 93%. For a representative fuel stockpiling operation involving one week of fuel storage in the primary unit, followed by transfer to and one week of storage in secondary units, the estimated system transfer efficiency is 80%. It must be made clear that these estimates are not based on the four, long, slender 60-gallon vessel arrangement shown in Figure 2 for the M-113. A more nearly optimum configuration involving a single, cylindrical 240 gallon vessel of length-to diameter ratio (L/D) of 2 was assumed as an approximation of the configuration to be actually employed. If the split-tank configuration illustrated in Figure 2 were employed, the estimated fuel storage and transfer efficiency for the stockpiling operation would be about 55%.

The system is designed for the safe handling of liquid hydrogen. The system is considered safe for two reasons. First, the properties and behavior of hydrogen are well known and understood. Secondly, the

system and its components have been conceptually designed on the basis of this knowledge to achieve a low probability of fuel ignition and minimal effects of ignition should it occur. As stated previously, much of the system safety and efficiency has been designed into the components so that the system will be operable under field conditions by personnel with only limited specialized training. For instance, the couplings for the transfer hose will be of the quick-connect, double-block-and-bleed-type to eliminate routine purging of the transfer lines and will allow the operator to quickly and safely connect and disconnect a hose containing hydrogen. The degree of safety achieved with this system will be commensurate with that obtained in commercial liquid hydrogen practice and will be very high compared with other hazards existing during military operations.

The development of the liquid hydrogen storage and distribution system is considered to be entirely feasible. The system will exhibit low specific weight and volume, a high degree of automation, ease of operation, inherent safety, high fluid storage and transfer efficiency, versatility as to storage time, and reliability. A few of the key components required for this system are not available under existing technology and must be developed. These are:

- . A highly flexible lightweight transfer hose capable of repeated flexing while cold;
- . A reliable quick-connect cryogenic coupling of the double-block-and bleed-type;
- . Low weight liquid hydrogen storage vessels which also possess low volume, low heat leak, and ruggedness;
- . Highly reliable lightweight, pneumatically operated cryogenic control valves with specific design features (e.g. internal bleed; and
- . Highly reliable, compact control system for fully automatic fuel transfer under field conditions.

It is further proposed that an engineering study be performed to select the optimum design pressures for the primary and secondary vessels for low vessel and piping weights, for high system storage and transfer efficiency, and as a function of ullage, based on the following items:

- . A variable-pressure vent relief valve, the relief pressure for which is a function of the liquid level;
- . Minimum metal thicknesses to be considered for the fabrication of the inner shells for the smaller capacity vessels;
- . The potential of chemically milling for the inner shells for small vessels to achieve weight reduction;

- . The interrelationships between system weight, system transfer efficiency, and the system design pressures; and
- . The design of a vent system for an airplane to allow safe air transport of hydrogen-fueled vehicles with a minimum of fuel loss due to venting at high altitudes.

The conceptual design discussed above is based on hardware technology which can exist after several years of directed component development. The Army may have a need to become familiar with and to experiment with the handling of liquid hydrogen in support of hydrogen-fueled vehicles during the period of development of the advanced system. As part of the system study it was determined that an interim system, based strictly on presently available technology, can be built to adequately serve the short-term, experimental needs of the Army. The interim system will be similar in all major respects to the conceptual system. For instance, the interim system will include a primary bulk storage unit (e.g. an over-the-road liquid hydrogen trailer) and a secondary storage unit (vessels and associated components built into an experimental vehicle). The degree of safety for the interim system will be as high as for the conceptual system. Because of the limitation to presently available hardware technology, the interim system will not possess certain advanced characteristics such as light weight, high component reliability, and extreme simplicity of operation.

From an operation standpoint, the interim system will differ from the conceptual system in the following respects:

- . The interim system will require more manual control for the transfer operation.
- . For the interim system, purging of the transfer lines with an inert gas before and after transfer will be required. This is necessary because present commercial hose couplings do not incorporate the double-block-and-bleed feature. Without purging, flammable air-hydrogen mixtures would exist in the transfer lines.
- . The transfer hose of the interim system will be flexible only when it is warm. For this reason the hose must be allowed to warm before it can be stored following a fuel transfer.

Tabel II is included for general comparison of the conceptual design and the interim design of the liquid hydrogen storage and distribution systems with respect to the availability of equipment.

TABLE II

COMPARISON OF THE CONCEPTUAL AND THE INTERIM
DESIGNS FOR THE LIQUID HYDROGEN STORAGE AND
DISTRIBUTION SYSTEM

<u>Component</u>	<u>Conceptual Design</u>	<u>Interim Design</u>
Vessels	Development required to reduce weight and volume	Commercially available but are heavy and bulky
Valves	Development required to provide internal bleeds, reliability, and low weight	Commercially available but are heavy and bulky
Transfer Hose	Development required to achieve high degree of cold flexibility	Commercially available but are rigid when cold
Hose Coupling	Development required for reliable quick-connect, double-block-and-bleed couplings	Commercially available but are not quick-connect or self sealing - Transfer line purge with inert gas required before and after transfer
Piping	Development required for the use of new materials for weight reduction	Commercially available but heavy
Instrumentation	Development required for fully automatic reliable system	Commercially available but not proven - Manual operation may be required

B. 50-GALLON VESSEL STUDY

Conceptual design and experimental studies were conducted for a 50-gallon liquid hydrogen vessel for Energy Depot. The 50-gallon size was selected as representative for small secondary vehicles (jeeps, trucks, etc.). Figure 3 is a schematic of the vessel design which has evolved from these studies. The vessel design involves: (1) an inner shell of aluminum alloy 2219; (2) multiple-radiation-shields-and-vacuum insulation; and (3) an outer vacuum jacket of aluminum honeycomb. The inner vessel is to be supported on a central

FIGURE 3

Air Products and Chemicals
INC.
ALLENTOWN, PENNSYLVANIA, U S A

SCHEMATIC REPRESENTATION
OF VESSEL DESIGN

DRAFTSMAN *D J Buss*
ENGINEER *P Stoltz*
DATE 3/20/64
SCALE NONE

DWG. No

67590 C

pipe or mandrel. The radial and rotational position of the mandrel will be fixed by a trunnion-type support arrangement at each end. Longitudinal support and positioning will be provided by an invar rod through the mandrel. All internal piping will enter the inner vessel through the center of the mandrel which will provide the only two penetrations of the insulation. This design is based on gaseous hydrogen delivery, the gas to be generated by an internal electric heater.

The principle design objective was to achieve low weight, low volume, low heat leak, and ruggedness in a single vessel. The resulting vessel concept therefore represents a design compromise in that these requirements are generally conflicting. Low weight will result from the use of a high strength alloy for the inner vessel and honeycomb construction for the outer shell. The use of highly efficient, low density insulation and an advanced design for the inner vessel support discs will contribute saving in weight, volume, and heat leak. The development of the improved support discs was one of the most significant products of this effort. The most advanced discs tested involve a 1/2-inch wide composite coil of steel and fabric ribbon.

Two 50-gallon vessels were designed and tested under this contract. The first was a thermodynamic mock-up of the design with no regard for light weight of the inner and outer shells. This vessel was called the boiler-plate vessel. The second was designed to exhibit light weight. Both vessels possessed a length-to-diameter ratio (L/D) of approximately 2.

Table III presents the total observed heat leak for the lightweight vessel tested with liquid nitrogen venting at atmospheric pressure and the estimated distribution of that heat leak. The value of 17.1 Btu/hr was the lowest average heat leak achieved during the test program. For the same vessel tested with liquid hydrogen under venting conditions, a heat leak of 17.7 Btu/hr was observed. Therefore, under these test conditions, test data obtained with liquid nitrogen closely approximate those obtained with liquid hydrogen.

On the basis of developmental testing with the boiler-plate vessel, a heat leak of 11.8 Btu/hr was predicted for the lightweight vessel. The predicted heat leak distribution is presented as the right-hand column in Table III. The discrepancy between the predicted and observed heat leaks is attributed to the insulation and the piping. Failure to achieve a high vacuum in the insulation space was felt to be responsible for the poor performance of the insulation. Apparently the gas leak was in the inner skin of the honeycomb shell which exposed a considerable quantity of epoxy resin to the vacuum space. The high piping heat leak is attributed to conduction heat leak down the silver plating of the pipes. The purpose of the plating

was to reduce the emissivity of the surface and thereby reduce radiative heat transfer. It is considered that both of these defects in the test vessel can be corrected in later designs. For example, the silver plating can be replaced by several layers of aluminized Mylar wrapped around the pipes.

Table IV presents the observed weight and weight distribution of the 50-gallon lightweight test vessel. The predicted values are included for comparison.

All actual component weights were close to the predicted weights with three major exceptions. The inner shell of the test vessel was heavier than predicted due to a last minute materials substitution to allow fabrication and testing of the vessel under this contract. The increased weight of the honeycomb cylinder was probably due to the weight of the adhesive. The support discs used in the test vessel were developed under this contract and were considerably lighter than was anticipated.

TABLE III
HEAT LEAK DISTRIBUTION FOR THE
LIGHTWEIGHT 50-GALLON TEST VESSEL

<u>Component</u>	<u>Heat Leak, Btu/hr*</u>	
	<u>Observed**</u>	<u>Predicted**</u>
Insulation	7.1	4.3
Support Discs	2.3	2.3
Support Rod	1.0	1.0
Piping	3.9	1.4
End Radiation	2.8	2.8
TOTAL HEAT LEAK	17.1	11.8

* Tested with liquid nitrogen venting at atmospheric pressure.

** The total heat leak and the heat leak through the support discs are the only directly measured quantities. The heat leak distribution among the remaining components was estimated from both experimental data and design analysis.

TABLE IV
WEIGHT DISTRIBUTION FOR THE
LIGHTWEIGHT 50-GALLON TEST VESSEL

	As Built	Predicted
Vacuum Jacket		
Cylindrical Section	24.0	20.4
End Closures	5.5	5.5
Heads	10.0	10.0
End Caps	3.0	3.3
Cover Plates	4.1	2.8
Inner Shell	37.0	14.5
Insulation	10.0	10.0
Support Discs	1.3	7.6
Spiders	.7	1.0
Piping	1.5	1.5
Support Tube	3.0	3.0
Heaters	1.0	---
Adsorbent and Container	1.5	1.0
Support Rod	.3	---
Bolts and Nuts	.2	---
TOTAL	103.1	80.6

Based on the experimental data obtained with the two test vessels, it is considered that the 50-gallon vessel conceptual design, as it exists at this time, can be characterized by a heat leak of 11.8 Btu/hr and a weight of 80 lbs. The failures of the test vessels to exhibit these properties were failures in details and not of the conceptual design itself. It is further considered that further development studies will prove the feasibility of heat leaks as low as 10 Btu/hr for essentially the same weight. Further engineering developments are required in the areas of:

- . Outer shell design - to achieve increased reliability of the vacuum seals, decreased weight, and decreased weight;
- . Support discs - to achieve decreased heat leak, increased structural rigidity, and reduced weight;

- Insulation - to achieve lower heat leak through improved application techniques; and
- General design improvements to reduce weight and heat leak through the piping and the end radiation. On the basis of the results of the storage and distribution study, it is recommended that the future vessel studies emphasize liquid delivery to an external vaporizer for increased system reliability.

The prime object of this study was the conceptual design and feasibility testing of two advanced 50-gallon vessels with L/D about 2. However, the weight, volume, heat leak, and boil-off loss have been extrapolated to larger sizes and to other cylindrical shapes to provide preliminary input for future operations analyses of the Energy Depot system. The extrapolations have been based on the tacit assumption that the fundamental vessel design concept for 50-gallon vessels will be applicable for larger vessels as well. This assumption is considered to be sufficiently accurate for the intended purpose.

Figure 4 presents the approximate specific weight for cylindrical liquid hydrogen vessels of various capacities and shapes. Specific weight is defined as the empty vessel weight divided by the net liquid contents weight. The base point for the extrapolation is a vessel with a net capacity of 50-gallons, an L/D ratio of 2, and a weight of 80 lbs. Similarly, Figure 5 presents the approximate specific vessel volume, defined as the outside gross volume of the vessel divided by the gross contents volume. Figure 6 is a plot of the approximate heat leak for cylindrical vessel as a function of vessel capacity and shape. The base point is for a 50-gallon net capacity vessel with an L/D equal to 2 and with a heat leak of 11.8 Btu/hr. Figure 7 presents the approximate effects of the heat leak shown in Figure 6 on the amount of fluid lost during a one-week period of storage for the indicated filling and venting conditions.

Figure 4. APPROXIMATE SPECIFIC WEIGHT OF LIQUID HYDROGEN STORAGE VESSELS

Figure 5. APPROXIMATE SPECIFIC VOLUME OF LIQUID HYDROGEN STORAGE VESSELS

Figure 6. APPROXIMATE HEAT LEAK FOR LIQUID HYDROGEN STORAGE VESSELS

Figure 7. APPROXIMATE HYDROGEN LOSS DURING THE FIRST WEEK AFTER FILLING

III. RECOMMENDATIONS

It is recommended that the development of the Energy Depot cryogenic fuel storage and distribution system be continued and expanded toward the following major goals:

1. Extension of the advanced technology for lightweight compact, rugged, and efficient 50-gallon liquid hydrogen vessels representing the fuel tanks for small vehicles. Areas of development should include:

- . Outer shell design;
- . Support members for inner vessel;
- . Insulation application technique;
- . Piping for liquid delivery; and
- . General design improvements.

The testing program should establish not only heat leak and weight, but also mechanical strength as well.

2. Development of advanced technology applicable for 300-gallon liquid hydrogen vessels representing fuel tankage for large vehicles.
3. Development of advanced technology applicable for 4000-gallon liquid hydrogen vessels representing the tankage for the primary distribution unit.
4. Development of advanced technology of transfer system components required for safe, efficient, and convenient fuel handling under field conditions. The components to be developed include:
 - . A highly flexible (when cold), lightweight cryogenic liquid transfer hose;
 - . A reliable, quick-connect, self-sealing cryogenic fluid coupling;
 - . Highly reliable lightweight pneumatically operated non-leaking cryogenic control valves of specified designs; and
 - . A highly reliable, compact control system for fully automatic fluid handling under field conditions.

5. Refinement of the design of the over-all fuel storage and distribution system to decrease the system weight, volume, and to increase the fuel transfer efficiency and system reliability. Areas for study should include:
 - . The selection of the optimum design pressure based on a fixed venting pressure for the primary and secondary system.
 - . If required in the light of the above item, the development of a variable-pressure vent relief system controlled by the liquid level sensor.
 - . A vent system to allow safe air transport of hydrogen-fueled vehicles with minimum venting loss.
6. Developmental testing of an essentially full-scale bread-board model of the Energy Depot cryogenic fuel storage and distribution system utilizing the advanced system and component design concepts.
7. Design, fabrication, and qualification testing of a prototype Energy Depot cryogenic fuel storage and distribution system.

A program decision may be made by the Government that ammonia is to be the fuel for the first generation Energy Depot. It should be **stated that the** methodical development of the fuel system, outlined above for liquid hydrogen is applicable for ammonia also and will lead to the successful development of the chosen fuel system in the shortest possible time.

IV. LIQUID HYDROGEN STORAGE AND DISTRIBUTION SYSTEM CONCEPTUAL DESIGN

A. INTRODUCTION

The purpose of this study was to ascertain the feasibility of developing a highly automated, efficient, and reliable Energy Depot liquid hydrogen storage and distribution system to be maintained and operated under typical field conditions. It is the result of this study that the development of the above mentioned system is considered to be entirely feasible.

The major portion of the work performed consisted of a conceptual design and performance evaluation of the system. The design employs certain component concepts not presently used in the handling of liquid hydrogen. These will need a period of development and testing to insure the degree of reliability and safety required for the Energy Depot system.

B. SYSTEM DESCRIPTION

1. Major Components

The Energy Depot fuel storage and distribution system is composed of those components utilized for fuel handling from the delivery manifold of the fuel production depot to the delivery manifold on the user's energy conversion device (i.e., fuel cells or internal combustion engine). For the case of the hydrogen-fueled Energy Depot, the fuel storage and distribution system extends from the product manifold of the hydrogen liquefier to the inlet manifold on the energy converter. For the purpose of the present study, the converter has been assumed to be fuel cells which generate electricity to power electric motors for propelling vehicles.

The general criteria for the Energy Depot as a whole and the fuel storage and distribution in particular are presented as Appendices A and B, respectively. These appendices define all of the governing factors which ultimately affect the system's fuel transfer efficiency, weight, operation, maintenance, reliability, redundancy, and safety. These criteria identify two major components of the fuel system--a primary unit and a secondary unit. The functions of the primary unit are: (1) to accept and store the product

from the hydrogen liquefier at the fuel depot; and (2) to provide bulk transportation of the fuel to the points of use. The secondary fuel unit consists of the storage vessels and associated fuel handling equipment on board the use vehicles. The secondary fuel storage capacity is at least an order of magnitude smaller than that for the primary unit.

The 8-ton GOER was identified as the model for the primary storage unit for the purposes of this study. The 8-ton GOER is a four-wheeled, modularized transport vehicle being developed by the Army: (1) to exhibit high off-road maneuverability; and (2) to be readily adapted for various transport missions by the interchange of the rear module. During this study, the rear module was considered to be a flat bed used to support the primary liquid hydrogen storage vessel and its associated equipment.

Figure 1, previously presented and described in Section II, is a conceptual layout of the 8-ton GOER modified for this use. The estimated net capacity of the vessel will be at least 3,200 gallons of liquid hydrogen. This estimate is based on the preliminary vessel layout for cylindrical vessel sections which conform to the existing module space limitations and which allow estimated volumes for the auxiliary equipment. It is anticipated that this capacity can be increased as a result of future design studies. The weight distribution of the load carried on the rear module of the 8-ton GOER is presented in Table I, which is to be found in Section II. It should be pointed out that the weight of the primary fuel system is far short of the 8-ton capacity of the vehicle; this is due in large part to the low density of liquid hydrogen.

The M-113 Armored Personnel Carrier was identified as the representative use vehicle to be studied under this contract. This vehicle was selected because a conceptual design study of a hydrogen-fueled, fuel cell powered, electric motor driven version of the M-113 was recently reported(7). Figure 2, which was presented and described in Section II, is a conceptual layout of the modified M-113 vehicle. The general vessel location and arrangement, established during the earlier study(7), was adhered to for the present study. Because of the preliminary nature of the vehicle conceptual design at this time, certain liberties were taken in the placement of the associated fuel handling equipment on board the vehicle.

Table I, to be found in Section II, presents the estimated weight distribution for the secondary fuel system based on the vessel configuration illustrated in Figure 2. This preliminary configuration was chosen for convenience during the conceptual design of the vehicle. It is the result of the present fuel system study that substantial improvements in fuel economy and fuel system weight and volume are possible by the use of a more-nearly optimum vessel configuration.

From a strictly cryogenic point of view, a single spherical storage vessel would yield the lowest vessel weight, volume, and heat leak because the sphere is a structurally strong configuration and exhibits the lowest surface area-to-volume ratio of any shape, for a given volume. A spherical vessel may not prove the most convenient for a vehicular system where the packaging of many components into a small volume is important. A more convenient vessel shape may be a cylindrical vessel with formed heads and a length-to-diameter ratio (L/D) in the range of one to two. Such a vessel may be more easily packaged into available space than a sphere and yet would be only slightly heavier and bulkier, and would exhibit only a slightly higher heat leak. It is therefore proposed that a single, short, stocky cylindrical vessel be considered as a replacement for the four, separated, long, slender vessels presently envisioned for the M-113. The implications of the split-tank configuration are discussed throughout this section.

2. Process Description

The interrelationships between the primary and secondary fuel units and between the various items of equipment within each unit can be best understood through a consideration of the entire fuel system from a process standpoint. Figure 8 is a flow schematic for the liquid hydrogen storage and distribution system. For the sake of simplicity, the two long vessels along each side of the M-113 (Figure 2) are represented in Figure 8 as a single, long cylindrical vessel.

The following is a process description. A more detailed description of the process equipment is presented as Appendix C. The general operating procedure is covered in Appendix D.

The GOER will be filled by means of a transfer hose connected to the product manifold of the hydrogen liquefaction plant. Electrical interconnection between the plant and the GOER will provide electrical grounding for safety and will facilitate monitoring and control of the transfer process by the depot operator. Cooldown of the primary storage system will be achieved by gradually cooling the equipment with a small quantity of liquid hydrogen from the plant in order to use all of the sensible heat from the liquid hydrogen. During normal usage, the primary vessel will be cold at subsequent filling operations. All liquid hydrogen vaporized during cooldown will be returned to the plant through a flexible hose for reliquefaction. The cooldown will be performed under automatic control and normally will need no manual correction. Once the unit has been cooled to liquid hydrogen temperature during transfer, vaporization will diminish and liquid will begin filling the primary vessel.

VALVE & INSTRUMENT LIST - PRIMARY VESSEL			
C-101	CPLG - V	R-102	RUPTURE DISC (JACKET)
C-102	CPLG - V	R-103	RUPTURE DISC (TRANSFER HOSE)
C-103	CPLG - AIR SUPPLY TO CONTROL VALVES	STC-101	VACUUM SENSING CONNECTION
C-104	CPLG - TRANSFER HOSE TO PRIMARY	SV-101	SAFETY VALVE (INNER VESSEL)
C-105	CPLG - VAPOR RETURN TO PLANT	SV-102	SAFETY VALVE (P.R. COIL)
CV-101	CHECK VALVE	V-101	SHUT-OFF VALVE (STC-101)
EC-101	ELECTRICAL CONNECTION (POWER SUPPLY)	V-102	SHUT-OFF VALVE (VACUUM PUMP)
EC-102	ELECTRICAL CONNECTION (PLANT OR SECONDARY)	V-103	VENT VALVE
LI-101	LEVEL INDICATOR (PRIMARY VESSEL)	V-104	SHUT-OFF VALVE (PI-101)
LI-102	LEVEL INDICATOR (SECONDARY VESSEL)	V-105	SHUT-OFF VALVE (PI-102)
LLS	LIQUID LEVEL SENSORS	V-106	LH ₂ TRANSFER VALVE
PCV-101	PRESSURE REGULATOR	V-107	AIR VALVE (TO OPEN V-106)
PI-101	PRESSURE INDICATOR (INNER VESSEL)	V-108	BYPASS VALVE (V-106)
PI-102	PRESSURE INDICATOR (AIR TO CONTROL VALVES)	V-109	CONTROL VALVE (GAS TO VAPOR SPACE)
PI-103	DIFFERENTIAL PRESSURE INDICATOR	V-110	AIR VALVE (TO OPEN V-109)
R-101	RUPTURE DISC (INNER VESSEL)	V-111	BYPASS VALVE (V-109)
		V-112	BLOCK VALVE
		V-113	BLOCK VALVE (LH ₂ TO P.R. COIL)

VALVE & INSTRUMENT LIST - SECONDARY VESSEL			
V-201	CPLG - TRANSFER HOSE TO SECONDARY	P-201	RUPTURE DISC (INNER VESSEL)
C-202	CPLG - GAS PHASE FROM INNER VESSEL	P-202	RUPTURE DISC (JACKET)
C-203	CPLG - AIR SUPPLY TO CONTROL VALVES	STC-201	VACUUM SENSING CONNECTION
C-204	CPLG - GAS TO FUEL CELL	SV-201	SAFETY VALVE (INNER VESSEL)
CA-201	CHECK VALVE	SV-202	SAFETY VALVE (COOL DOWN GAS BYPASS LINE)
EC-201	ELECTRICAL CONNECTION (POWER SUPPLY)	TS-201	TEMPERATURE SENSOR
EC-202	ELECTRICAL CONNECTION (TO PRIMARY)	V-201	SHUT-OFF VALVE (STC-201)
		V-202	SHUT-OFF VALVE (VACUUM PUMP)
		V-203	VENT VALVE (INNER VESSEL)
		V-204	SHUT-OFF VALVE (PI-201)
		V-205	SHUT-OFF VALVE (PI-202)
		V-206	LH ₂ TRANSFER VALVE
		V-207	AIR VALVE (TO OPEN V-206)
		V-208	VENT VALVE (COOL DOWN GAS BYPASS LINE)
		V-209	CONTROL VALVE (GAS TO FUEL CELL)
		V-210	AIR VALVE (TO OPEN V-209)
		V-211	SHUT-OFF VALVE (PI-203)
LI-201	LEVEL INDICATOR (SECONDARY VESSEL)		
LLS	LIQUID LEVEL SENSORS		
PCV-201	PRESSURE REGULATOR		
PI-201	PRESSURE INDICATOR (INNER VESSEL)		
PI-202	PRESSURE INDICATOR (AIR TO CONTROL VALVES)		
PI-203	INDICATING PRESSURE TRANSMITTER (GAS TO FUEL CELL)		

REVISIONS						
NO	LETTER	ZONE	DESCRIPTION	DATE	MADE	CHK'D
1	A		MISCELLANEOUS PIPING	3-23-64	FEY	QY
			CHANGES			
2	B		SAME AS ABOVE	5-9-64	FEY	QY

V-212	BLOCK VALVE
V-213	AIR VALVE (TO OPEN V-212)
V-214	CONTROL VALVE
V-215	AIR VALVE (TO OPEN V-214)
V-216	CONTROL VALVE
V-217	AIR VALVE (TO OPEN V-216)

LINE LEGEND

—————	LIQUID & GAS
—————	VACUUM JACKETED
-----	ELECTRICAL
-----	AIR
-----	CONNECTION PATH

REQ'D	PART NO	NOMENCLATURE	MAT'L SPEC	UNIT WT	FINISH	ZONE
UNLESS OTHERWISE SPECIFIED DIMS ARE IN INCHES	JR Young	DATE 12/12/63				
FRACTION DECIMAL ANGLES	DRAFTSMAN	12/16/61				
FIRST USED ON	CHECKED	12/17/63				
	ENGINEER	12/17/63				
	APPROVED	12/13/64				
TITLE			AIR PRODUCTS AND CHEMICALS, INC.			
FLOW DIAGRAM OF LIQUID HYDROGEN STORAGE AND DISTRIBUTION SYSTEM			ALLENTOWN, PENNSYLVANIA U. S. A.			
SCALE			67580			
			D			

During the filling of a primary vessel, liquid hydrogen will flow from the plant through a delivery manifold. The manifold will be arranged so that an empty GOER can connect to it while another is being filled. The purpose of this arrangement will be to keep the plant in continuous operation until all primary storage units have been filled, thus decreasing the total time required to complete filling operations. During the length of time required to fill one primary unit, presently estimated to be about 17 hours(7), friction and heat leak into the transfer line will cause liquid to vaporize. This vapor, like the cool-down vapor, will be returned to the plant for reliquefaction.

When one primary vessel is nearly full, a small rate of flow of hydrogen will automatically be directed to a second (empty) GOER connected to the manifold to achieve transfer line and vessel cooldown. When the first primary unit has been filled, the full hydrogen flow will be automatically diverted to the second unit. The filled primary unit will then proceed either to a storage area or to a rendezvous with user vehicles.

If the filled primary unit is to be held in storage for approximately a week or more, the manual by-pass valve (V-112) in the vent system will be opened to allow the check valve (CV-101) to maintain a vessel pressure of about 16 psia. The economy expected from the use of this valve during fuel stockpiling is discussed later in this section. When the primary unit is removed from storage, V-112 will be closed, restoring the pressure control to an automatic relief valve (SV-101) set near the vessel design pressure.

To prepare for filling of a secondary unit, the transfer hose from the primary unit will be connected to coupling C-201 after a small electric cable and a small gas pressure hose have been connected. The cable and gas hose are to allow the monitoring and control of the secondary fuel system from the system controller on the primary unit. The vessels on board the user vehicle will most likely contain liquid hydrogen, even when "empty" and therefore will not require cooldown during secondary transfer. Only the transfer lines will be warm at the beginning of secondary transfer. The initial cooldown of the transfer lines is to be accomplished in approximately one minute by allowing a small flow rate of liquid hydrogen into the line while the hose is attached to the secondary unit. Losses during cooldown will be a function of the time elapsed since the previous transfer. The cooldown vapors will bypass the secondary vessel and will pass directly to the vent line.

When the temperature of the secondary inlet reaches about -350°F , the main transfer valves will be automatically opened, initiating liquid filling of the secondary unit at full flow rate. The liquid transfer between primary and secondary storage tanks will be effected

by a predetermined pressure differential between the two units. During the filling operation this pressure differential will be maintained with a vaporizing coil located on the primary vessel. The flow rate from the primary vessel will be controlled automatically from 50 to 100 gallons per minute (gpm) depending upon the size of the secondary vessel. It is assumed that the secondary vessels will be filled while the vehicle is on relatively level ground. Therefore the left-hand and the right-hand vessels will be filled simultaneously to about the same level.

During operation of the secondary vehicle, the fuel system will automatically supply the proper amount of fuel to the fuel cell power supply. The primary control point will be the pressure in the gas feed line. As the fuel cell power is varied, the varying pressure in the fuel feed line will be used for modulation of the power to the hydrogen vaporizer to maintain a constant feed line pressure.

Inspection of Figure 8 reveals that the piping of the secondary fuel unit can be greatly simplified if the vehicle were designed to accommodate a single storage vessel. Attention is called to Appendix E in which the arrangement of vent and liquid withdrawal piping for multiple, separated, long, slender vehicular hydrogen storage vessels is discussed. The vent piping, shown in Figure 8 in simplified form, will involve two long, criss-crossed pipes above the vessels on each side of the vehicle. This arrangement is to inhibit liquid from entering the vent line as the vehicle negotiates non-level terrain. As discussed in Appendix E, the liquid feed line arrangement is to insure liquid delivery to the vaporizer under similar conditions. If a single vessel with L/D equal to two were employed, it is estimated that the valve and piping weights shown in Table I (Section II) could be cut at least in half and the vessel weight could be reduced by about one-third. This would yield approximately a 500 pound (35%) savings in the weight of the fully loaded secondary fuel system. This estimate is approximate and is included only to provide incentive for re-evaluation of the present conceptual arrangement for the hydrogen-fueled M-113.

C. ANALYSIS OF FUEL SYSTEM STORAGE AND TRANSFER EFFICIENCY

1. Definition of Operation Models

System fuel storage and transfer efficiency is here defined as the ratio of the pounds of hydrogen delivered to the fuel utilization device of the user to the pounds of hydrogen originally charged to the primary storage vessel at the Depot. It is of great importance to the over-all Energy Depot that the fuel system

efficiency be as high as possible to maximize the number of military vehicles that can be supported by any given number of Energy Depots. The present analysis was undertaken to estimate the magnitude of the fuel efficiency for the conceptual fuel storage and distribution system developed under this contract. This estimation will provide guidance for efficiency improvements during follow-on studies.

The fuel system efficiency has been examined for two representative types of Energy Depot operation. These are defined below:

- Fuel Stockpiling - Fuel will be stockpiled in anticipation of a large-scale and complex military maneuver. The following operating model was assumed for the purpose of analysis. After it is filled at the depot, the primary storage unit will be held in a storage area for seven days. After this one-week primary storage period, the contained hydrogen will be transferred into twelve secondary storage vessels. The secondary transfers will occur one each hour for eleven consecutive hours. The primary unit will then be returned to the depot for refilling. After the twelve secondary units have been fueled, they will stand idle for a one-week period, after which each unit will consume 240 gallons of liquid hydrogen in about 12 hours. The total elapsed time for the hydrogen from the depot to reach the fuel cells is approximately 14 days.
- Normal Operation - It is assumed that under normal conditions, the secondary vehicles will utilize fuel at the same rate as it is being produced and that very little fuel storage (or hold-up) will occur. The following operational model has been assumed for the purpose of analysis. After being filled at the depot, the primary unit will be held in storage for 24 hours, after which the fuel will be transferred into twelve secondary units. The twelve secondary transfers will be accomplished at one-hour intervals over an elapsed time of eleven hours. The primary unit will then return to the depot for refilling. Immediately following the secondary transfers, each of the twelve secondary units will proceed to consume 240 gallons of liquid hydrogen in about 12 hours.

The following assumptions and approximations were made to simplify the mathematical analysis:

- The secondary vehicle will store its liquid hydrogen in a single cylindrical vessel which has an L/D of two, one vent nozzle, and one liquid delivery nozzle. It must be emphasized that this is not the configuration of multiple, long, slender, separated vessels represented in Figure 2. The more optimum single-vessel configuration was selected for this analysis as a closer approximation to the configuration which will finally be adopted for vehicles.

- . Hydrogen loss during primary transfer will be zero. Gaseous hydrogen generated as a result of heat leak and friction during primary transfer will be recycled to the liquefier and ultimately re-delivered as liquid to primary storage. Also, some of the liquid which leaves the depot will be vaporized in the primary's pressure raising (PR) coil during secondary transfers. This gas will be returned to the depot in an "empty" primary unit and will be recycled to the plant for liquefaction. The energy required to produce hydrogen by the electrolysis of water is roughly five times that required for hydrogen liquefaction. Since the recycled vapors contribute only several percent to the liquid product stream, the fractional energy required for reliquefaction of the returned vapors is quite small. For this reason the vaporization during primary transfer and in the primary's PR coil are not considered as true losses. Therefore the assumption is made that all hydrogen liquefied is delivered to primary storage.
- . The primary and secondary vessels will be at liquid hydrogen temperature at the beginning of the filling operation. This condition represents the vessels during normal operation. The primary and secondary vessel cool-down losses will therefore be zero.
- . After the specified primary and secondary storage times, the twelve secondary vehicles will consume a total of 2880 gallons or approximately 1700 pounds of hydrogen.
- . Temperature stratification within the hydrogen will not occur. Therefore the vapor pressure of the hydrogen will be in equilibrium with the mixed bulk liquid temperature.
- . The heat leaks into the primary and secondary vessels were estimated from heat leak information presented early in the project (Figure 24 of the First Quarterly Report⁽⁴⁾). That information is an extrapolation to larger vessel sizes of the then expected heat leak of 10 Btu/hr for the 50-gallon test vessel. Similar information, presented as Figure 6 of the present report, is based on 11.7 Btu/hr, a value derivable from the 50-gallon vessel test data obtained during the contract. Since 10 Btu/hr heat leak is still considered the reasonable goal of the vessel development program, the earlier data are considered valid for the present estimates of fuel system efficiency.

Based on the definition of the operating models and the above assumptions, the general expression for the fuel system storage and transfer efficiency may be written as:

$$E (\%) = \frac{1700}{C} \times 100 \quad (1)$$

where E is the efficiency and C is the pounds of hydrogen originally charged into the primary vessel in order to accomplish the missions of the 12 secondary vehicles.

2. Results and Discussion

The derivation of the approximate expressions for the various fuel losses contributing to quantity C is presented as Appendix F, as are the numerical results of the analysis.

Table V contains the results of the analysis. As is shown in the table, five different operating modes were considered for the fuel stockpiling model. These are described in detail in the appendix. These five modes include the two techniques for absorbing the heat leak into a vessel: (1) the vaporization of liquid at a constant temperature by venting at a constant pressure; and (2) the sealing of the vessel and allowing the liquid to warm and the vapor pressure to rise. The latent heat of vaporization of hydrogen decreases as the liquid temperature increases. Therefore liquid vaporization during transfer will increase as the temperature of the liquid being transferred increases. The five modes represent variations in the manipulation of fluid temperature and pressure and are surveyed to determine how best to absorb heat leak during fuel stockpiling to maximize the fuel system efficiency.

Two of the stockpiling modes are seen to yield high efficiencies. As is discussed in the appendix, mode number 2 will exhibit the highest efficiency for primary storage times greater than one week and therefore this mode was selected as the most promising. The pressure profile will be as follows. The primary vessel will be continuously vented at about 16 psia during the primary storage period. During secondary transfer the primary vessel will be pressurized to effect the transfer into the secondary vessel which will be vented at about 16 psia. After secondary transfer, the pressure in the secondary vessel will be allowed to rise, due to heat leak, until the vent opens at 32 psia. Thereafter the vessel will vent continuously at that pressure until fuel is utilized from the vessel at the end of the secondary storage period. It should be pointed out that the selection of mode number 2 as most promising for fuel stockpiling was the reason for the inclusion of the block valve (V-112 in Figure 8) in the primary vent circuit.

TABLE V

ESTIMATED FUEL STORAGE AND TRANSFER
EFFICIENCY FOR VARIOUS OPERATING MODES

Operation Model	Operation Mode	Vessel Pressures (psia)						System Efficiency (%)
		Primary Storage		Transfer Period		Secondary Storage		
		P ₁	P ₂	P ₃	P ₄	P ₁	P ₂	
Fuel	1	16	16	32	16	16	16	74.4
Stockpiling	2	16	16	32	16	16	32	79.0
	3	16	32	32	16	16	16	74.3
	4	16	32	48	32	32	32	76.6
	5	16	32	32	16	16	32	78.9
Normal Operation	6	16	—	32	16	16	—	93.4

P₁ = Pressure of unit after transfer

P₂ = Pressure at which vent relief valve opens

P₃ = Pressure of primary unit during transfer

P₄ = Pressure of secondary unit during transfer

With this valve, the GOER operator can set his vent for 16 psia in anticipation of an extended primary storage period.

The preliminary performance criteria for the fuel storage and distribution system (Appendix B) specifies that:

- . After seven days storage time, 85% of an original primary full load of fuel will be deliverable to secondary storage; and
- . After seven days storage time, 75% of an original secondary full load of fuel will be deliverable to the user's power plant.

This specification indicates an over-all system efficiency of about 64% for consecutive primary and secondary storage times of one week. The estimated efficiency for all five of the stockpiling modes considered is greater than 64%. The most promising mode for consecutive primary and secondary storage times of one week yields an estimated 79%. Therefore, it is considered that the liquid hydrogen storage and distribution system being studied will meet, or exceed, the specified system performance criteria relating to transfer efficiency. It is emphasized that the tankage arrangement considered here is a single, short cylindrical vessel with one liquid delivery line. Preliminary analysis, discussed in Part IV of Appendix F, indicates that the tankage configuration presently conceived for the M-113 Armored Personnel Carrier (Figure 2) will exhibit an efficiency for stockpiling operation of approximately 55% and therefore will not meet the specified criteria.

There is one aspect of vapor venting which should be discussed in connection with system efficiency. The system efficiency and also the system weight are functions of the pressure at which the vent relief valves are set. The interrelationships are complicated, however, and no attempt has been made to select the optimum venting pressures for the primary and the secondary vessels. It has been established⁽⁴⁾ that the density of liquid hydrogen decreases significantly as the saturated vapor pressure is increased. Therefore as the venting pressure is increased, the lower is the initial quantity of liquid hydrogen, at its normal boiling point, which can be charged into a given vessel to insure that liquid does not appear in the vent line. On the other hand, the higher the vent pressure, the longer it will take for the pressure to build up to vent pressure, due to heat leak, and therefore the higher will be the fuel efficiency. As a third effect, the higher the vent pressure, and therefore the vessel design pressure, the heavier the vessel will be, unless of course, the calculated vessel wall thickness is lower than a minimum

value established for fabrication reasons. Therefore it is generally true that the higher the vent pressure is (below the critical pressure) the lower will be the initial net liquid capacity and the storage venting losses, and the higher will be the vessel weight. The optimum venting pressure should be sought during follow-on studies as a guide for both the vessel and system development programs.

The venting pressure of 32 psia used for this estimate of fuel system efficiency is based on the assumptions of a vessel initially filled to 90% of gross capacity with liquid hydrogen at its normal boiling point and a maximum liquid volume fraction of 95% to accommodate the vent system. These assumptions were selected as a preliminary attempt to obtain high initial net liquid volume fraction, high storage efficiency, low vessel specific weight and high over-all safety due to storage of the hydrogen at a relatively low pressure.

Future studies should consider the feasibility and desirability of developing a variable-pressure automatic relief valve. The valve spring tension, and therefore the relief pressure, could be increased automatically as the liquid level dropped. By means of this arrangement, a vehicle operator who had already used a part of his fuel and was required to park for a matter of days would benefit from the decreased venting loss due to the increased venting pressure.

D. ANALYSIS OF SYSTEM SAFETY

1. Introduction

There is every reason to expect that the Energy Depot liquid hydrogen storage and distribution system can achieve a degree of safety commensurate with that obtained in commercial liquid hydrogen practice. This will be so because: (1) the properties of liquid hydrogen are known and understood; and (2) the equipment for handling hydrogen is designed specifically for this service on the basis of the above knowledge. It is the purpose of this portion of the report to discuss the potential hazards of the liquid hydrogen storage and distribution system as one step in the process of assuring the actual safety of the system. The following paragraphs contain a summary of the pertinent properties of hydrogen and a discussion of the safety aspects of the conceptual system designed under this contract.

2. Properties of Hydrogen

A great deal is known about the physical and chemical properties of gaseous and liquid hydrogen, especially as these affect the safety of handling this fluid. The pertinent properties are listed and discussed in detail in Appendix G. In summarizing the potential hazards associated with handling hydrogen, it is noted that the general problems are the same as those encountered during handling

of the more common hydrocarbon fuels and cryogenic liquids. In some cases, the degree of the problem is greater; in some cases, less.

The minimum ignition energy is lower and the combustion concentration range is wider for hydrogen than for hydrocarbon fuels and, therefore, ignition is potentially more probable. The low flame quenching distance makes hydrogen fires more difficult to put out, compared to other fuels. Hydrogen flames, particularly small ones, are nearly invisible. Large flames show color, possibly from impurities in the air, or from equipment being heated, or possibly from the flame itself. Infra-red photographs show that the high temperature zone extends beyond the visible flame envelope.

On the other hand, the high diffusivity of gaseous hydrogen hinders the maintenance of flammable hydrogen-air mixtures, thereby reducing the probability of ignition in relatively open spaces. If ignition occurs, the high diffusivity reduces the probability of a transition from mere conflagration (no over-pressure wave) to detonation (strong over-pressure wave) to essentially zero. The low emissivity of a hydrogen flame greatly reduces the radiation damage from a hydrogen fire (i.e., the probability and extent of the damage to nearby objects). The large temperature difference between liquid hydrogen and the ambient surroundings greatly reduces the duration of a fire over a pool of liquid hydrogen. The necessity for storing and transferring liquid hydrogen in closed, well insulated equipment protected by relief devices almost eliminates the exposure of people to burns and puts the pressure rupture problem on the same level with other pressure vessel applications.

For fuel handling in relatively open spaces, it can generally be concluded that hydrogen fires will be easier to ignite, harder to put out and will do less damage than fires with other common fuels. From a practical point of view the following general guidelines can be established. To minimize the probability of ignition, the equipment should be designed: (1) to prevent the fuel/oxidant concentration ratio from reaching the flammability limits; and/or (2) to eliminate any sources of ignition. The control of both (1) and (2) simultaneously is preferable.

The most practical approach to fire control is either: (1) to shut off the supply of hydrogen, if this is possible; or (2) to clear the area and let the fire burn itself out, which it will do quickly (in several minutes) if, for instance, a large quantity of hydrogen is dumped on the ground and ignited. After the hydrogen fire is out, personnel may be able to re-enter the area and put out by usual techniques any small fires which might have been started. These practical guidelines form the basis for the following safety critique of the conceptual fuel storage and distribution system.

3. Safety Considerations in the Design and Operation of the Conceptual Liquid Hydrogen Storage and Distribution System

a. Primary Vessel

Pressure relief devices for the vessel will be provided, SV-101 for normal venting due to heat leak and R-101 sized to accommodate vaporization due to loss of vacuum in the event of rupture of the outer shell. The vent stack will be located at a height of eleven feet above grade and a check valve (CV-101) will be provided to prevent air entry into the stack. The vent stack will be located away from the electrical controls, which will be at the rear of the unit. The vapor flash during filling is to be returned to the plant to eliminate the possibility of exposing personnel and equipment to a flammable atmosphere, and at the same time to conserve hydrogen gas. The volume of gas vented during normal operation or while the vessel is in storage is small and the gas will disperse readily into the atmosphere.

The thermister-type liquid level sensing device is to be operated on a milliwatt electrical system with electrical leads sealed at the entrance of the vacuum system. This type of system has been used on stationary storage vessels and has not created any safety problems. "Intrinsically safe" capacitance probe-type liquid measuring devices are now being developed and may have application to this system. These units are to operate with a maximum energy output of .01 millijoules which is one-half of the minimum energy required to ignite an air-hydrogen flammable mixture. Thus any electrical failure of the system would not be hazardous even if flammable mixture were present.

The system controller and control console located at the rear of the unit are to be provided with equipment suitable for Class I, Group B, Division 2 areas⁽⁸⁾. Consideration has been given to designing all of the electrical system Class I, Group B, Division 1 due to the possibility of leaks created by road accidents or other system failures. However, the vessel or piping failure could also be the source of ignition, thereby lessening the advantage of Division 1 electrical equipment. The system controller is to be located the maximum distance feasible from the transfer hose and the control console. This is considered advisable to minimize electrical sources of ignition in close proximity to potential hydrogen leakage on the transfer hose or at instrument tubing connections. Formation of a flammable mixture caused by leakage around the console can be prevented by eliminating an enclosure on the back of the control console panel or providing

ventilating holes at the top of the enclosure with an air inlet at the front.

The control valves will be fail-safe, i.e., they will be designed to close on loss of pneumatic pressure or electrical power. In addition an alarm will be provided to indicate a failure of the control valve actuating systems.

A bypass is to be incorporated into valve V-106 by drilling a hole through the upstream and downstream chambers on the valve body. This hole will serve to relieve line pressure in the event vacuum is lost on the line between V-106, V-108, and the coupling check valve assembly C-104.

Electrical grounding to the filling plant or secondary vessel is to be accomplished by a grounding clamp secured to the transfer hose. Storage areas should be provided with a system to permit grounding of the primary units during storage.

When the fabrication of the system is completed at its manufacturing site, the storage tank and associated piping will be purged with inert gas. Pure hydrogen gas will then be purged into the system thereby expelling the inert gas. The system will then be pressurized with hydrogen gas and sealed off for shipment. This procedure will remove all air from the system thereby preventing the formation of a flammable mixture and will prevent the freeze-out of the inert gas which would remain in the system as a contaminant. For any maintenance activity which would allow air to enter the piping and vessels, the system must be returned to a service facility where purge gas is available.

b. Secondary Vessel

Pressure relief devices provided on the secondary vessel are to be similar to devices on the primary vessel. In addition rupture disc R-201 will be sized to adequately relieve flow if the pressure build-up heater is operating at maximum capacity, the tank vacuum system is lost and the discharge shut-off valve V-209 is closed. Limitations on the height of equipment on the secondary vessel (seven feet) necessitate a stack extension or a telescoping design to elevate the stack to about 15 feet during secondary fill. This vent height specification has been established empirically and is in common usage. The extent to which this specification can be lowered safely is not known at this time. The vent stack should be located such as to obtain the maximum distance feasible from the electrical controls and personnel compartment.

Comments in the preceding sections on the liquid level sensing device and control systems on the primary vessel are also applicable to the secondary vessel design.

Prior to initial filling of the secondary vessel with liquid hydrogen, the vessel must be purged with an inert gas. As noted before, the inert gas will be purged with hydrogen gas before filling with liquid hydrogen. As nitrogen purging will not be generally feasible in the field, it will be necessary to store the secondary vessels under hydrogen pressure. Any maintenance on the hydrogen system will necessitate returning the vessel to a service area where purge nitrogen is available.

In the design of secondary vehicles for transporting personnel, consideration should be given to the partitioning of the vessel compartments from the personnel compartments to minimize exposure of personnel to hydrogen leaks. The storage tank compartments should be ventilated by means of an air scoop at the front of the compartment with adequate openings at the rear.

Air transport of a hydrogen fueled vehicle is to be considered a definite possibility. One approach would be to design into the aircraft a vent system which would attach to the vent pipes of the contained vehicles, and which would direct any vent gas outside of the airplane. Some sort of variable pressure relief valve must be designed into the plane's system to compensate for the reduced atmospheric pressure at high altitude and to prevent premature venting from the vehicles.

An electrical vaporization heater has been specified because of its rapid response, fine control characteristics, and the availability of electrical power. It should be recognized, however, that a failure (electrical short or overheating and burn out) of the heater can supply sufficient energy for ignition. It is true that under normal conditions there will be no oxygen in the hydrogen system and thus no flammable mixture to ignite. However, operator error, such as inadequate initial purging, can provide sufficient air to form a flammable mixture.

c. Liquid Transfer

The transfer hose under consideration will incorporate check valves that close on disconnecting. The possibility that the hose from the primary vessel may be disconnected when liquid hydrogen is in the line requires that the pressure relieving device, R-103, be provided to prevent overpressurization of the line by vaporization of trapped liquid.

As the transfer hose will be under pressure, the design of the transfer hose coupling check valve must be as leak-free as possible.

Storage of the hose will be in a ventilated compartment remote from any foreseeable source of ignition. Design of the hose coupling check valve should minimize, if not eliminate, entrapped air between the check valves during hose connection. Design of the coupling should include provision for purging entrapped air during or following "making up" the coupling. Entrapped air could result in particles of ice and solid air collecting in the storage tanks. Accumulations of solid air in hydrogen, however, are difficult to ignite⁽⁹⁾ and may present more of a mechanical problem (flow restrictions or prevention of the check valve from closing) than a flammability problem. Minimizing the volume of air entrapped by the check valves will also serve to also limit the quantity of hydrogen vented during the coupling and uncoupling of the hose.

Failure of the transfer hose or any part of the transfer system may expose the transfer operator to the hazards of a readily ignited fuel as well as cryogenic burns. It is highly recommended that, as a minimum, all operators be provided with and should wear gauntlet gloves which are non-porous and suitable for handling metals at cryogenic temperatures. White coveralls, gauntlet gloves and face shields are worn by personnel for many commercial liquid hydrogen operations. The relative hazards due to fuel handling and the military environment will have to be evaluated further in future studies to determine if the logistics problem and inconvenience of full protective clothing are warranted.

d. Hydrogen Leak Detection and Fire Extinguishment

In the operation of hydrogen systems experience has shown that fixed leak sensing devices are not practical since a leak inches away from a sniffer may not be detected because of the rapid rise and dispersion of the gas. To permit periodic checking for leaks a portable hydrogen detector should be provided on each secondary and primary vehicle. Location of liquid leaks are readily detected with the naked eye by observing the air condensate.

Consideration was given to providing an inert gas purge to the vent stack to extinguish a vent fire. The small quantity of purge gas, as might be considered due to vehicle weight and volume restrictions, would not be effective. A sustained inert gas purge or vent cooling with water would be required, as venting hydrogen may be reignited by the high temperature

of the stack exit following a vent fire. Without vent cooling or sustained purging, it is improbable that a vent fire could be extinguished under field conditions. Therefore, the most practical and effective fire fighting technique will consist of closing valves to stop the flow of hydrogen, when possible, and to allow the venting hydrogen to quickly burn away. In the event of a liquid spill that cannot be controlled by a shut-off valve, the only action to be taken would be to have personnel vacate the area as promptly as possible. Explosions of hydrogen-air mixtures would not be expected if the mixtures are not confined. Recent experimental work⁽¹⁰⁾ demonstrates that near stoichiometric hydrogen/air mixtures ignited by a detonator can detonate, although the probability of achieving the necessary confinement appears remote under field conditions.

On the basis of the information presented in Appendix G, there does not appear to be any unusual explosion hazard due to the puncture of liquid hydrogen vessels by bullets or by other rupture of the vessel.

e. Consideration of Additional Support Equipment

The field maintenance of the secondary hydrogen circuit without a nitrogen purge is not recommended. Consideration was given to supplying the primary vehicle with a small liquid nitrogen storage and vaporization system for the purging of secondary vessels. The liquid nitrogen would be supplied by the nitrogen refrigerator which is part of the hydrogen liquefaction plant. Later studies should evaluate whether the required added complexities within the Depot and the primary unit will be compensated for by the elimination of delays in effecting minor repairs on user vehicles, i.e., pneumatic control valve replacement.

V. INTERIM LIQUID HYDROGEN
STORAGE AND DISTRIBUTION SYSTEM

The preliminary design was prepared for a liquid hydrogen storage and distribution system that could be presently fabricated for immediate testing and experimentation by the Army. This system was designed to perform the various functions required by utilizing presently available commercial materials and equipment. Consequently, the interim system will not exhibit certain advanced characteristics, such as extreme simplicity of operation, lightweight, and high reliability, of the conceptual system described in the previous section. The interim system will allow the Army to experiment with hydrogen-fueled vehicles during the period of development of the prototype Energy Depot.

The interim system, like the conceptual system, is for distribution of the fuel from its point of manufacture to its point of use (e.g. engine or fuel cells). The two systems are similar in all major respects (e.g. both involve a primary bulk storage and distribution unit and a secondary storage unit. However, the interim system design is based on the use of equipment commercially available at this time or which can be fabricated now on the basis of existing technology.

The important differences between the interim and the conceptual (advanced) systems are summarized below:

- . The interim system will be for experimental and training purposes, and is not designed to be a fully developed, operational system.
- . The interim system can be built now, while the conceptual system requires development.
- . The transfer hose couplings for the interim system will not be self-sealing. Therefore the transfer lines and hoses must be purged before and after each transfer. Purging will not be required for the conceptual system during normal operation.
- . The interim system will involve some automatic control of the transfer, and the reliability of the system and its components will be lower than for the conceptual system. The interim system will be more complex to operate.
- . The weight and volume of the interim system may be higher, compared to the conceptual system.

The primary unit will be a liquid hydrogen semitrailer which can be used for bulk storage. The limited mobility of the semitrailer will restrict bulk transportation of liquid hydrogen to improved roads. The secondary unit will be the fuel storage and handling system on board the user vehicle. It is assumed here that no unusual space limitations will exist when mounting the experimental tankage system on a vehicle.

With very few exceptions the equipment types and functions for the interim system are the same as those already discussed for the conceptual system. The description of the process equipment for the interim system is included in Appendix H which also presents the preliminary interim process flow sheet. In Appendix J, the general operating procedures for the interim system are outlined. In particular, purging procedures required for the interim system are described.

A detailed system efficiency analysis, similar to that reported for the conceptual system, has not been performed for the interim system. The purpose of the interim system would be to allow the Army to conduct testing programs with hydrogen-fueled vehicles. It is here assumed that these studies will generally be carried out within the Continental United States and that system efficiency will not be a major criteria, as would be the case for the fully developed, operational Energy Depot System.

In most respects, the interim and the conceptual fuel systems will be identical from a safety standpoint. The one important exception is a result of differences in the design of the transfer couplings. For the advanced system the couplings at the ends of the transfer hose and at the vehicles manifolds will be of the quick-connect, double-block-and-bleed type. No purging of the transfer lines or hose will be required during routine operation because no air can leak into the system. Basically, the volume to be purged will have been reduced from the total volume of the transfer system to the small volume between two couplings during connection and disconnection. The "bleed" feature of the couplings will be adequate for this purpose. For the interim system, the manifolds and hosing will not involve double-block couplings. Therefore, before each transfer operation, the air in the system must be purged with nitrogen and the nitrogen then purged with hydrogen. After each transfer, the hydrogen remaining in the lines must be purged with nitrogen. These purges will eliminate the possibility of achieving a flammable hydrogen/air mixture in the system.

It should be reiterated that there are two principal methods of preventing fires: (1) to avoid flammable fuel/oxidant mixtures, and (2) to eliminate all sources of ignition. If either one of the above conditions can be eliminated with certainty, then the chances of having a fire are eliminated with certainty. The interim system has been designed to eliminate

the possibility of achieving a flammable fuel/air mixture for normal operations. It is recognized that hydrogen transfers have been made in which the air in the lines has been purged with hydrogen. On only rare occasions has there been a fire during these transfers. The exact causes of these ignitions are not known. While hydrogen transfer through non-inerted piping is considered relatively safe under certain conditions (e.g. piping electrically grounded) it is not possible to say that all sources of ignition can be controlled and eliminated. The probability of ignition will not be absolutely zero. Further study will be required to evaluate the relative hazards and advantages of transfer with a non-inert system for this particular application.

VI. VESSEL CONCEPTUAL DESIGN

A. INTRODUCTION

The 50-gallon vessel size was given primary attention during the vessel study for two reasons: (1) it represents the lower limit of the size range of interest in the Energy Depot Program; and (2) low vessel weight and heat leak are more difficult to achieve in the small vessel sizes. The approach was to solve the most difficult problem (small vessels) first in order to maximize the success of the over-all program. While emphasis was given to the 50-gallon vessel size, the conceptual design is not limited to that size. Many of the features included in the 50-gallon design are applicable to larger vessels as well.

Commercial superinsulated liquid hydrogen dewars, available in the size range of 150 to 175 liters (40 to 46 gallons), have very good heat leak rates (about 1% of capacity per day or 2.2 Btu/hr) but are heavy (specific weight of 6.5 to 8 lb/lb H₂). These designs are not considered to represent the desired compromise between heat leak rate, weight, volume, and ruggedness required for the Energy Depot System. It was also determined that merely changing to higher strength materials would not yield sufficient improvement. For these reasons, a vessel design study was initiated with the goal of attaining a design which combines compactness, simplicity, ruggedness, low weight and low boiloff into one vessel. It was also deemed necessary that the basic design should be adaptable to either a vertical or horizontal orientation.

B. DESIGN CRITERIA

The preliminary criteria established by the Atomic Energy Commission Energy Depot System are listed as Appendix A, and those for the Fuel Storage and Distribution Systems are presented as Appendix B. The criteria of interest in vessel design are:

- . The vessel is to be light in weight relative to the weight of its contents.
- . The vessel should be compact relative to its usable volume.
- . The heat leak rate should result in a fluid loss of less than 25% of the original tank contents during a 7-day period.
- . The vessel should be sufficiently rugged to withstand normal handling and transportation shock loads.

It should also be noted that compliance with the ASME Pressure Vessel Code or any of the other usual criteria observed for commercial vessels is not presently required.

Based on the above rather general structural criteria, the following specific design criteria were chosen:

- . The vacuum jacket should be able to sustain a two-atmosphere external pressure load.
- . The maximum working stress (hoop stress at maximum allowable working pressure) in the inner vessel should be one-half of the room temperature yield stress of the vessel material at its weakest point (This represents an attempt to reduce vessel weight by reducing the design safety factor below that presently specified by the ASME codes.).
- . The support system should be capable of sustaining a 10-g loading in any direction.

No attempt was made by either the Atomic Energy Commission or Air Products to specify an acceptable level of weight or volume because this program was undertaken more to determine what the vessel weight and volume could be, rather than attempt to develop a design to specific criteria which would of necessity be chosen arbitrarily.

The heat leak criteria above is sufficiently specific as originally stated. This requirement is somewhat stringent for a hydrogen vessel and dictated the use of the best available insulation; viz., a combination of high vacuum and multiple radiation shields. In addition, careful attention to the heat leak through the supports and piping was necessary.

C. CONCEPTUAL DESIGN

The vessel conceptual design as it presently exists has already been presented in Section IIB as Figure 3. This vessel will be basically a cylinder within a cylinder, with a trunnion-type of support system. The inner, or pressure shell will be of aluminum alloy 2219-T87. This selection is based on the results of the inner shell study which are reported as Appendix K. The insulation will consist of multiple-radiation-shields and vacuum. The cylindrical section of the outer shell, to contain the vacuum in the insulation space, will be of aluminum-honeycomb construction. The vapor vent and liquid fill lines will enter the vessel through the central support pipe. A suitable aluminum-stainless steel welding transition piece will be required in the piping design (See appendix L).

Longitudinal support and positioning of the inner vessel will be by means of an Invar rod in tension. The rod will extend the full length of the vessel through the central support pipe. Radial support is to be provided by a support disc in compression at either end of the vessel.

This radial support concept is somewhat unconventional in cryogenic vessels because it places the support members in compression. In compression, stability frequently dictates the size of the member, rather than compressive strength, hence the cross-sectional area is usually much larger than would be the case for a tension support. Because of this it will be necessary to use materials with a very low thermal conductivity. For example, in the case of the 50-gallon test vessels, organic insulators such as nylon or micarta were not suitable because their thermal conductivity is too high. However, a disc of composite construction, utilizing a contact resistance between each of its many layers, has been found to be acceptable. This type of support disc is discussed in more detail in Appendix M.

The primary advantage of this method of support is its compactness, which means that the outer vessel may be smaller and therefore lighter. In addition, it eliminates any support loading on the outer shell by transmitting the inner vessel load out to the ends of the outer vessel at which points the external supports are located. A third desirable feature of this support system is that it will not necessitate additional breaks through the insulation thereby contributing to ease of vessel assembly and to low heat leak. It is also possible to pass the piping through this support, eliminating the necessity of further breaks in the insulation, and also of additional points of stress concentration in the inner shell.

This entire vessel design concept is discussed in detail, including various design calculations, in Reference 4. That discussion will not be repeated here. Appendix N consists of the general specifications of a 50-gallon vessel of this design.

D. ESTIMATED CHARACTERISTICS OF THE DESIGN

1. General

In the First Quarterly Report⁽⁴⁾ parametric curves were presented to indicate the anticipated relationships of vessel weight, volume, heat leak and venting loss to the vessel size and shape for cylindrical vessels. This present section is a representation of this information which has been revised and refined on the basis of the studies performed under this contract. The major design effort under the contract has been related to the 50-gallon vessel size. The weight, volume, heat leak and venting loss characteristics presented here for larger vessels are intended solely to provide basic input data for future parametric analyses of the Energy Depot System.

2. Weight and Volume

The approximate specific weight of a vessel designed as described in the previous sections is shown in Figure 4 as a function of vessel shape and capacity. Figure 4 is to be found in Section IIB. Specific weight is defined as empty vessel weight divided by the contents weight for the vessel filled to 90% of gross capacity. This graph is an approximation only, since the design was left unchanged in concept as size was increased beyond 50 gallons. In actual practice, the basic design would be re-evaluated for each specific case of interest and the weight for the larger vessels may be less than shown. It should be noted that these data replace data previously presented in Figure 22 of the First Quarterly Report⁽⁴⁾, the difference being that the results have been modified to include the experience gained in the intervening months. The calculation procedures were reported in Appendix C of the First Quarterly Report.

The most significant changes in the weight calculations are:

- (a) A yield stress of 35,000 nsi through the weld of the inner vessel is used for an unmilled sheet. Previously, a chemically milled sheet was assumed and a parent metal yield stress of 58,000 psi was used. A discussion of the application of chemically-milled inner shells is included in Reference 4, Section IVB.
- (b) Now the minimum thickness of inner vessel is 0.060 inches instead of 0.020 inches.
- (c) Allowance for miscellaneous equipment was changed from 10 to 15%.
- (d) An adjustment was made to the outer shell head weights.

An explanation of (a) is that for large vessels, it may not be practical to use chemical milling, hence to keep the weight calculations more general, unreinforced sheets were used. For unreinforced sheets (not chemically milled) the weakest point of the vessel is the weld joint; hence, material strengths of the joint must be used. For milled sheets, the joint is sufficiently reinforced so that the parent metal becomes the weakest point; hence, parent metal properties may be used. For (b), experience has shown that welding high strength aluminum alloys becomes increasingly difficult as the thickness decreases, so that for the production of vessels in quantity a minimum thickness of 0.060 inches is thought to be more reasonable at this time. Note that this restriction accounts for the odd behavior of the weight curves at the lower capacity. For (c), the 15% figure is consistent with the results of the "as-built" lightweight 50-gallon vessel.

Unfortunately, all of the above changes are in the direction to add to the vessel weight. However, it is considered that the design results in a vessel that must be considered "light" by present commercial standards other than those used for aircraft and space vehicle applications.

The results shown in Figure 4 are smooth curves drawn through calculated points. Vessel specific weights were calculated for each L/D at gross volumes of 50, 150, 500, 1500 and 5000 gallons. The rather odd behavior of the L/D = 8 curve is caused by arbitrary design restrictions such as minimum inner shell thickness, minimum honeycomb sandwich facing thickness, and maximum sandwich thickness. The curve for L/D = 8 is affected most by these thickness restrictions because, for a given vessel volume, the higher the L/D ratio is, the smaller will be the vessel diameter and the required wall thickness.

The estimated specific vessel volume is plotted as a function of vessel capacity and shape as Figure 5, which is located in Section IIB. Specific volume is defined as the total outside volume of the vessel divided by the gross volume of the inner vessel. It should be noted that the net liquid capacity of the vessel will be 90% of the gross capacity.

The apparent minimum in specific volume resulted from the assumption that the facing thickness for the vacuum shell did not vary with vessel diameter, an assumption made to achieve low shell weight for these preliminary calculations. The minimum can be eliminated and the specific weight can be reduced by increasing the facing thickness with diameter, which will, however, increase the specific vessel weight.

3. Heat Leak

Based on the experience gained in this program, Figures 24 and 25 of Reference 4 have been revised and are presented as Figures 6 and 7 (in Section IIB). The results shown in the new figures are extrapolations to different shapes and sizes of vessels from the value of 11.8 Btu/hr shown in Table VII in Section VIID of this report. Again, as in Reference 4, the equation used for extrapolation is of the form

$$Q = C_1 A_s + C_2 V^{1/2}$$

where C_1 and C_2 are constants and A_s and V are the inner vessel surface area and volume, respectively. Figure 6 presents the estimated heat flow rate into the vessel divided by the liquid hydrogen weight plotted as a function of vessel size and shape. Figure 7 shows the effect of the rate of heat addition presented in Figure 6 on the amount of fluid lost during a seven day period of non-use. The conditions of 90% initial fuel volume fraction, one atmosphere initial vessel pressure and venting at a constant pressure of 32 psia have been assumed.

VII. 50-GALLON LIGHTWEIGHT VESSEL EXPERIMENTAL STUDY

A. INTRODUCTION

This part of the program was performed to demonstrate the feasibility of the vessel design concepts developed during the earlier parts of the program. This study was necessary in order to determine what practical limitations and shortcomings exist with this design concept.

The lightweight vessel incorporates thermal design improvements to the original design which were developed during the boiler-plate study. (See Appendix P for the final experimental results of that study). The lightweight vessel is therefore very similar to the boiler-plate vessel with respect to heat leak, but it also incorporates the advanced structural design developed earlier in the program.

This section of the report is a description of the lightweight vessel as actually built, the tests conducted, the results, and the conclusions based on the test results.

B. TEST DESCRIPTION

1. Test Vessel

The 50-gallon lightweight test vessel, for which Figures 9, 10, and 11 are the assembly and detail drawings, consists of a 20-inch diameter, 45-inch long inner tank enclosed in a 24-inch diameter, 49-inch long honeycomb vacuum jacket. The cylindrical shell of the inner tank was fabricated of 0.125-inch thick type 5083-aluminum shell and 0.060-inch minimum thickness 7039-aluminum ASME 2/1 elliptical heads. These heads were spun from 0.090-inch thick material. The shell of the vacuum jacket was fabricated from a honeycomb sandwich consisting of two 0.020-inch thick aluminum alloy faces bonded to a 1/2-inch thick core. This shell was welded to ASME torispherical heads spun from 0.090-inch thick 6061-aluminum.

It was previously reported⁽⁶⁾ that the cylindrical shell of the inner tank would be fabricated from 0.063-inch thick, 7039-aluminum alloy chemically milled to a thickness of 0.045-inch. Since that time, it has been impossible to obtain consistently high quality, high strength welds with this material in this thickness. Thus, it was necessary to utilize 5083-aluminum, which was available, in order to fabricate a vessel which could be tested for heat leak characteristics before expiration of this contract. The welding problems with 7039-aluminum alloy are discussed in detail in Section D of Appendix K.

NOTE:
END ASSEMBLIES ARE SIMILAR EXCEPT FOR 7/8 TUBES & RELATED ITEMS.

FIGURE 9

Air Products and Chemicals
ALLENTOWN, PENNSYLVANIA, U. S. A.

50 - GALLON
LIGHTWEIGHT TEST VESSEL ASSEMBLY

DRAFTSMAN D. T. BUSS
ENGINEER P. STELTS
DATE 9/27/63
SCALE NOTED

DWG. No.

67581 - D
REV. A

REVISIONS	DATE	MADE	APPR'D	DESCRIPTION
A	3-15-64	D.T.B.	P.S.	REVISED & REDRAWN
LETTER	DATE	MADE	APPR'D	DESCRIPTION

NOTE:
SEE DWGS. 67582-D & 67583-D FOR DETAILS.

FIGURE 10

NOTE:
1. SEE DWG. 67581-D FOR
NEXT ASSEMBLY.

					DRAFTSMAN D. T. BUSS		DWG. No. 67582 - D REV. A
					ENGINEER P. STELTS		
A	REVISED & REDRAWN	3-11-64	B.J.G.	PS	DATE 1-6-64		
LETTER	DESCRIPTION	DATE	MADE	APPRD	SCALE 1/2 & FULL		
REVISIONS

FIGURE 11

Air Products and Chemicals
INC
ALLENTOWN, PENNSYLVANIA, U.S.A.

DETAILS - 50 - GALLON
LIGHTWEIGHT TEST VESSEL

NOTE:
1. SEE DWG 67581-D FOR NEXT ASSEMBLY

DRAFTSMAN D. T. BUSS				DWG. No.	
ENGINEER P. STELTS					
DATE 11-12-63					
SCALE FULL					
A	REVISED & REDRAWN	3-13-64	OB	PS	
LETTER	DESCRIPTION	DATE	MADE	APPR'D	
REVISIONS					

67583 - D
REV. A

Flanged end caps 6 5/8-inches in diameter and 3-inches long were provided at each end of the vacuum jacket to house the support discs and radiation shields. The vent and fill lines from the inner vessel leave the jacket through one end cap, as shown in Figure 9. The vacuum pump connection to the vessel is also made at this end. The annular space between the inner and outer tanks is evacuated through this port. A connection for monitoring the vacuum in the annular space is incorporated in the opposite end cap.

The inner vessel is supported radially by Nylon-stainless steel ribbon composite support discs located in the end caps. An Invar rod fastened between 4-point support spiders in the end caps and also inside the central support pipe provides longitudinal stability to the inner tank. Two mirrors in each end cap, one attached to the spider and one attached to the support disc, act as radiation shields to reduce heat leak down the 3-inch support tube. A bed of activated charcoal adsorbent is located within one end of the 3-inch diameter support tube. Flash butt-welded aluminum-stainless steel welding transition pieces (see Appendix L) were used to join the steel vent and fill tubing to the aluminum mandrel assembly.

2. Test Stand

The test apparatus for the heat leak study of the 50-gallon light-weight vessel was essentially identical to that employed for the boiler-plate vessel tests. The major items of equipment, instrumentation, and piping are shown in Figure 12. The main flow stream is indicated by the heavy line.

Cold vapor generated by heat leak into the inner tank was vented through an electric heater. The heater was designed to warm the cold vapor to room temperature. In actual practice, it was found that operation of the heater was not required. The vapor then flowed through an adjustable back-pressure regulator which could be set to maintain a constant pressure of up to 100 psig in the vapor space of the vessel. Vapor from the regulator next passed through a long coil to stabilize the vapor temperature to facilitate flow measurement. The coils were originally designed to be immersed in water. It was found in practice that ambient air warming of the coils was adequate. The vapor flow rate from the vessel was measured by means of a calibrated wet-test meter. Vapor from the wet-test meter was vented to the atmosphere through a 2-inch exhaust stack which terminated about 10 feet above the roof of the laboratory. During testing, the vessel was suspended from a tension load cell equipped with remote digital readout. This allowed an additional determination of the rate of boiloff from the vessel by a technique which was independent of the flow rate method.

FIGURE 12

 ALLENTOWN, PENNSYLVANIA, U.S.A.	
FLOW DIAGRAM FOR 50-GALLON VESSEL STUDY	
DRAFTSMAN	D. J. Buss
CHECKER	P. J. Buss
ENGINEER	C. M. B. Clouder
DATE	9/1/63
SCALE	NONE
D.W.C. No. 67586-D	

A temperature recorder was included in the test apparatus to continuously record the outer shell temperature as indicated by eight thermocouples on the vessel. A manometer and pressure gauges were provided to monitor the vapor pressure in the vessel. Additional hardware shown in the drawing consisted of safety equipment and miscellaneous instrumentation.

3. Test Procedure

The test procedure for the 50-gallon, lightweight vessel study was planned and conducted as follows:

- (a) A heat leak test was made with liquid nitrogen and liquid hydrogen under venting conditions. The heat leak during the test period was evaluated from both the instantaneous and integrated flow rates of the vented vapor, and from the total weight loss of the vessel. The shell temperature was monitored continuously to observe its effect on the instantaneous heat leak.
- (b) A heat leak test under no-flow conditions was made with liquid nitrogen and liquid hydrogen. The vapor pressure in the vessel and the shell temperature were continuously monitored throughout the test. The total heat input to the vessel was calculated as the difference in the internal energy of the fluid contents and the cold parts of the vessel at the initial and final pressures.

The calculation procedures are discussed in Appendix P in relation to the 50-gallon boiler-plate vessel study.

C. TEST PROGRAM AND RESULTS

This section is a summary of the results of the heat leak testing of the lightweight vessel. Detailed results are presented in Appendix Q.

Initial testing was with liquid nitrogen and the results indicated an insulation space pressure in excess of 10^{-5} mm Hg and a heat leak rate of 20 Btu/hr. The high heat leak rate was thought to be caused by the relatively poor vacuum, which in turn was probably caused by a leak in the inner face of the honeycomb vacuum jacket. The vessel was modified in an attempt to evacuate this honeycomb space separately. The insulation space pressure decreased to about 5×10^{-6} mm Hg and the heat leak dropped to 17 Btu/hr. These heat leaks were determined from the boiloff of liquid nitrogen. The vent line was then closed, allowing the pressure to increase. This test indicated a heat leak of 21 Btu/hr. The vessel was then emptied and refilled with liquid hydrogen. Steady

state boil-off measurements with liquid hydrogen indicated a heat leak of 17.7 Btu/hr. The vent line was then closed allowing pressure to increase. Results from this method indicated a heat leak of 26.3 Btu/hr.

D. COMPARISON OF TEST RESULTS WITH PREDICTED RESULTS

In this section, the estimated distribution of the observed lightweight vessel heat leak and its distribution predicted from the boiler-plate vessel test results are compared.

1. Heat Leak

Table VI presents the estimated distribution of heat leak in the boiler-plate vessel and also lists predictions of the equivalent values for the lightweight vessel. These predicted values were based on the results obtained from the boiler-plate vessel test and on the evaluation of the effect of design modifications made on the lightweight vessel. In addition, this table includes the actual observed heat leak and the estimated heat leak distribution for the lightweight vessel. Conditions are similar in all cases, viz., the test fluid is nitrogen and the vessels are venting at one atmosphere pressure.

TABLE VI

HEAT LEAK DISTRIBUTION - TEST VESSELS

<u>Component</u>	<u>Boiler-Plate Vessel Test Results</u>	<u>Predicted for Light- weight Vessel</u>	<u>Lightweight Vessel Test Results</u>
Insulation	4.1 Btu/hr	4.3 Btu/hr	7.1 Btu/hr
Support Discs	6.6	2.3	2.3
Support Rod	1.0	1.0	1.0
Piping*	4.4	1.4	3.9
End Radiation	2.8	2.8	2.8
Total Heat Leak	18.9 Btu/hr	11.8 Btu/hr	17.1 Btu/hr

*In the case of venting, piping heat leak refers essentially to one pipe only.

The estimated heat leak distribution for the boiler-plate vessel was adapted from data given in Appendix P.

The end cap radiation in both vessels is based on an effective emissivity factor of 0.1. The end radiation is known to have been reduced by about 3 Btu/hr in the final tests of the boiler-plate vessel as a result of a foil lining of the end cap components (see Appendix P). This corresponds to a reduction in emissivity from 0.3 to 0.1 for these components. This modification was incorporated into the lightweight vessel, so no further change in this value was expected. The support rod contribution in the boiler-plate vessel is based on an emissivity of 1.0. No modification in this component was made in the lightweight vessel.

Heat leak values for the support discs were measured experimentally in a test calorimeter. These data are discussed in detail in Appendix M. The final tests of the boiler-plate vessel utilized discs wound from 1/2-inch wide, stainless steel strip which was dusted with 200-mesh manganese dioxide. The lightweight vessel utilized discs wound from 1/4-inch wide, stainless steel with a Nylon spacing strip.

The calculated piping heat leak in the boiler-plate vessel corresponds to an emissivity of 1.0 for the fill and vent lines. The value of 1.4 Btu/hr, predicted for the lightweight vessel, is based on an assumed emissivity of 0.1. The "observed" value of 3.9 Btu/hr is based on the fact, reported in Appendix Q, that the apparent heat leak for a non-venting lightweight vessel was 3.9 Btu/hr higher than that for a venting vessel. This difference is considered to be a rough measure of the heat leak down one of the two tubes. This is so, because the heat leak down the vent pipe during venting increases the sensible heat of the escaping vapor and does not contribute to the boil-off rate. Therefore, the heat leak value shown in Table VI is for essentially one pipe only. The heat leak attributed to the insulation in the boiler-plate vessel is based on published heat flux data⁽¹¹⁾ multiplied by a factor of three to compensate for joints and variations in wrapping technique. The small predicted increase in heat leak through the insulation for the lightweight vessel is the result of a reduction in the thickness of the end blanket from 3 to 2 inches. (The cylindrical portion of the insulation was 2 inches thick for both vessels.) The "observed" insulation heat leak in the lightweight vessel is the difference between the total observed heat leak and the sum of the values for the other components. The high value indicated is qualitatively consistent with the experimental conclusion that the insulation may not be operating properly due to high gas pressure.

A further word is necessary concerning the piping heat leak, which was unexpectedly high. As was stated above, the difference between the heat leaks observed during non-venting and venting tests indicates the heat leak down one pipe. The results reported in

Appendix Q indicate 3.9 Btu/hr-tube and 8.6 Btu/hr-tube for liquid nitrogen and liquid hydrogen, respectively. These high values are now considered to be due to the extremely high thermal conductivity of the silver plating on the piping. The pipes were plated to reduce their emissivity.

The plating thickness was measured to be between 0.00015 inches and 0.0002 inches, while the tube thickness was 0.010 inches. The thermal conductivity of high purity (99.999%) silver and type-321 stainless steel (tube material) is shown as Figure 13. The thermal conductivity of high purity silver is seen to be approximately 30 to 3000 times that for the tube material in the temperature range of interest. Although the tubing wall is 50 to 70 times thicker than the plating, the product (conductivity) $\times$ (thickness) of the plating can be very significant compared to that for the tubing wall. At low temperatures, axial conduction along the plating is probably the principal mode of heat transfer for the piping. There is no way to accurately determine the plating purity, a parameter which strongly affects the conductivity. Therefore, an analytical solution for the heat leak along the pipes is not possible. It should be noted, however, that the indicated increase in piping heat leak as the fluid temperature is decreased is qualitatively consistent with the variation in the thermal conductivity of high purity silver with temperature.

It is therefore concluded that silver plating is not the most promising technique for reducing the emissivity of the piping. A more promising way may be to wrap the tubes with a few layers of aluminized Mylar.

2. Weight

The total measured weight of the experimental vessel is 103 pounds. The distribution of this measured weight, together with the predicted weight distribution(5) is shown as Table IV, which was presented in Section IIB of this report.

It may be seen in the table that the major contributor to the increased vessel weight over the earlier weight prediction is the increase of 22.5 pounds in the weight of the inner vessel. This problem is discussed more fully in Appendix K. In addition, the honeycomb cylinder was overweight by 3.6 pounds, which was probably caused by an increase in adhesive weight. A reduction of 6.3 pounds is attributed to the use of a thin, fabric-spacer-type of support disc as discussed in Appendix M. All other actual weights were quite close to the estimated values.

Figure 13. THERMAL CONDUCTIVITY OF PLATED PIPING MATERIALS

APPENDIX A

ENERGY DEPOT SYSTEM CRITERIA

(Letter May 2, 1963 from Robert E. Berlin, Chief MPB,
Reactor Development Division AEC-NYOO, to Richard G. Clark)

1.0 General

- 1.1 A single Energy Depot system shall be designed to maximize the energy available to the user. Specific output goals will depend on both operational and technical feasibility as developed during the program. Equipment including the nuclear power plant contained within the depot will be minimized consistent with providing the required reliability.
- 1.2 The only raw materials available in quantity are air and water. It should be borne in mind that water may be scarce in particular world areas, therefore large volumes of cooling water are not available. Additives may be used provided the quantities are so small that no logistics burden is encountered in transporting a 30-day supply. The depot should be capable of either regenerating reactants or generating fuel from the available raw materials.
- 1.3 The Energy Depot system should not introduce nullifying safety, maintenance, or logistic problems, nor should it impose significantly unattractive characteristics on the vehicles or equipment that it supports.
- 1.4 For planning purposes, three reactor types are assumed to be available in the near future. Power output are 500 kwe, 3000 kwe, and 10,000 kwe. Degree of mobility roughly correspond to the ML-1, MCR, and MH-1A. The primary source of nuclear energy for Energy Depot use is considered to be the MCR which is suitably modified to generate the power required by processing equipment, i.e., 400 cycle alternators will be used if a significant weight advantage may be realized.

2.0 Operations and Applications

- 2.1 Modules of the system should have land mobility compatible with that of the units they support and be capable of operation in the fast moving, flexible environment required for mechanized task forces without requirement for elaborate or time consuming installation procedures. The energy utilization device should be as small, lightweight, reliable, and long-lived as practicable. It must be well adapted to vehicular propulsion.

- 2.2 The system shall be capable of refueling vehicles and equipment on short notice in a period equal to or less than that now required for petroleum refueling.
- 2.3 All equipment shall be capable of 24 hours per day operation, but production rates will be estimated using an average operating day of 16 hours. The remaining time is to be used for maintenance, relocation, shut-down time, and buildup of stored energy in the form of fuel.
- 2.4 Vehicle operational endurance with Energy Depot fuels or regenerated reactants will be at least equal to that afforded by conventional fuels.
- 2.5 There will be a minimum of operating personnel. Control systems should be centralized.
- 2.6 The use of air as an oxidant for Energy Depot fuels should be a goal for the energy utilization device development.
- 2.7 The preferred system will incorporate only one nuclear power plant per Energy Depot for ease of control and operational deployment.

3.0 Maintenance, Lifetime, Reliability

- 3.1 The system will be designed to have 95% reliability.
- 3.2 Major overhaul interval is 5000 operational hours with a unit lifetime of 25,000 operation hours. Units will be designed for minimum maintenance between major overhauls.
- 3.3 The system shall be capable of operation under conditions as specified in AR 705-15 "Operation of Material Under Extreme Conditions of Environment". The output of units shall not be downgraded to less than 75% of design values under any conditions listed in the AR above. "Cold weather kits" may be specified if required for low temperature operation.
- 3.4 Connections between packages shall permit easy, rapid assembly and disassembly; a minimum number of connections will be employed.
- 3.5 Maximum protection from small arms fire as to be incorporated without undue weight or size limitations.
- 3.6 Only a minimum of special tools and equipment will be available for field maintenance. All equipment required for movement or operation of the Energy Depot must be an integral part of the system. However, special equipment for reactor refueling will not normally be a part of the system.

3.7 Reactor core life shall be of the order of 5,000-10,000 full power hours.

4.0 Safety

4.1 Vulnerability of the Energy Depot shall not be greater than that of a conventional fuel depot or power generating equipment.

4.2 System shall be of fail-safe design.

4.3 Operating crew will not be subject to hazards (other than normal low radiation exposure from a reactor) in excess of a conventional chemical or power plant and associated fuel supplies.

5.0 Transportability

5.1 Components of the system should be compact and lightweight and should be transportable in Phase III of airborne operations. The plant should consist of convenient modules or packages. The structures of the packages should provide adequate support and housing for plant equipment both during transport and operation. Modules should be designed for convenient handling and transport and may be either skid or wheel mounted, depending on final characteristics and application.

5.2 Module weight should not exceed 30,000 pounds. The dimensions should be a minimum but may not exceed 8-1/2 by 8-1/2 by 24 feet.

5.3 Applicable regulations include:

AR 705-8 Department of Defense Engineering for
Transportability Program

AR 705-35 Criteria for Air Transportability and
Air Delivery of Material

MIL-A-8421B (USAF) Air Transportability Requirements

6.0 Cost

Equipment cost will be as low as possible, consistent with other criteria.

APPENDIX B

DESIGN CRITERIA ENERGY DEPOT
FUEL STORAGE AND DISTRIBUTION

(Letter June 21, 1963 Robert E. Berlin, Chief MPB,
Reactor Development Division, AEC-NYOO, to Richard G. Clark)

I. General

This document supplements the general criteria for Energy Depot systems, "Nuclear Powered Energy Depot System Criteria", in the specific area of fuel handling and distribution. It is to be used as a guide in studies of anhydrous ammonia and liquid hydrogen storage, handling, and distribution. Since this criteria is applicable to conceptual design studies it is anticipated that contractor assumptions will be required within the general guidance of this document, the system criteria, and applicable regulations and specifications as therein quoted.

II. Operations

- A. The philosophy of fuel distribution from Energy Depots to the user involves bulk transport directly from the production unit to the fuel tank at the user location. The following definitions apply:
 - 1. Primary storage -- those vessels, vehicular mounted, which serve as bulk storage for depot output and transport vessels from depot to user.
 - 2. Secondary storage -- those vessels designed to store fuel for the user at his location or in his equipment. These tanks are generally at least an order of magnitude less in capacity than primary storage units.
 - 3. Primary transfer -- transfer of depot fuel from the production facility to the primary storage.
 - 4. Secondary transfer -- transfer of fuel from the primary storage to the secondary storage.
- B. No integral storage will be designed into depot equipment other than the tanks that may be required to insure continuous operation of the fuel producer. Primary storage will be accomplished through bulk storage transport vehicles. This does not preclude secondary storage of fuel in the vehicles required for Energy Depot mobility.
- C. A sufficient number of primary storage units will be used with each depot fuel manufacturing plant to provide a capacity for storing the full output of 96 hours continuous operation. Primary storage units will be in the 3000-5000 gallon size range.

- D. Loss rates from storage vessels and transfer lines for perishable fuels will be a minimum. The acceptable fuel retention rates for storage vessels are as follows:
1. Primary Storage - 85% of an original full load of fuel will be deliverable to the secondary storage after seven days of storage.
 2. Secondary Storage - 75% of an original full load of fuel will be deliverable to the utilization device (engine, fuel cell, etc.) after seven days of storage.
 3. Conditions will be selected from AR 705-15 that will cause maximum loss rate when applying these limitations.
- E. Storage vessels will not be designed solely to meet the criteria imposed by various codes and specifications not directly applicable to military equipment. Contractors should be cognizant of deviations from such codes or specifications (e.g. I.C.C. regulations) and aware of the possibility of later re-design within these restrictions.
- F. Crew requirements for operation of Energy Depot Systems will be held to a minimum. Equipment for primary and secondary transfer should be capable of being operated by one man with only limited technical training.
- G. Secondary transfer rates will be such that secondary storage of 50 gallons capacity and greater can be filled at a minimum rate of 50 gallons per minute. Secondary storage of less than 50 gallons will be filled in one minute or less.
- H. Equipment shall be capable of operation under the extreme conditions of environment specified in AR 705-15.

III. Vulnerability

Systems and components will be designed for maximum protection from small arms fire and blast damage without undue weight or size penalties. Armor plating is not required specifically for this purpose.

IV. Transportability

- A. Typical shock and vibration data as experienced by military vehicles during various operating conditions is incorporated as an attachment. This information will be used as a guide in designing any vehicular or vehicular mounted components for depot fuel storage and distribution equipment.

B. All components of depot systems, when assembled, must not exceed 132 inches in height. (AR 705-8).

C. Shock and vibration protection for transport of Energy Depot components by various transportation methods will be consistent with the following:

1. Emergency landing shock loads during transit by air with no loss of serviceability.

	<u>Loading</u>	<u>Duration</u>
Forward	4.5 g	0.1 sec
Sideward	1.5 g	0.1 sec
Vertical (up)	3.0 g	0.1 sec
Aft	2.0 g	0.1 sec

2. Emergency landing shock loads during transit by air with questionable serviceability.

	<u>Loading</u>	<u>Duration</u>
Forward & Aft	8 g	0.1 sec
Vertical	4.5 g	0.1 sec
Sideward	1.5 g	0.1 sec

3. The data of the attachment pertaining to vibration will be used as guidance for design. Equipment must remain serviceable under the conditions specified therein.

D. Gradeability and side slope criteria to be met by Energy Depot equipment will be correlated with the general specifications outlined below:

1. Combat and tactical vehicles must be capable of negotiating a 60 per cent grade of smooth, dry concrete and must be able to brake adequately on the same grade.
2. Wheeled vehicles with towed trailers must be able to negotiate and brake adequately on a 30 per cent slope.
3. Vehicles are required to operate on side slopes as follows:
 - a. tactical vehicles - 20 per cent
 - b. combat vehicles - 30 per cent
 - c. jeep-type vehicles - 40 per cent

V. Maintenance

- A. Energy Depot equipment should be so designed as to be capable of maintenance during severe military use by means of readily available skills, tools, and supplies wherever possible.
- B. Routine maintenance required to maintain equipment at an acceptable level of performance should be done at the lowest echelon possible.
- C. Maximum interchangeability of components and sub-components is desired to reduce repair parts supply, maintenance, etc.

VI. Safety

The nature of Energy Depot fuels dictates that a great deal of attention be given to the safety aspects of fuel storage, distribution, and utilization. In Energy Depot studies consideration will be given to but not be limited to the following:

- 1. Hazards associated with flammability of fuels.
- 2. Hazards associated with toxicity of fuels.
- 3. Safety valves and rupture discs on fuel tankage and other safety mechanisms as appropriate.
- 4. Interactions with other materials in the environment of intended use.
- 5. Special protection of personnel such as gloves, coats, goggles, respirators, etc.
- 6. Detection methods for leaks and fires.
- 7. Results of battle damage to various components of systems.
- 8. Venting of fuel to atmosphere and the hazards resulting from such action.

SHOCK AND VIBRATION DATA

Type of Operation	Part of Vehicle Considered	Shock (Accel. in g's)	VIBRATION					
			Vertical		Longitudinal		Transverse	
			g's	cps	g's	cps	g's	cps
High Speed on Hard Pavement	Hull		4	500	3.8	500	2.3	520
	Instr. Panel		2.6	300	1.8	400	2	350
	Eng. Mount.		12.5	450	15	900	14.1	650
	Generator		10.3	650	18.7	700	18	800
Medium Speed Off-the-Road	Hull		2.3	540	2	520	0.6	430
	Instr. Panel		1.2	120	1.3	120	0.9	120
	Eng. Mount.		11.4	500	18.7	900	11.3	850
	Generator		8	650	10	700	25.4	900
Low Speed Rough Terrain	Hull	8						
	Instr. Panel							
	Eng. Mount.		5	500	10.8	850	13	700
	Generator		11	550	10	750	14	900
	Axle (Semitrailer)		9.4	300	3.6	350		
					5.4	10	2.5	400
	Fifth Wheel Plate		36.4	150	21.9	400	12.2	100
	Cargo Bed Above Fifth Wheel		14.5	100	4.8	250	2.3	30
	Cargo Bed Above Axle (Semitrailer)		3	3			0.8	16
							0.4	90
Side Wall of Van Semitrailer			2.4	20			3.3	40
Shipment by Truck	Vehicle Assembly	8	2	300	2	300	2	300
Shipment by Rail	Vehicle Assembly	20	2	70	2	70	2	70
Shipment by Fixed Wing Aircraft	VEHICLE ASSEMBLY							
	Fwd.	8			5	300		
	Side	1.5					5	300
							0.25	10
Vert. (up)		3	5	300				
	Aft.	1.5	0.5	10				

SHOCK AND VIBRATION DATA

Type of Operation	Part of Vehicle Considered	Shock (Accel. in g's)	VIBRATION					
			Vertical		Longitudinal		Transverse	
			g's	cps	g's	cps	g's	cps
Shipment by Rotary Wing Aircraft	VEHICLE ASSEMBLY Fwd.	4						
	Side	1.5						
	Vert. (up)	2						
	Aft.	2						
Parachute Drop	Vehicle Assembly	16						
Ballistic Impact	Turret	20(0.75" ampl.)	50	1000			140	600
HE Blast	Turret & Hull	25(1.0" ampl.)						

APPENDIX C

CONCEPTUAL LIQUID HYDROGEN STORAGE AND
DISTRIBUTION SYSTEM: PROCESS EQUIPMENT DESCRIPTION

I. Primary Storage Unit

The conceptual layout of the primary unit has been presented as Figure 1. The estimated weight distribution for this unit is tabulated in Table I. Figure 8 includes a flow schematic of the primary unit. The following major items of equipment will comprise the primary unit.

Vacuum-Jacketed Storage Vessel - The liquid hydrogen will be contained in a single inner vessel with a net capacity of about 3,200 gallons. The vessel will be of welded construction and fabricated from aluminum alloy-2219. Operating pressure will be approximately 32 psia. The vessel will be wrapped with a multilayer reflective-type insulation and mounted in an evacuated jacket. The jacket will be fabricated from a honeycombed aluminum alloy. Closures for both the inner vessel and the jacket will be formed heads.

Pressure Raising Circuit - The pressure head required for transfer of liquid hydrogen from the primary vessel to a secondary vessel will be obtained by allowing liquid hydrogen to flow from the bottom of the primary vessel into an ambient-type pressure raising coil located on the bed of the GOER. This liquid will be vaporized in the coil, and the resulting vapor will be returned to the top of the vessel, thereby increasing the gas phase pressure. Flow of hydrogen gas to the vapor space will be automatically controlled by an air-actuated control valve, which will operate on a signal from a differential-pressure indicating-transmitter. A manually-operated by-pass valve is to be located in the line for increased reliability, and will be used only in the event of power failure or malfunction of the automatic control valve. A block valve will be located between the vessel and the coil to shut off the supply of liquid during emergencies or maintenance. A safety valve will be provided to protect the coil in the event it is isolated with cold gas or liquid in it. The coil itself is to be of the ambient-air, parallel-flow type capable of generating enough vapor to effect transfer rates in the approximate range from 50 gpm to 100 gpm.

Some consideration was given to the possibility of using an external pump as a means of transferring liquid from the primary unit to secondary units. A two-horsepower, 100 gpm capacity pump would require approximately eight gallons of liquid hydrogen for cool-down. The utilization of a pump would require some additional equipment to maintain a positive liquid pressure at the inlet of the pump. This could be accomplished by adding a small pressure raising coil, or an electric resistance heater with a minimum power output of 3.9 kilowatts. Coupled with its power requirement for adequate operation, and the amount of hydrogen lost in cool-down and the decreased system reliability, an external pump is not considered the most promising means of effecting liquid transfer. Also considered was a cold pump located inside the

vessel. This arrangement would avoid pump cool-down. Pump reliability, the inaccessability of the pump for repairs, and shaft sealing appear to be the major problem areas here. While more consideration should be given to this approach, the pressure raising coil was selected as most promising for the present study.

Transfer Line - The transfer line will consist of a lightweight vacuum-jacketed pipe, a control valve, and an insulated highly flexible metal hose. The hose will be equipped with valving on both ends and must be capable of repeated flexing when cold. The high cold flexibility combined with low heat leak and weight characteristics is not available in present day flexible hose and must be developed. The valves are to be quick-disconnect, double-check, bayonet-type couplings providing positive shutoff upon disconnect. This feature will eliminate leakage of air or other contaminants into the transfer line. The couplings envisioned for this application are not commercially available at this time. It is considered that the coupling configuration shown in Figure C-1 will provide a satisfactory solution to this problem.

The main control valve will be air actuated and will operate on a signal from a temperature sensor (to open) and a liquid level sensor (to close), both located in the secondary vessel. A small hole is to be drilled in the inlet side of the valve stem to permit a low quantity flow of liquid hydrogen during the cool-down period. A manually operated, vacuum jacketed by-pass valve will be located in the line for increased reliability, and will be used only in the event of power failure or malfunction of the automatic control valve.

Vent and Safety System - The vent and safety system valves are to be mounted in parallel on uninsulated piping, and consist of the following:

- . A spring-loaded safety valve to prevent pressure in the inner vessel from exceeding the maximum allowable working pressure (MAWP);
- . A solid rupture disc to prevent pressure in the inner vessel from exceeding 1.50 MAWP;
- . A solenoid-operated vent valve for blowing down the inner vessel;
- . A manually operated block valve to bypass the normal pressure relief system during long-term storage (see below);
- . A spring-loaded check valve to keep the back pressure in the vent line between 15 and 16 psia (This valve will serve to keep air and other contaminants out of the vent line and to hold the inner vessel pressure at near atmospheric pressure during long-term fuel storage); and

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				SCALE	
			Figure C-1. COUPLING CRYOGENIC LIQUID, QUICK-DISCONNECT, DOUBLE-BLOCK-AND-BLEED, BAYONET-TYPE		
					NO. 67591A

- An uninsulated flexible hose with a quick-connect coupling to return the vapor generated during primary transfer to the plant for reliquefaction.

Compressed Air System - The compressed air system will consist of an air reservoir, air regulator, solenoid valves, and uninsulated tubing. The air supply will be obtained from the pneumatic braking system of the GOER. The air regulator will be set to give a downstream pressure of 40 psig for operation of the control valves. The solenoid valves will provide for individual operation of the control valves located in the pressure raising circuit (V-109) and the transfer line (V-106). The solenoid valve controlling V-109 will operate on an electrical signal from a differential-pressure indicating-transmitter. When the differential pressure between the primary and secondary vessels drops below a preset value, the solenoid valve will open, permitting air to flow into the control valve actuator, thus opening the valve. When the differential pressure between the vessels reaches the desired value, the solenoid valve will close, and the air in the valve actuator will exhaust to the atmosphere. The solenoid valve controlling V-106 will operate on electrical signals from a temperature sensor and a liquid level sensor located in the secondary vessel. When the temperature at the secondary vessel inlet reaches -350°F , the solenoid valve will open, permitting air to flow into the control valve actuator, thus opening the valve. When the secondary vessel is full, the solenoid valve will close, and the air in the valve actuator will exhaust to the atmosphere.

It should be stated in passing that pneumatically operated valves controlled by solenoid valves were chosen for V-109 and V-106 because of their proven reliability in cryogenic service. Motor-operated valves were not considered adequate for this application due to their comparatively long response time and questionable reliability. Pilot-operated solenoid valves were also considered. These depend on the flowing fluid for actuation and have developed repeated malfunctions when used with cryogenic fluids. Therefore they were not selected for this application.

Instrumentation - All instruments are to be mounted on the control console at the rear of the GOER and will consist of the following:

- Primary vessel liquid level indicator - a panel of pinpoint lights actuated by an electrical signal from a series of thermistors (solid state devices) located in the vessel;
- Secondary vessel liquid level indicator - same as primary vessel liquid level;
- Primary vessel pressure indicator - Bourdon tube-type gage connected to the vapor space of the primary vessel;
- Air supply pressure indicator - Bourdon tube-type gage connected to the air line downstream of the air regulator;

- . Differential-pressure indicating-transmitter - bellows-type, connected to the vapor space of the primary and secondary vessels (The gas pressure of the secondary vessels is to be read by means of a light, flexible hose connecting C-102 and C-202.); and
- . Switches - to energize controller, select transfer rate and operate various valves.

System Controller - All controls are to be housed in the system controller and will consist of transmitters, relays, wiring, etc., necessary for automatic operation of the unit.

II. Secondary Storage Unit

The conceptual layout of the secondary unit has been presented as Figure 2. Table I contains the estimated weight distribution for the secondary fuel unit. Figure 8 includes a flow schematic of the secondary fuel system for which the major items of equipment are described below.

Vacuum-Jacketed Storage Vessels - According to the existing conceptual design of the hydrogen-fueled M-113⁽⁷⁾, the liquid hydrogen will be stored in four interconnected inner vessels arranged in pairs, each pair being mounted within a single outer vacuum jacket. The general vessel configuration is represented later as Figure E-2. Each inner vessel will contain approximately 60-gallons of liquid hydrogen. The vessel materials of construction will be the same as those stated for the primary vessels. The liquid line connecting the right-hand and left-hand vessels will consist of lightweight, vacuum jacketed pipe and air-activated control valves.

Vaporizing Equipment - Hydrogen gas required to operate the fuel cells will be obtained by allowing liquid hydrogen to flow from the storage vessel to an electric resistance heater. A signal from a pressure indicating-transmitter in the feed gas line will be used to modulate the heater current and insure sufficient hydrogen flow to the fuel cell. This arrangement will provide immediate response to changing demands from the fuel cell. Flow of liquid hydrogen to the heater will be automatically regulated by an air-activated control valve, which will operate on a signal from a liquid level sensor located in the heater. Due to the multivessel arrangement, two of these control valves will be required in order to prevent flooding of the vent system (Refer to Appendix E). The control valves and piping between the storage vessels and the heater are to be vacuum jacketed.

The heater was rated for 1.25 kw, the power required to vaporize a maximum of about 18.3 lb H₂/hr. This flow rate approximates that for full power operation of the fuel-cell powered M-113 vehicle⁽⁷⁾. Figure C-2 presents the heater arrangement envisioned for this application. The design involves a low mass, bare heating element immersed in the liquid hydrogen to achieve fast response.

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			Figure C-2. ELECTRIC RESISTANCE HEATER-TYPE VAPORIZER			NO. 67592A

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A vaporizer utilizing heat from ambient air was considered as the source of hydrogen gas for the secondary unit. Compared to the electric vaporizer, the ambient-heat vaporizer is considered to be less attractive from the standpoint of required volume and its effect on the environment if located inside the armor of the secondary vehicle. If located outside the vehicles armor, vulnerability to blast damage will be a problem. This entire question should be re-examined for each particular secondary vehicle under consideration. For the purpose of this study, the electric vaporizer was selected as most promising for an armored vehicle with a wide operating power range.

Transfer Line - The transfer line will consist of a lightweight, vacuum-jacketed pipe and control valve. The end of the pipe will be equipped with a quick-connect coupling, the mate of which will be located on the transfer hose of the primary unit. The control valve will be air-activated and will operate on a signal from a temperature sensor on the valve (to open) and liquid level sensors in the inner vessels (to close). A small hole will be drilled in the inlet side of the valve body and will be connected to the vent and safety system through a length of tubing. By means of this arrangement, the warm hydrogen gas generated in cooling down the transfer line and valve will bypass the cold storage vessels.

Fuel Cell Supply Line - The fuel cell supply line will consist of piping and a control valve. The air-activated control valve will operate on a signal from the starting mechanism of the secondary vehicle. Provision must be made to warm the cold gas to ambient temperature for delivery to the fuel cells. The technique used will depend on the exact piping layout for the vehicle being designed. Both ambient air and electric-types will be considered.

Vent and Safety System - The vent and safety system valves are to be mounted in parallel on uninsulated piping and will consist of the following:

- . A spring-loaded safety valve to prevent the pressure in the inner vessels from exceeding 32 psia, the operating pressure;
- . A solid rupture disc to prevent the pressure in the inner vessels from exceeding 48 psia, which is 1.5 times the operating pressure;
- . A solenoid-operated vent valve for blowing down the vessels;
- . A spring-loaded check valve to maintain a slight positive back pressure in the vent line to keep air and other contaminants out of the line;
- . A solenoid-operated block valve in the line used to bypass gas generated during transfer line cooldown directly to the vent stack; and

- . A spring-loaded safety valve to prevent pressure buildup in the above bypass line due to warmup of trapped cold gas.

Since flooding of the vent system will be possible when the vehicle is negotiating grades as specified in the design criteria, special consideration was given to the arrangement of the vent piping. The results and recommendations of this phase of the study are presented as Appendix E.

Compressed Air System - As was the case for the primary fuel system, pneumatically operated valves were selected for their proven reliability. The compressed air system will consist of an air reservoir, air regulator, solenoid valves and tubing. The regulator will maintain a downstream pressure of 40 psig for operation of the control valves. If the secondary vehicle will be equipped with pneumatic brakes, the air supply will be obtained from the braking system; otherwise, a compressed air cylinder will be required. A cylinder rated for 250 SCF capacity at 2260 psi will contain enough air to perform approximately 80 transfer operations. In actual practice, the air bottle will have to be recharged more frequently than this, due to the control valve air requirements during operation of the vehicle. The assumption is also made that the air system is free of leaks.

Fuel transfer from a primary to a "dead" secondary is an important capability. When the secondary unit is not furnished with a compressed air cylinder, the air supply of the primary system can be hooked into the instrument air lines of the secondary system to facilitate automatic fuel transfer operation. The small couplings necessary for this hookup are not shown in Figure 3, but can be placed in any convenient location. Although a "dead" secondary vehicle can be fueled, it must be supplied with instrument air before it can be operated.

The source of air consumption, other than leakage, will be the operation of the control valves. Solenoid valves will activate the control valves located in the vaporizing system (V-214 and V-216), the transfer line (V-206), the fuel cell supply line (V-209), and the piping connecting the storage vessels (V-212). The solenoid valves controlling V-214 and V-216 will operate on an electrical signal from the starting mechanism of the secondary unit vehicle. When the vehicle is started, one of these solenoid valves will be opened, depending on which storage vessel is to be used, to permit air to flow into the associated control valve actuator, thus opening the valve. After the vehicle has been started, the control of these solenoids will be taken over by the liquid level sensor located in the heater. When the heater is full, the solenoid valve will close and the air in the valve actuator will discharge to the atmosphere. When the level drops to a predetermined point, the solenoid valve will reopen. When the operator switches from one vessel to the other, the control function will switch automatically from one solenoid valve to the other. The solenoid valve controlling V-206 will be operated on electrical signals from a temperature sensor and a liquid level sensor located in the

storage vessel. When the temperature at the secondary inlet reaches -350°F during cool-down, the solenoid valves will open permitting air to flow into the control valve actuators, thus opening the valves. When the vessel is full, the solenoid valve will close, and the air in the valve actuators will exhaust to the atmosphere. The solenoid valve controlling V-209 will operate on an electrical signal from the starting mechanism of the secondary unit vehicle. When the vehicle is started, the solenoid valve will open permitting air to flow into the control valve actuator, thus opening the valve. When the vehicle is de-energized, the solenoid valve will close, and the air in the valve actuator will exhaust to the atmosphere.

Instrumentation - All instruments are to be mounted on the control console and consist of the following:

- . Secondary vessel level indicator - a panel of pinpoint lights activated by a signal from a series of thermistors (solid state devices) located within the vessels;
- . Secondary vessel pressure indicator - Bourdon tube-type connected to the vapor space of the inner vessel;
- . Air supply pressure indicator - Bourdon tube-type connected to the air line downstream of the air regulator;
- . Fuel cell supply pressure indicating-transmitter - Bourdon tube-type connected to the fuel cell supply line; and
- . Switches - to energize the controller, select storage vessel supplying heater and operate various valves.

System Controller - All controls will be housed in the system controller and consist of transmitters, relays, wiring, etc., necessary for automatic operation of the unit.

APPENDIX D

CONCEPTUAL LIQUID HYDROGEN STORAGE AND DISTRIBUTION

SYSTEM: GENERAL OPERATING PROCEDURE

Tables D-I and D-II define seven operating conditions and the state of the equipment for each condition for the primary and secondary units, respectively. The component designations are those identified in Figure 8, to be found in Section IVB.

Although a high degree of automation has been designed into both the primary and secondary systems, certain manual duties must be performed during normal operations. The following is a description of the system operation in terms of the manipulations to be performed by the operator. The conditions refer to those defined in Tables D-I and D-II.

Condition (1) - To prepare the primary vessel for filling, the operator must connect a small electrical cable (EC-102), a small hose (C-105), and the transfer hose (C-101), in that order, to the delivery manifold of the plant. The filling operation and shutoff will be controlled automatically from the plant. When the primary vessel is full, the operator will disconnect the above-mentioned couplings in the reverse order.

Condition (2) - Prior to placing the GOER in a storage area, the block valve (V-112) in the vent system should be opened. This will keep the pressure in the inner vessel in the range of 15-16 psia, thereby increasing the overall efficiency of the system.

Condition (3) - When the GOER is taken from long-term storage, the valve (V-112) should be closed before leaving the storage area. If the GOER is to be taken directly from the plant to user vehicles, V-112 will remain closed and no manual corrections need to be made to the system.

Conditions (4) and (5) - Prior to transfer of liquid hydrogen, the primary system controller must be energized and the desired transfer rate selected by operation of switches which will set up the appropriate pressure driving force. The operator will then connect the electrical cable (EC-102), the small gas hose (C-102), and the transfer hose (C-101), in that order, to the secondary vehicle. The cool-down, transfer and shutoff will then be performed automatically. When the secondary vessel is full, the operator will de-energize the primary system controller and will disconnect couplings C-101, C-102, and EC-102, in that order.

Condition (6) - When the operator parks the secondary unit, he will merely turn off the drive unit (similar to turning off an automobile engine). No manual corrections need to be made to the system.

Condition (7) - To increase the flow of hydrogen gas to the fuel cell, the operator needs only to depress the accelerator lever of the drive unit. In the system considered, liquid will be drawn from one storage vessel at a time. When a gage on the dashboard indicates the tank to be empty, the operator will operate a switch to put the other vessel into service.

TABLE D-I

STATE OF EQUIPMENT DURING OPERATION - PRIMARY UNIT

- Condition (1) - Filling of primary unit
 Condition (2) - Long-term liquid storage in primary unit
 Condition (3) - Primary unit en route to rendezvous with secondary unit
 Condition (4) - Primary unit cooling down transfer lines when coupled to secondary unit
 Condition (5) - Transfer from primary to secondary unit
 Condition (6) - Secondary unit filled, or partially filled, and not in use
 Condition (7) - Vapor generation in secondary unit when in use

COMPONENT	CONDITION						
	(1)	(2)	(3)	(4)	(5)	(6)	(7)
C-101	Conn.	Disc.	Disc.	Conn.	Conn.	N.A.	N.A.
C-102	Disc.	Disc.	Disc.	Conn.	Conn.	N.A.	N.A.
C-103	Conn.	Conn.	Conn.	Conn.	Conn.	N.A.	N.A.
C-104	Conn.	Conn.	Conn.	Conn.	Conn.	N.A.	N.A.
C-105	Conn.	Disc.	Disc.	Disc.	Disc.	N.A.	N.A.
EC-101	Conn.	Conn.	Conn.	Conn.	Conn.	N.A.	N.A.
EC-102	Conn.	Disc.	Disc.	Conn.	Conn.	N.A.	N.A.
Sys. Cont.	De.	De.	De.	E.	E.	N.A.	N.A.
V-101	C.	C.	C.	C.	C.	N.A.	N.A.
V-102	C.	C.	C.	C.	C.	N.A.	N.A.
V-103	D.C.	D.C.	D.C.	D.C.	D.C.	N.A.	N.A.
V-104	O.	O.	O.	O.	O.	N.A.	N.A.
V-105	O.	O.	O.	O.	O.	N.A.	N.A.
V-106	O.	C.	C.	C.	O.	N.A.	N.A.
V-107	E.C.A.	D.O.A.	D.O.A.	D.O.A.	E.C.A.	N.A.	N.A.
V-108	C.	C.	C.	C.	C.	N.A.	N.A.
V-109	C.	C.	C.	O.	O.	N.A.	N.A.
V-110	D.O.A.	D.O.A.	D.O.A.	E.C.A.	E.C.A.	N.A.	N.A.
V-111	C.	C.	C.	C.	C.	N.A.	N.A.
V-112	C.	O.	C.	C.	C.	N.A.	N.A.
V-113	O.	O.	O.	O.	O.	N.A.	N.A.

Abbreviations

O.	- Open	Conn.	- Connected	D.O.A.	- De-energized and Open to Atmosphere
C.	- Closed	Disc.	- Disconnected	E.C.A.	- Energized and Closed to Atmosphere
E.	- Energize	E.O.	- Energized and Open	N.A.	- Not Applicable
De.	- De-energized	D.C.	- De-energized and Closed		

TABLE D-II

STATE OF EQUIPMENT DURING OPERATION - SECONDARY UNIT

- Condition (1) - Filling of primary unit
 Condition (2) - Long-term liquid storage in primary unit
 Condition (3) - Primary unit en route to rendezvous with secondary unit
 Condition (4) - Primary unit cooling down transfer lines when coupled to secondary unit
 Condition (5) - Transfer from primary to secondary unit
 Condition (6) - Secondary unit filled, or partially filled, and not in use
 Condition (7) - Vapor generation in secondary unit when in use

COMPONENT	CONDITION						
	(1)	(2)	(3)	(4)	(5)	(6)	(7)
C-201	N.A.	N.A.	N.A.	Conn.	Conn.	Disc.	Disc.
C-202	N.A.	N.A.	N.A.	Conn.	Conn.	Disc.	Disc.
C-203	N.A.	N.A.	N.A.	Conn.	Conn.	Conn.	Conn.
C-204	N.A.	N.A.	N.A.	Conn.	Conn.	Conn.	Conn.
EC-201	N.A.	N.A.	N.A.	Conn.	Conn.	Conn.	Conn.
EC-202	N.A.	N.A.	N.A.	Conn.	Conn.	Disc.	Disc.
Heater	N.A.	N.A.	N.A.	De.	De.	De.	E.
Sys. Cont.	N.A.	N.A.	N.A.	De.	De.	De.	E.
V-201	N.A.	N.A.	N.A.	C.	C.	C.	C.
V-202	N.A.	N.A.	N.A.	C.	C.	C.	C.
V-203	N.A.	N.A.	N.A.	D.C.	E.O.	D.C.	D.C.
V-204	N.A.	N.A.	N.A.	O.	O.	O.	O.
V-205	N.A.	N.A.	N.A.	O.	O.	O.	O.
V-206	N.A.	N.A.	N.A.	C.	O.	C.	C.
V-207	N.A.	N.A.	N.A.	D.O.A.	E.C.A.	D.O.A.	D.O.A.
V-209	N.A.	N.A.	N.A.	C.	C.	C.	O.
V-210	N.A.	N.A.	N.A.	D.O.A.	D.O.A.	D.O.A.	E.C.A.
V-211	N.A.	N.A.	N.A.	O.	O.	O.	O.
V-212	N.A.	N.A.	N.A.	C.	O.	C.	C.
V-213	N.A.	N.A.	N.A.	D.C.	E.O.	D.C.	D.C.
V-214	N.A.	N.A.	N.A.	C.	C.	C.	C.*
V-215	N.A.	N.A.	N.A.	D.C.	D.C.	D.C.	D.C.*
V-216	N.A.	N.A.	N.A.	C.	C.	C.	O.*
V-217	N.A.	N.A.	N.A.	D.C.	D.C.	D.C.	E.O.*

* V-214 and V-215 are shown closed while V-216 and V-217 are shown open. This situation may be reversed depending on which storage vessel is supplying liquid to the heater.

Abbreviations

O. - Open
 C. - Closed
 E. - Energize
 De.- De-energized

Conn. - Connected
 Disc. - Disconnected
 E.O. - Energized and Open
 D.C. - De-energized and Closed

D.O.A. - De-energized and Open to Atmosphere
 E.C.A. - Energized and Closed to Atmosphere
 N.A. - Not Applicable

APPENDIX E

VENT AND LIQUID LINE ARRANGEMENT FOR VEHICLES

As was stated in Section IV, the M-113 Armored Personnel Carrier was selected as the representative secondary user vehicle for the purpose of this study. The preliminary conceptual design of a hydrogen-fueled version of this vehicle has recently been established(7). The present conceptual vehicle design requires that the liquid hydrogen fuel be stored in two, long slender compartments, one along each side of the vehicle. The present space allotments require that each of the two storage vessels be other than of circular cross-section. While a variety of non-circular configurations are possible, the arrangement presented later in this appendix as Figure E-2 was utilized for the purposes of this study. Long, slender cylinders are far from ideal for the storage of cryogenic liquids.* Not only heat leak and weight, but vapor venting and liquid removal become more difficult problems for multiple tanks which are horizontal, long, and slender cylinders of non-circular cross-section. It is the purpose of this appendix to discuss the venting and liquid withdrawal problems for just such an arrangement on board a vehicle.

One of the problems to be solved is the one of vent system flooding. The major disadvantages of this type of flooding in a cryogenic storage vessel is the excessive loss rate caused by the introduction of cryogenic liquid into comparatively warm pipelines. The resultant vaporization will bring about a pressure rise in the vessel, and gas may become discharged through the vent to the atmosphere and therefore become lost.

The M-113, being a tactical vehicle, must be capable of negotiating a 60% grade in the longitudinal direction and a 20% side slope (see Appendix B). Figure E-1 indicates possible piping arrangements to eliminate or minimize vent flooding for a vehicle negotiating the above mentioned grading and side slope. This sketch represents the tanks that are to be along each side of the vehicle as a single right circular cylinder.

Configuration A shows criss-crossed vent lines connected to the top of the vessel at either end. Although this configuration will eliminate the possibility of trapping any liquid in the vent line, the addition of the two nozzles through the multilayer insulation will result in an increase in the over-all heat leak into the vessel.

Configuration B is basically the same as A with the exception of the location of the nozzle connections. Running the vent lines through the ends of the supporting trunnion will eliminate the problem of additional penetrations

*The sphere is the ideal storage shape for this purpose. By exhibiting the lowest surface-to-volume ratio of any geometry and high strength, the sphere presents the lowest tankage weight and heat leak. The next best configuration, from a cryogenics view-point, is a short cylinder with a length-to-diameter ratio (L/D) in the range of 2 (See Figures 4 and 6).

Configuration A

Configuration B

Figure E-1. VENT LINE CONFIGURATIONS FOR
LONG, SLENDER, CYLINDRICAL VESSELS

through the insulation. Configuration B does have the disadvantage of trapping a small amount of liquid in one end of the vent line when negotiating a change in grade from 60% to 0% or less. The amount of liquid vaporized would be less than one-tenth of one gallon and would result in a pressure rise of less than three psi when the vessel is full. It would take approximately four minutes for the fuel cells to convert this quantity of hydrogen into usable energy at low speed conditions and eight seconds for the conversion at high speed conditions. Therefore, the liquid hydrogen vaporized in the vent line may not be lost from the system.

With the available information on the size, configuration, and operating conditions of the M-113 Armored Personnel Carrier, it is presently considered that the arrangement shown as configuration B in Figure E-1 offers the most desirable arrangement of the vent piping for this vehicle. Figure 2 "Conceptual Layout Typical Secondary Liquid Hydrogen User Vehicle", is based on this vent line arrangement.

The withdrawal of cryogenic liquid from long, slender, cylindrical vessels, rather than from short, fat vessels, on board a vehicle maneuvering over rough terrain can be readily accomplished with a sacrifice in the weight and heat leak of the system. Figure E-2 represents the selected piping arrangement for the tankage configuration utilized in this study. The valve in the line interconnecting the right-hand and left-hand vessels will be open during secondary transfer which is assumed to occur while the secondary vehicle is on relatively level ground. After transfer, the block valve will be closed. Figure E-2 represents the liquid levels within the vessels so isolated as the vehicle negotiates the specified side slope. Isolation of the liquid phase of the twin vessels is considered to be the simplest method of preventing vent line flooding on slopes.

To prevent liquid isolation at one end of the tanks while the vehicle negotiates a 60% longitudinal grade, four liquid drainage nozzles will be required as shown in Figure E-2. Without the four drainage nozzles, the possibility of isolating at least 50% of the liquid content from the liquid delivery line to the secondary's vaporizer will exist. On the other hand, the use of the four drainage nozzles will greatly increase the heat leak into the vessel thereby increasing the boil-off rate when the secondary is not in use. This will be so because the liquid must be withdrawn by gravity and the liquid lines must penetrate the insulation space at the bottom of the vessel, therefore reducing its effectiveness. The liquid lines cannot be introduced into the vessel through the end support trunnions, as can the vent lines for a horizontal vessel.

It is highly recommended that further consideration should be given to the liquid hydrogen storage location and configuration on board user vehicles. From a cryogenic standpoint the fuel storage location and configuration presently envisioned for the M-113 necessitates a comparatively inefficient and complex system for storing liquid hydrogen. It is certain that considerable improvements in fuel system weight and heat leak can be obtained by the choice of a more optimum vessel location and configuration.

View A-A

Figure E-2. LIQUID DELIVERY LINE CONFIGURATION FOR SPLIT, LONG, SLENDER, CYLINDRICAL VESSELS

APPENDIX F

ANALYSIS OF FUEL SYSTEM STORAGE AND TRANSFER EFFICIENCY

I. INTRODUCTION

This appendix is included: (1) to define the various operating modes assumed for the stockpiling and the normal models of the Energy Depot operation; and (2) to present the derivation of approximate expressions for the individual fuel losses which occur throughout the system. This information is to document the basis for the estimated fuel system efficiencies discussed in Section IVC of this report. The reader's attention is drawn to Figure F-1, a schematic representation of the fuel storage and distribution system, as an aid in understanding the various fuel losses as they are discussed. The nomenclature employed is identified at the end of the appendix.

II. DEFINITION OF THE OPERATING MODES

The following operating modes for fuel stockpiling were evaluated. These are the same modes identified previously in Table V and in Table F-1.

Mode No. 1 - The hydrogen will be delivered from the Depot to primary vessels at 16 psia (-422°F). During primary storage, the vessel will be continuously vented at 16 psia. The liquid will be transferred into secondary storage vessels which, after transfer will also be maintained at 16 psia.

Mode No. 2 - The hydrogen is to be delivered to primary storage at 16 psia (-422°F) and held during storage at 16 psia. The liquid will be transferred to secondary storage which, after transfer, will be at 16 psia because the liquid temperature is -422°F . Heat leak into the secondary vessels will be absorbed by allowing the pressure to rise to 32 psia (-418°F) at which pressure the vent will open to allow continuous venting at that pressure.

Mode No. 3 - Liquid hydrogen will be delivered to primary storage at 16 psia (-422°F). The pressure in the primary tank will be allowed to rise to 32 psia (-418°F), at which pressure the vent valve will open. After primary storage, the liquid at -418°F will be transferred into secondary storage vessels which have been blown down to 16 psia. The secondary vessels will be vented at 16 psia during storage.

Mode No. 4 - The liquid hydrogen is to be delivered to primary storage at 16 psia (-422°F). The pressure in the primary vessel will be allowed to increase to 32 psia (-418°F) at which pressure the vent will open. After transfer into secondary storage, the liquid will be held in secondary storage at 32 psia.

Mode No. 5 - The liquid hydrogen will be delivered to primary storage at 16 psia (-422°F). The pressure in the primary tank will be allowed to rise to 32 psia (-418°F), at which pressure the vent will open. The liquid at -418°F will then be transferred to secondary storage which has been blown down to 16 psia. Heat leak into the secondary vessels will be absorbed by allowing the pressure to rise to 32 psia at which pressure the vent will open.

NOTE:

SYSTEM EQUALS ONE PRIMARY AND TWELVE
SECONDARY VESSELS

Figure F-1. SCHEMATIC REPRESENTATION OF THE
FUEL STORAGE AND DISTRIBUTION SYSTEM

TABLE F-1

ESTIMATED FUEL LOSSES AND STORAGE AND TRANSFER EFFICIENCY
FOR VARIOUS OPERATING MODES

Operation Model	Operation Mode	Vessel Pressures						Fuel Losses									System Efficiency %
		Primary Storage		Transfer Period		Secondary Storage											
		P ₁ Psia	P ₂ Psia	P ₃ Psia	P ₄ Psia	P ₁ Psia	P ₂ Psia	ψ ₁ lbs	ψ ₂ lbs	ψ ₃ lbs	ψ ₄ lbs	ψ ₅ lbs	ψ ₆ lbs	ψ ₇ lbs	C lbs	FC lb	
Fuel	1	16	16	32	16	16	16	99	21.6	0	45.7	9.3	42.2	368	2285.8	1700	74.4
Stock-piling	2	16	16	32	16	16	32	92.8	21.6	0	45.3	8.8	39.7	244	2152.2	1700	79.0
	3	16	32	32	16	16	16	0	21.6	99.5	46.5	9.3	42.2	368	2287.1	1700	74.3
	4	16	32	48	32	32	32	0	21.0	0	46.8	0	86.1	366	2219.9	1700	76.6
	5	16	32	32	16	16	32	0	21.6	93.6	45.9	8.8	39.7	244	2153.6	1700	78.9
Normal Operation	6	16	--	32	16	16	--	0	21.6	14.5	42.5	7.5	35.5	0	1821.5	1700	93.4

P₁ - Pressure of unit after transfer.

P₂ - Pressure at which vent relief valve opens

P₃ - Pressure of primary unit during transfer

P₄ - Pressure of secondary unit during transfer

C - Hydrogen contained in primary unit after filling

ψ₁ - Hydrogen vented from primary unit during 168 hours of storage

ψ₂ - Hydrogen vented from primary unit transfer hose and piping after 12 secondary transfers

ψ₃ - Hydrogen flashed off and vented from transfer piping during 12 secondary transfers

ψ₄ - Hydrogen vented during 12 secondary transfers due to cooldown, heat leak, and frictional losses in transfer piping

ψ₅ - Hydrogen vented from 12 secondary units due to blowdown prior to secondary transfer

ψ₆ - Hydrogen vented from 12 secondary units as displacement gas during transfer

ψ₇ - Hydrogen vented from 12 secondary units during 168 hours of storage

FC - Hydrogen delivered to the fuel cells of 12 secondary units

The following operating mode, Mode No. 6, has been previously identified (Section IVC) as that used as the model for evaluation of a normal operating condition of the Energy Depot.

Mode No. 6 - The primary unit will be filled at the Depot with liquid hydrogen at 16 psia (-422°F). After 24 hours of primary storage, the fuel will be transferred into twelve secondary storage units. After each secondary vehicle is filled, it will proceed to consume 240 gallons of liquid hydrogen. Therefore, the secondary storage period will be zero.

II. DERIVATION OF EXPRESSIONS FOR SYSTEM EFFICIENCY

The most general definition of system efficiency is, from Figure F-1:

$$E(\%) = \frac{FC}{A} \times 100 = \frac{1700}{A} \times 100, \quad (1)$$

where A is the total quantity of hydrogen liquefied by the Depot in order that 12 user vehicles, each of 240 gallon-fuel capacity, will consume 1700 lbs of hydrogen. By inspection of Figure F-1, it is evident that:

$$A = B + C, \quad (2)$$

where B is that quantity of hydrogen vaporized due to cooldown, heat leak and friction losses in the transfer piping during filling of the primary unit, and C is the quantity of liquid hydrogen which is present in the primary unit at the completion of the filling operation. The hydrogen gas returned to the Depot in an "empty" primary unit and which is displaced from the storage vessel during filling is conceptually included in B. This contribution to B is the liquid which was vaporized in the pressure raising (PR) coil to facilitate liquid delivery during secondary fuel transfer. All of quantity B is piped directly to the plant, is reliquefied, and is subsequently returned to primary storage. For this reason B is not considered as a fuel loss for this analysis.

On this basis, the efficiency of the system is here defined as:

$$E(\%) = \frac{1700}{C} \times 100. \quad (3)$$

The definition of C is obtained by inspection of Figure F-1:

$$C = \psi_1 + \psi_2 + \psi_3 + \psi_4 + \psi_5 + \psi_6 + \psi_7 + 1700. \quad (4)$$

The following is the derivation of approximate expressions for each of the above fuel losses. For the sake of simplicity, the analysis begins with the losses incurred in the secondary unit and proceeds to those in the primary unit.

- a. ψ_7 The heat available for liquid vaporization in twelve secondary units during a 168-hour storage period can be represented by the expression:

$$Q_b = q_b - (h_c - h_d) W_b, \quad (5)$$

where q_b is determined by utilizing Figure 24 of the First Quarterly Report⁽⁴⁾.

The total weight of liquid in the twelve secondary units at the beginning of a 168-hour storage period is:

$$W_b = (1700 + W_c). \quad (6)$$

The weight of liquid vaporized in twelve secondary units during the 168-hour storage period is:

$$W_c = \frac{q_b - (h_c - h_d) W_b}{\lambda_b}. \quad (7)$$

By substitution and arrangement of terms:

$$W_c = \frac{\left(\frac{q_b - 1700 (h_c - h_d)}{\lambda_b} \right)}{\left(1 + \frac{(h_c - h_d)}{\lambda_b} \right)}. \quad (8)$$

The weight of hydrogen vented from twelve secondary storage units during a 168-hour storage period is therefore:

$$\psi_7 = W_c \left(1 - \frac{\rho_a}{\rho_b} \right). \quad (9)$$

- b. ψ_6 The weight of hydrogen vented from twelve secondary units as displacement gas is defined by inspection of Figure F-1.

$$\psi_6 = (\psi_6 + \psi_7 + 1700) \left(\frac{\rho_e}{\rho_f} \right). \quad (10)$$

Since ψ_3 and ψ_4 are losses incurred during transfer and ψ_5 is a loss incurred just before transfer, these terms are not to be included in the above expression for ψ_6 . By rearrangement of terms in Equation (10):

$$\psi_6 = \frac{\frac{\rho_e}{\rho_f} (\psi_7 + 1700)}{1 - \frac{\rho_e}{\rho_f}} \quad (11)$$

- c. ψ_5 Assuming that each secondary unit contains only saturated vapor just before filling, the amount of hydrogen vented during blowdown of twelve secondary units is dependent upon the total contained volume of fluid:

$$V = 1.10 \left(\frac{1700 + \psi_7 + \psi_6}{\rho_f} \right) \quad (12)$$

The above expression includes a 10% allowance for vapor space.

The weight of hydrogen vented during blowdown of twelve secondary units is therefore:

$$\psi_5 = 2 \left(\frac{\text{gm}}{\text{mole}} \right) \left(\frac{\text{lb}}{454 \text{ gm}} \right) \left[\frac{P_a V}{RT_a} - \frac{P_b V}{RT_b} \right] \quad (13)$$

or:

$$\psi_5 = 2.74 V \left(\frac{P_a}{T_a} - \frac{P_b}{T_b} \right) \quad (14)$$

- d. ψ_4 Based on the specific heats of the various materials which make up the transfer piping of the primary unit and one secondary unit, the over-all heat capacity of the proposed piping system when cooling it from 70°F to -423°F is approximately 2625 Btu.

During the initial cooldown of the proposed piping system, the available heat capacity of the hydrogen equals the sum of its latent heat plus the sensible heat gained as its temperature is raised to 70°F, or:

$$190 \frac{\text{Btu}}{\text{hr}} + 320 \frac{\text{Btu}}{\text{lb}} = 510 \frac{\text{Btu}}{\text{lb}} \quad (15)$$

The over-all heat leak into the transfer piping, during transfer, based on a 70°F ambient temperature, is estimated to be 262 Btu/hr. At a delivery rate of 100 gpm, the over-all frictional loss in the piping is equivalent to an approximate heat gain of 2550 Btu/hr. Therefore, the frictional loss plus the heat leak of the piping system is 2812 Btu/hr at a delivery rate of 100 gpm.

One hour after the first transfer the cool-down heat capacity of the primary and one secondary units transfer piping will be approximately 543 Btu. This reduced heat capacity is due to the fact that the piping of the primary unit will still be relatively cold at that time. Therefore, little of the sensible heat of the hydrogen vaporized during cooldown will be absorbed during the 2nd through 12th transfers, which are assumed to occur consecutively at one hour intervals following the first transfer. Therefore, the cooldown loss of the 2nd through 12th transfers can be expressed as:

$$11 \left(\frac{543}{\lambda_c} \right). \quad (16)$$

The total transfer loss for twelve transfers can be expressed as:

$$\psi_4 = \frac{2625 \text{ Btu}}{510 \text{ Btu/lb}} + 11 \left(\frac{543}{\lambda_c} \right) + 12 \left(2812 \frac{\text{Btu}}{\text{hr}} \right) \left(\frac{\omega}{\lambda_c} \right) \quad (17)$$

or:

$$\psi_4 = 5.15 + \left(\frac{5,973 + 33,744 \omega}{\lambda_c} \right), \quad (18)$$

where the total time to complete twelve transfers at 100 gpm is:

$$\omega = \frac{1.86 \times 10^{-6} (\psi_5 + \psi_6 + \psi_7 + 1700)}{\rho_h} \quad (19)$$

- e. ψ_3 By inspection of Figure F-1, the weight of hydrogen vented as flash-off vapor due to the pressure drop across the transfer piping can be expressed as:

$$\psi_3 = \frac{Z (\psi_4 + \psi_5 + \psi_6 + \psi_7 + 1700)}{1 - Z}, \quad (20)$$

where Z is that percentage of hydrogen which is flashed off and is determined by the equilibrium enthalpies of the liquid in both primary and secondary units during transfer.

- f. ψ_2 The volume of liquid contained in the transfer piping of the primary unit after each transfer is approximately 0.435
Therefore:

$$\psi_2 = 12 (0.435) \rho_g = 5.22 \rho_g. \quad (21)$$

- g. ψ_1 The heat available for liquid vaporization in the primary unit during a 168-hour storage period can be represented by the expression:

$$Q_a = q_a - (h_a - h_b) W_a, \quad (22)$$

where q_a is determined by utilizing Figure 24 of the First Quarterly Report⁽⁴⁾.

The total weight of liquid in the primary unit at the beginning of a 168-hour storage period is:

$$W_a = (\psi_2 + \psi_3 + \psi_4 + \psi_5 + \psi_6 + \psi_7 + 1700 + W_d). \quad (23)$$

The weight of liquid vaporized in the primary unit during 168 hours of storage is:

$$W_d = \frac{q_a - (h_a - h_b) W_a}{\lambda_a}. \quad (24)$$

By substitution and arrangement of terms:

$$W_d = \frac{\left[\frac{q_a - (\psi_2 + \psi_3 + \psi_4 + \psi_5 + \psi_6 + \psi_7 + 1700) (h_a - h_b)}{\lambda_a} \right]}{1 + \frac{(h_a - h_b)}{\lambda_a}}. \quad (25)$$

The weight of hydrogen vented from the primary unit during a 168-hour storage period can be represented by the expression:

$$\psi_1 = W_d \left(1 - \frac{\rho_c}{\rho_d} \right). \quad (26)$$

III. RESULTS OF EFFICIENCY ANALYSIS

Table F-1 presents the results of the estimations of fuel losses for each of the identified operating modes. It is seen in the table that stockpiling modes 2 and 5 appear to be essentially equal. However, if the primary storage period is extended beyond one week, the efficiency of mode 5 will decrease more rapidly than that of mode 2. This will be caused by the disproportionate increase in the flash loss of mode 5. Therefore, mode 2 was selected as the most promising for long term storage of fuel.

IV. ESTIMATED HEAT LEAK INTO THE M-113 FUEL SYSTEM WITH A SPLIT TANK CONFIGURATION

As was stated in Section IVC of this report, the tankage geometry assumed for the system efficiency analysis was a single cylindrical vessel ($L/D = 2$) with a single liquid delivery line. The justification for altering the configuration for these analyses is that the more optimum single vessel configuration is considered to be a more reasonable approximation of the final design.

In this section of the appendix the heat leak into the split vessel system (Fig. 2) is estimated. Consider four 60-gallon vessels, each $L/D = 8$. From Figure 24, Reference 4, the heat leak into the vessel is:

$$0.305 \frac{\text{Btu}}{\text{hr lb}} \times 60 \frac{\text{gal}}{\text{vessel}} \times 4 \text{ vessels} \times 0.591 \frac{\text{lb}}{\text{gal}} = 43.2 \frac{\text{Btu}}{\text{hr}}. \quad (27)$$

The total equivalent length of interconnecting piping in the system is 94 ft. This value includes an estimate of the piping length which is equivalent to the four penetrations through the insulation for liquid removal (see Appendix C). The apparent heat leak into this equivalent length of 2-inch piping is

$$0.79 \frac{\text{Btu}}{\text{hr ft}} \times 94 \text{ ft} = 74.2 \text{ Btu/hr}, \quad (28)$$

where the heat leak per unit length of 2" vacuum piping is based on experience.

On this basis, the total heat, $q_b/12$, into one vessel in one week is:

$$\frac{q_b}{12} = (43.2 + 74.2) \frac{\text{Btu}}{\text{hr}} \times 168 \text{ hr} = 19,700 \text{ Btu}. \quad (29)$$

The total quantity of hydrogen vented, ψ_7 , is approximately:

$$\psi_7 = \frac{q_b}{12\lambda_b} \left(1 - \frac{\rho_a}{\rho_b} \right) = \frac{19,700 \text{ Btu}}{195 \text{ Btu/lb}} \left(1 - \frac{0.09}{4.40} \right) = 99 \text{ lb.} \quad (30)$$

The estimated fractional loss of contents from each secondary unit is:

$$\frac{99 \text{ lb}}{240 \text{ gal} \times 0.591 \text{ lb/gal}} \times 100 = 69.8\% \quad (31)$$

It is estimated that the system efficiency for the fuel stockpiling model considered would be about 55% if the split, slender vessel configuration were used. This value can be computed by substituting $12 \times 99 = 1188 \text{ lbs}$ for ψ_7 for mode 2 in Table F-1. This value should be compared with the approximately 80% efficiency predicted for the more optimum vessel geometry assumed in Section IVC. Because the tankage configuration strongly influences the fuel system transfer efficiency, weight, total volume and complexity, more emphasis must be given to the cryogenic fuel system during design of user vehicles in order to increase the attractiveness of the over-all Energy Depot System.

V. NOMENCLATURE

- A - Liquid hydrogen delivered from plant into primary unit (lb)
- B - Hydrogen vented back to plant from primary unit (lb)
- C - Liquid hydrogen contained in primary unit after filling (lb)
- PR - Liquid hydrogen vaporized to effect liquid delivery during 12 secondary transfers (lb)
- h_a - Enthalpy of liquid in primary unit when vent opens during one week of storage (Btu/lb)
- h_b - Enthalpy of liquid in primary unit at beginning of one week of storage (Btu/lb)
- h_c - Enthalpy of liquid in secondary unit when vent opens during one week of storage (Btu/lb)
- h_d - Enthalpy of liquid in secondary unit at beginning of one week of storage (Btu/lb)
- P_a - Pressure in secondary unit before blowdown (atm)
- P_b - Pressure in secondary unit after blowdown (atm)

- q_a - Total heat leak into primary unit during one week of storage (Btu)
- Q_a - Total heat available for liquid vaporization in primary unit during one week of storage (Btu)
- q_b - Total heat leak into twelve secondary units during one week of storage (Btu)
- Q_b - Total heat available for liquid vaporization in 12 secondary units during week of storage (Btu)
- R - Gas constant ($0.00161 \frac{\text{ft}^3 \cdot \text{atm}}{\text{mole} \cdot ^\circ\text{R}}$)
- T_a - Saturated vapor temperature @ P_a ($^\circ\text{R}$)
- T_b - Saturated vapor temperature @ P_b ($^\circ\text{R}$)
- V - Volume capacity of twelve secondary storage vessels (ft^3)
- W_a - Total weight of liquid in primary unit at beginning of one week of storage (lb)
- W_b - Total weight of liquid in twelve secondary units at beginning of one week of storage (lb)
- W_c - Hydrogen vaporized in twelve secondary units during one week of storage (lb)
- W_d - Hydrogen vaporized in primary unit during one week of storage (lb)
- λ_a - Latent heat of liquid in primary unit when vent opens during one week of storage (Btu/lb)
- λ_b - Latent heat of liquid in twelve secondary units when the vent opens during one week of storage (Btu/lb)
- λ_c - Latent heat of liquid in primary unit during transfer from primary to secondary (Btu/lb)
- ρ_a - Density of saturated vapor in secondary unit when vent opens during one week of storage (lb/ft³)
- ρ_b - Density of liquid in secondary unit when vent opens during one week of storage (lb/ft³)
- ρ_c - Density of saturated vapor in primary unit when vent opens during one week of storage (lb/ft³)
- ρ_d - Density of liquid in primary unit when vent opens during one week of storage (lb/ft³)
- ρ_e - Density of vapor in secondary unit during transfer from primary to secondary unit (lb/ft³)
- ρ_f - Density of liquid in secondary unit after transfer from primary to secondary unit (lb/ft³)
- ρ_g - Density of liquid in primary unit after transfer from primary to secondary unit (lb/ft³)
- ρ_h - Density of liquid in primary unit during transfer from primary to secondary unit (lb/ft³)

- ψ_1 - Hydrogen vented from primary unit during one week of storage (lb)
- ψ_2 - Hydrogen vented from primary unit piping and transfer hose after twelve transfers from primary to secondary unit (lb)
- ψ_3 - Hydrogen vented from 12 secondary units due to flashoff during twelve transfers (lb)
- ψ_4 - Hydrogen vented from secondary units due to cooldown, heat leak, and frictional losses in transfer piping when transferring from primary into twelve secondary units (lb)
- ψ_5 - Hydrogen vented from twelve secondary units due to blowdown of same prior to secondary transfer (lb)
- ψ_6 - Hydrogen vented from secondary units as displacement gas during twelve transfers (lb)
- ψ_7 - Hydrogen vented from twelve secondary units during one week storage (lb)
- ω - Total time to transfer from primary unit into twelve secondary units (hr)
- FC - Hydrogen delivered to fuel cells of twelve secondary units, equal to 1700 (lb)

APPENDIX G
PROPERTIES OF HYDROGEN

An analysis of the hazards associated with a particular fuel must take into consideration many factors including volatility, vapor density, rate of mixing with air, flammable mixture range, composition of the lower limit of flammability, minimum spontaneous ignition temperature, ignition energy requirements, burning rate, flame radiation, extinguishment criteria, toxicity, etc. There are no simple criteria that can be used to assess the relative importance of each of these factors; the importance of each is undoubtedly affected by the particular environment. In practice, the importance of a particular factor is often assessed relative to that associated with a common fuel such as gasoline or liquefied petroleum gas. Often, many of the foregoing factors are of little importance in a particular accident situation. For example, spillage of a fuel into an open flame at temperatures above the fire point creates a condition that is not affected by most of these factors. However, spillage of fuel into a diked area adjacent to a hot steam line creates a hazardous condition that depends quite markedly on the relative locations of the flammable mixtures and of the hot lines, the ignition temperature of the fuel, the temperature of the hot line, etc.

For a given set of conditions or a given design a firm set of safety criteria can be established and followed. However, when considering a new design or changing conditions, it is necessary to begin with the fundamental factors and develop safety criteria that truly fit the situation.

To compare the hazards associated with hydrogen relative to those associated with other common fuels, it is convenient to group the pertinent flame and ignition characteristics of each fuel according to the underlying chemical or physical properties that are involved. Representative data are given in Table G-1 below.

TABLE G-1
FLAME AND IGNITION CHARACTERISTICS FOR VARIOUS FUELS

(1) Fuel	(2) Spontaneous Ignition Temperature °F	(3) Laminar Burning Velocity cm/sec	(4) Minimum Ignition Energy Millijoules	(5) Flammable Range in Air, %	(6) Limiting Oxygen Index (a) Volume %
Hydrogen	1075	170	.02	4 --74	5.0
Methane	999	39	.30	5 --14	12.1
Propane	871	44	.30	2.2--9.9	11.4
Benzene	1044	48	--	1.4--7	11.2
i-Octane	784	--	.30	1.1--6.0	--
Methyl Alcohol	--	--	--	7.1--36.5	10.3
Ethylene	--	74	.08	3.1--28.6	10

(a) Minimum oxygen concentration to support flame propagation when stoichiometric fuel-air mixture is diluted with excess nitrogen.

Spontaneous Ignition Temperature (S.I.T.)

Column (2) shows the spontaneous ignition temperature by hot surfaces. It is noted that hydrogen has the highest S.I.T. The significance of this fact is that, in the absence of an open flame, hydrogen must be heated to a relatively high temperature before ignition will occur.

Laminar Burning Velocity

The laminar burning velocity is relevant to the violence of explosions and to the tendency of a flame to "run up" into a detonation. The burning velocity of hydrogen is markedly greater than that of other common fuels, indicating that the hazard of explosions and detonations must be carefully considered. A practical consideration that affects the probability of "run up" is the degree of confinement of the flammable mixture; the greater the degree of confinement, the greater the probability of an explosion.

An indication of actual burning rates of several hydrogen-air and hydrogen-oxygen mixtures is given below in Table G-2.

TABLE G-II

FLAME AND DETONATION VELOCITIES FOR VARIOUS FUEL/OXIDANT MIXTURES

<u>Flammable Mixture</u>	<u>Velocity (Ft/sec)</u>
Hydrogen 43% - Air 57%	8.7
Hydrogen 70% - Oxygen 30%	29.1
Hydrogen 66.7% - Oxygen 33.3%	9270
Methane 10% - Air 90%	1.15
Methane 35% - Oxygen 65%	10.8
Methane 33.3% - Oxygen 66.7%	7040

Minimum Ignition Energy

Spark ignition energies may be used as a relative measure of sensitivity to ignition by localized sources. Hydrogen stands out from other fuels, requiring only about one-tenth the energy for ignition that the others do. This more than offsets the advantage of the higher spontaneous ignition temperature. In practice, this is borne out by the fact that venting or leaking hydrogen ignites very easily. Frictional heating or impact of particles of insulation or metal being carried in a high velocity venting stream can provide sufficient energy for ignition. Vent fires have also been caused by static electrical conditions created by nearby thunderstorms. In spite of the high autoignition temperature, reignition of vented hydrogen by contact with the hot vent stack or other system component must always be considered.

In connection with the low minimum ignition energy, it should be stated that the explosion hazards due to a catastrophic failure of a liquid hydrogen vessel appear to be low. The following information supports this conclusion.

"A limited number of tests were carried out to determine the effects of heavy weights and fragments impacting on dewars containing liquid hydrogen. In each case, the dewar was destroyed, but neither fire nor explosion resulted. In one test, a standard 5-liter dewar was punctured several times by projectiles from a rifle without ignition.

"The interest in tank-rupture tests was generated by the failure of a dewar during pressure testing. The rupture of the vessel was accompanied by ignition of the gas and what was described as an explosion. This latter phenomenon might have been a detonation or simply the blast wave associated with the release of pressure. Therefore, a program was initiated to determine the effects from such a sudden and violent release of liquid hydrogen.

"The initial tests to determine the effects of tank rupture--especially whether or not auto-ignition resulted--were carried out with 5-liter copper dewars in which provision was made for releasing the vacuum to increase the rate of pressure buildup. Rupture occurred at a calculated pressure of 300 psi. Although failure took place with explosive violence, there was no ignition of the liquid hydrogen.

"Further tests were conducted with 5-liter dewars designed to simulate actual storage containers. These test dewars were cylindrical vessels with evacuated perlite insulation. Two inner-shell thicknesses were used, 3/16 in. and 0.035 in. The heavier shell was chosen to provide a stress level, and hence an energy level, approximating that of full-scale storage vessels. However, since the pressure necessary to develop the ultimate stress in the shell was approximately 7,000 psi, it was decided to test also a 0.035-in. shell which would rupture at the failure pressure level of actual storage tanks. In addition to varying the inner-shell thickness, two insulation thicknesses, 1-1/2 in. and 6 in., were used. Each dewar had a valve to release the vacuum and thereby increase the rate of pressure buildup.

"The test dewar was filled with liquid hydrogen and the fill and vent line capped. The vacuum was released and the pressure allowed to increase until the vessel failed. The inner shell of the thin-walled vessels was the first to burst, followed by failure of the outer shell 30-45 sec later. With the heavier-walled vessels, failure of inner and outer shells occurred simultaneously.

"Ignition occurred in two instances; however, the ignition of the escaping gases did not produce a detonation nor did it augment the pressure wave associated with the burst of the vessel. Only a small portion of the available gas was ignited in each instance, producing a fireball not over 3 ft. in diameter. No afterburning of the gas remaining in the shell was observed. The conditions of rupture were somewhat artificial, since the outer-shell vacuum was broken by admitting air in sufficient quantity to produce considerable condensation on the inner-shell wall. Nevertheless, even with this convenient source of oxygen, only two ignitions and no detonations were observed in the nine tests conducted."⁽¹⁴⁾

It should be stated that for the two cases of ignition, out of nine tests, the exact source of ignition is not known. The experiments were to study autoignition effects by the vessel walls during vessel rupture. It is specified that the dewars involved evacuated perlite insulation. Operating experience⁽¹⁵⁾ would indicate that the presence of perlite in a non-inerted vent line increases the probability of a vent fire. It is therefore logical to question whether the two ignitions (above) were due to the rupturing dewar walls or to the perlite.

Flammability Range

Appreciation of the wide flammability range for hydrogen is very important in the safe handling of hydrogen. Hydrogen exhibits the widest range of any of the fuels listed in Table G-1. For hydrogen/oxygen mixtures, the range is from 4% to 94%.

Hydrogen does not have the widest range of all. Hydrazine, for example, has a range of about 5% to 100% and acetylene of 2.5% to 82%. Acetylene actually will decompose explosively so it may be considered a problem from 2.5% to 100%.

Limiting Oxygen Index

The limiting oxygen index is the minimum concentration of oxygen to support flame propagation when a stoichiometric fuel-air mixture is diluted with nitrogen. In other words, it is the least amount of oxygen required to support combustion. In general, hydrocarbon fuels require 10 to 12% oxygen to support combustion. Hydrogen requires only 5% oxygen. This value, of course, sets a limit on the quantity of oxygen that can be tolerated when purging or inerting equipment. This limit is applicable only to the problem of flammability. Other considerations, such as solubility, liquefaction or freezing at the temperature under consideration, may require the limiting of the oxygen to much lower concentrations.

Quenching Distance

Another important factor is the quenching distance. This is the distance between two parallel walls which will just permit a flame to pass without cooling it to extinction. The smaller the quenching distance, the greater

resistance the flame has to cooling, and the more difficult it will be to extinguish. The quenching distance varies depending on the ratio of fuel and oxidant and is compared with propane in Figure G-1. The quenching distance is plotted as a function of the equivalence ratio, which is defined as the actual fuel/air ratio divided by the stoichiometric fuel ratio.

As hydrogen fires are relatively difficult to extinguish, the most reliable approach for extinguishing them is to shut off the source of hydrogen supply. The very high hydrogen volatility and relatively low radiant energy emission from a hydrogen flame make this approach very practical when it can be used.

Other Physical Properties

Table G-III below lists some physical properties of hydrogen in comparison with propane, octane, oxygen and air.

TABLE G-III

PHYSICAL PROPERTIES OF VARIOUS FUELS AND OXIDANTS

Boiling Point, °F	-422.99	-43.7	+258.2	-297.3	-317.9
Liquid Density, lb/ft ³ @ B.P.	4.43	36.2	44	71.2	54.57
Latent Heat,	398	8076	16,620	2932	2556
Gas Density, lb/ft ³ at 70°F, 1 ATM	0.00522	0.1154	-----	0.0828	0.07493
Liquid to Gas Expansion Ratio, B.P. to 70°F	849	313	-----	860	729
Diffusion Coefficients (into air) gm/cm ² /sec	0.68	-----	-----	0.17	-----

Boiling Point

Except for helium, hydrogen has the lowest boiling point of any liquid. Liquid hydrogen will vaporize very quickly when spilled or otherwise exposed to ambient conditions as heat will readily flow across the high temperature gradient to the liquid.

Being a cryogenic liquid, liquid hydrogen presents a freezing hazard to human tissue. Careful selection of materials is also required of all cold components as some materials lose their impact strength and cannot be used at very low temperatures.

Liquid Density

If the same volume were spilled onto the same surface area, almost nine pounds of propane, or ten pounds of octane would cover the same space as one pound of hydrogen.

Figure G-1. FLAME QUENCHING DISTANCE FOR HYDROGEN AND PROPANE

Latent Heat

On a pound mole basis, the latent heat of propane is more than 20 times that of hydrogen. Looking at this another way, when we consider the relatively great temperature differential between liquid hydrogen and ambient, and the low latent heat value, it becomes apparent that when spilled, liquid hydrogen will vaporize very quickly. These factors, of course, intensify the mechanical problems of maintaining liquid hydrogen as a liquid, but are of great value in case of an accidental spill of liquid hydrogen.

Gas Density

Hydrogen gas at -420°F is about as dense as ambient air, but thins rapidly to 6% of that value when warmed to ambient conditions. Cold oxygen will cling to the ground and will fill ditches and low points. This is also a trait of most hydrocarbon gases, methane and acetylene being exceptions. Gas densities greater than air have been an important factor in a number of accidents involving oxygen as well as hydrocarbon gases. However, when hydrogen is warmed slightly above -420°F , it becomes lighter than air and will rise and diffuse readily into the atmosphere. The very low molecular weight contributes strongly to the high diffusion rate.

Liquid-to-Gas Expansion

The liquid-to-gas density ratio for hydrogen and oxygen is approximately 850. The problem with hydrogen is no different than with other cryogenic liquids.

APPENDIX H

INTERIM LIQUID HYDROGEN STORAGE AND DISTRIBUTION SYSTEM:
PROCESS EQUIPMENT DESCRIPTION

I. PRIMARY STORAGE UNIT

The preliminary process flow diagram for the interim fuel system is presented as Figure H-1. The primary unit will be a conventional 5,000-gallon liquid hydrogen trailer. The equipment types and functions will generally be the same as previously described in Appendix C for the conceptual design. The following major items will comprise the interim primary unit.

Vacuum Jacketed Storage Vessel - The liquid hydrogen is to be contained in a single inner vessel with a net capacity of about 5,000 gallons. The vessel will be of welded construction and fabricated from 18-8 stainless steel. Operating pressure will be approximately 32 psia. The vessel will be wrapped with a multilayer reflective-type insulation and mounted in an evacuated jacket. The jacket will be of welded construction and fabricated from carbon steel. Closures for both the inner vessel and the jacket will be formed heads.

Pressure Raising Circuit - The pressure raising circuit will be as described in Appendix C for the conceptual primary unit.

Transfer Line - The transfer line will consist basically of a lightweight, vacuum-jacketed pipe, a control valve, a by-pass valve, and a flexible vacuum-insulated metal hose. The hose is to be equipped with bayonet-type couplings on either end to minimize heat leak during transfer and filling operations. A nipple with a quick-disconnect, double-check coupling will be provided on the hose for purging the transfer line. The control valve will be pneumatically activated, and will operate on a signal from a temperature sensor (to open) and a liquid level sensor (to close) located in the secondary vessel. By opening the by-pass valve slightly, the operator can obtain a low flow rate for cooling down the transfer line prior to filling of the secondary vessel. The by-pass valve is to be used in the event of power failure or malfunction of the automatic control valve.

Vent and Safety System - This system will be as described in Appendix C with the following addition: the vent line will be tapped to supply gaseous hydrogen for purging the transfer line.

Compressed Nitrogen System - This system will be operated in the same way as the compressed air system described in Appendix C for the primary unit. The line downstream of the pressure regulator will be tapped to provide a supply of nitrogen gas for purging the transfer line. The nitrogen will be supplied from a cylinder which will contain enough gas to perform approximately 14 secondary transfer operations. Because nitrogen gas will also be expended during primary transfer, the cylinder will be recharged more frequently than is indicated above. The assumption is made that the nitrogen system is leak-free.

VALVE & INSTRUMENT LIST - PRIMARY VESSEL

C-101	CPLG. - TRANSFER HOSE TO SECONDARY & PLANT	R-101	RUPTURE DISC (INNER VESSEL)
C-102	CPLG. - GAS PHASE FROM SECONDARY	R-102	RUPTURE DISC (JACKET)
C-103	CPLG. - N ₂ TO CONTROL VALVES		
C-104	CPLG. - TRANSFER HOSE TO PRIMARY	STC-101	VACUUM SENSING CONNECTION
C-105	CPLG. - PURGE CONN.		
C-105A	CPLG. - N ₂ GAS PURGE	SV-101	SAFETY VALVE (INNER VESSEL)
C-105B	CPLG. - H ₂ GAS PURGE	SV-102	SAFETY VALVE (P.R. COIL)
		V-101	SHUT-OFF VALVE (STC-101)
CV-101	CHECK VALVE	V-102	SHUT-OFF VALVE (VACUUM PUMP)
		V-103	VENT VALVE
EC-101	ELECTRICAL CONNECTION (POWER SUPPLY)	V-104	SHUT-OFF VALVE (PI-101)
EC-102	ELECTRICAL CONNECTION (SECONDARY & PLANT)	V-105	SHUT-OFF VALVE (PI-102)
LI-101	LEVEL INDICATOR (PRIMARY VESSEL)	V-106	LH ₂ TRANSFER VALVE
LI-102	LEVEL INDICATOR (SECONDARY VESSEL)	V-107	N ₂ VALVE (TO OPEN V-106)
LLS	LIQUID LEVEL SENSORS	V-108	BY-PASS VALVE (V-106)
PCV-101	PRESSURE REGULATOR	V-109	CONTROL VALVE (GAS TO VAPOR SPACE)
PI-101	PRESSURE INDICATOR (INNER VESSEL)	V-110	N ₂ VALVE (TO OPEN V-109)
PI-102	PRESSURE INDICATOR (N ₂ TO CONTROL VALVES)	V-111	BY-PASS VALVE (V-109)
PI-103	DIFFERENTIAL PRESSURE INDICATOR	V-112	BLOCK VALVE
		V-113	BLOCK VALVE (LH ₂ TO P.R. COIL)

VALVE & INSTRUMENT LIST - SECONDARY VESSEL

C-201	CPLG. - TRANSFER HOSE TO SECONDARY	STC-201	VACUUM SENSING CONNECTION
C-202	CPLG. - GAS PHASE FROM INNER VESSEL		
C-203	CPLG. - N ₂ TO CONTROL VALVES	SV-201	SAFETY VALVE (INNER VESSEL)
C-204	CPLG. - GAS TO FUEL CELL	SV-202	SAFETY VALVE (COOL-DOWN GAS BY-PASS LINE)
CV-201	CHECK VALVE		
EC-201	ELECTRICAL CONNECTION (POWER SUPPLY)	TS-201	TEMPERATURE SENSOR
EC-202	ELECTRICAL CONNECTION (TO PRIMARY)		
		V-201	SHUT-OFF VALVE (STC-201)
		V-202	SHUT-OFF VALVE (VACUUM PUMP)
		V-203	VENT VALVE (INNER VESSEL)
		V-204	SHUT-OFF VALVE (PI-201)
		V-205	SHUT-OFF VALVE (PI-202)
		V-206	LH ₂ TRANSFER VALVE
		V-207	N ₂ VALVE (TO OPEN V-206)
		V-208	VENT VALVE (COOL-DOWN GAS BY-PASS LINE)
		V-209	CONTROL VALVE (GAS TO FUEL CELL)
		V-210	N ₂ VALVE (TO OPEN V-209)
		V-211	SHUT-OFF VALVE (PI-203)
		V-212	CONTROL VALVE (LH ₂ TO HEATER)
		V-213	N ₂ VALVE (TO OPEN V-212)
R-201	RUPTURE DISC - (INNER VESSEL)		
R-202	RUPTURE DISC (JACKET)		

REVISIONS						
NO	LETTER	ZONE	DESCRIPTION	DATE	MADE	CHK'D
1	A		MISCELLANEOUS PIPING CHANGES	6-9-64	JTH	WJ

FIGURE H-1

REQ'D	PART NO.	NOMENCLATURE	MAT'L SPEC.	UNIT	FINISH	ZONE
UNLESS OTHERWISE SPECIFIED DIM'S ARE IN INCHES	WILKINS	2-21-64				
FRACTION DECIMAL ANGLES	DRAFTSMAN	3/5/64				
± ± ±	CHECKED	3/5/64				
FIRST USED ON	ENGINEER	3/5/64				
	APP'D	3-5-64				
TITLE			AIR PRODUCTS AND CHEMICALS, INC.			
FLOW DIAGRAM OF INTERIM SYSTEM - LH ₂ STORAGE AND DISTRIBUTION			ALLENTOWN, PENNSYLVANIA U. S. A.			
SCALE			WT.			
67587			D			

Instrumentation and System Controller - These systems will be basically the same as described in Appendix C. A small gas analyzer will be installed in the vent line and additional signal lamps will be required on the console to indicate the presence of hydrogen and nitrogen gas at the outlet of the vent stack on the secondary unit. Its purpose will be to inform the operator when the purging of the transfer lines is complete.

II. SECONDARY STORAGE UNIT

Figure H-1 contains the preliminary process flow diagram for the interim secondary fuel system. The experimental vehicle to be adapted for hydrogen fuel has not been specified. The following is a description of the major items of equipment which will comprise the interim secondary fuel system. The components will be generally similar in type and function to the equipment already described for the conceptual secondary unit.

Vacuum Jacketed Storage Vessel - The liquid hydrogen will be contained in a single vessel of essentially the same general construction as previously specified for the interim primary vessel.

Vaporizing Equipment - The operation and control of the vaporizing equipment will be as described in Appendix C. Since the system considered will consist of only one inner vessel, only one control valve will be required to regulate the flow of liquid hydrogen to the heater.

Transfer and Fuel Cell Supply Lines - These lines will be as described in Appendix C for the conceptual secondary unit.

Vent and Safety System - This system will be as described in Appendix C with the following addition:

- Hydrogen and nitrogen detector to be located in vent stack to indicate completion of hydrogen and nitrogen purges.

Compressed Nitrogen System - This nitrogen system will be essentially the same as the compressed air system described for the conceptual secondary unit in Appendix C except that the solenoid valves to operate V-212 and V-214 (Fig. 8) on the multivessel system will not be required. The compressed nitrogen cylinder will contain enough gas to perform approximately 80 transfer operations. Because compressed nitrogen gas is required during vehicle operation, the cylinder will be recharged more often than is indicated above. A leak-tight system is assumed.

Instrumentation and System Controller - These systems will be as described in Appendix C with the addition of signal lamps to indicate completion of purging operations.

APPENDIX J

INTERIM LIQUID HYDROGEN STORAGE AND DISTRIBUTION SYSTEM:

GENERAL OPERATING PROCEDURE

Tables J-I and J-II define nine operating conditions and the state of the equipment for each condition for the primary and secondary unit, respectively. The component designations are those identified in Figure H-1.

Although many of the basic automatic operations which have been designed into the conceptual fuel system will be available in the interim system also, certain duties must be performed manually during normal operations. The following is a description of the system operation in terms of the manipulations to be performed by the operator. The conditions refer to those defined in Tables J-I and J-II.

Condition (1) - To prepare the primary vessel for normal filling, the operator will connect a small electric cable (EC-102) and the transfer hose (C-101), in that order, to the delivery manifold of the hydrogen plant and open the vent valve V-103. The filling operation and shut-off will be controlled manually by the operator. When the primary is full, the flow will be stopped at the plant, C-101 will be disconnected, V-103 will be closed, and EC-102 will be disconnected, in that order. The system controller will then be de-energized.

Condition (2) - After the primary unit has been placed in the storage area, the block valve (V-112) should be opened. This will keep the pressure in the inner vessel in the range of 16 psia, thus minimizing losses.

Condition (3) - When the primary unit is taken from long-term storage, valve V-112 should be closed prior to leaving the storage area. If a transfer operation is to be performed shortly after the primary unit is filled, no manual corrections will need to be made to the system, since valve V-112 will already be closed.

Condition (4) - Before liquid hydrogen is introduced to the transfer line, a purging operation must be performed to expel air from the transfer circuit. After the primary and secondary units have arrived at the transfer site, the operator will connect the electrical cable (EC-102), the gas pressure hose (C-102), and the transfer hose (C-104 and C-101), in that order. He then will select the desired transfer rate and will energize the system controller by operating the appropriate switches. The operator will then initiate the purging operation by connecting C-105A to C-105 on the primary unit. This will introduce warm nitrogen gas to the transfer line to displace the air in the circuit through the warm cool-down gas by-pass line to the atmosphere. When a signal at the control console indicates that the air has been expelled, the operator will disconnect C-105A and will connect C-105B to C-105. This will permit hydrogen gas to flow from the primary system vent line to the transfer line, thus displacing the nitrogen gas from the transfer circuit. When a signal at the control console indicates that the nitrogen has been expelled, C-105B will be disconnected. The transfer circuit will then be ready for cool-down prior to fuel transfer.

TABLE J-I

STATE OF EQUIPMENT DURING OPERATION - INTERIM PRIMARY SYSTEM

- Condition (1) - Filling of primary unit
 Condition (2) - Long term liquid storage in primary unit
 Condition (3) - Primary unit en route to rendezvous with secondary unit
 Condition (4) - Transfer line purge prior to filling of secondary unit
 Condition (5) - Primary unit cooling down transfer lines when coupled to secondary unit
 Condition (6) - Transfer from primary to secondary unit
 Condition (7) - Transfer line purge after filling of secondary unit
 Condition (8) - Secondary unit filled, or partially filled, and not in use
 Condition (9) - Vapor generation in secondary unit when in use

COMPONENT	CONDITION								
	(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)
C-101	Conn.	Disc.	Disc.	Conn.	Conn.	Conn.	Conn.	N.A.	N.A.
C-102	Disc.	Disc.	Disc.	Conn.	Conn.	Conn.	Conn.	N.A.	N.A.
C-103	Conn.	Conn.	Conn.	Conn.	Conn.	Conn.	Conn.	N.A.	N.A.
C-104	Conn.	Disc.	Disc.	Conn.	Conn.	Conn.	Conn.	N.A.	N.A.
C-105	Disc.	Disc.	Disc.	Conn.	Disc.	Disc.	Conn.	N.A.	N.A.
C-105A	Disc.	Disc.	Disc.	Conn. Disc.	Disc.	Disc.	Disc. Conn.	N.A.	N.A.
C-105B	Disc.	Disc.	Disc.	Disc. Conn.	Disc.	Disc.	Conn. Disc.	N.A.	N.A.
EC-101	Conn.	Conn.	Conn.	Conn.	Conn.	Conn.	Conn.	N.A.	N.A.
EC-102	Conn.	Disc.	Disc.	Conn.	Conn.	Conn.	Conn.	N.A.	N.A.
Sys. Cont.	E.	De.	De.	E.	E.	E.	E.	N.A.	N.A.
V-101	C.	C.	C.	C.	C.	C.	C.	N.A.	N.A.
V-102	C.	C.	C.	C.	C.	C.	C.	N.A.	N.A.
V-103	E.O.	D.C.	D.C.	D.C.	D.C.	D.C.	D.C.	N.A.	N.A.
V-104	O.	O.	O.	O.	O.	O.	O.	N.A.	N.A.
V-105	O.	O.	O.	O.	O.	O.	O.	N.A.	N.A.
V-106	O.	C.	C.	C.	C.	O.	C.	N.A.	N.A.
V-107	E.C.A.	D.O.A.	D.O.A.	D.O.A.	D.O.A.	E.C.A.	D.O.A.	N.A.	N.A.
V-108	C.	C.	C.	C.	C.	C.	C.	N.A.	N.A.
V-109	C.	C.	C.	O.	O.	O.	O.	N.A.	N.A.
V-110	D.O.A.	D.O.A.	D.O.A.	E.C.A.	E.C.A.	E.C.A.	E.C.A.	N.A.	N.A.
V-111	C.	C.	C.	C.	C.	C.	C.	N.A.	N.A.
V-112	C.	O.	C.	C.	C.	C.	C.	N.A.	N.A.
V-113	O.	O.	O.	O.	O.	O.	O.	N.A.	N.A.

Abbreviations:

O. - Open	Conn. - Connected	D.O.A. - De-energized and open to atmosphere
C. - Closed	Disc. - Disconnected	E.C.A. - Energized and closed to atmosphere
E. - Energized	E.O. - Energized and open	N.A. - Not Applicable
De. - De-energized	D.C. - De-energized and closed	

TABLE J-II

STATE OF EQUIPMENT DURING OPERATION - INTERIM SECONDARY UNIT

- Condition (1) - Filling of primary unit
 Condition (2) - Long term liquid storage in primary unit
 Condition (3) - Primary unit en route to rendezvous with secondary unit
 Condition (4) - Transfer line purge prior to filling of secondary unit
 Condition (5) - Primary unit cooling down transfer lines when coupled to secondary unit
 Condition (6) - Transfer from primary to secondary unit
 Condition (7) - Transfer line purge after filling of secondary unit
 Condition (8) - Secondary unit filled, or partially filled, and not in use
 Condition (9) - Vapor generation in secondary unit when in use

COMPONENT	CONDITIONS								
	(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)
C-201	N.A.	N.A.	N.A.	Conn.	Conn.	Conn.	Conn.	Disc.	Disc.
C-202	N.A.	N.A.	N.A.	Conn.	Conn.	Conn.	Conn.	Disc.	Disc.
C-203	N.A.	N.A.	N.A.	Conn.	Conn.	Conn.	Conn.	Conn.	Conn.
C-204	N.A.	N.A.	N.A.	Conn.	Conn.	Conn.	Conn.	Conn.	Conn.
EC-201	N.A.	N.A.	N.A.	Conn.	Conn.	Conn.	Conn.	Conn.	Conn.
EC-202	N.A.	N.A.	N.A.	Conn.	Conn.	Conn.	Conn.	Disc.	Disc.
Heater	N.A.	N.A.	N.A.	De.	De.	De.	De.	De.	E.
Sys. Cont.	N.A.	N.A.	N.A.	De.	De.	De.	De.	De.	E.
V-201	N.A.	N.A.	N.A.	C.	C.	C.	C.	C.	C.
V-202	N.A.	N.A.	N.A.	C.	C.	C.	C.	C.	C.
V-203	N.A.	N.A.	N.A.	D.C.	D.C.	E.O.	D.C.	D.C.	D.C.
V-204	N.A.	N.A.	N.A.	O.	O.	O.	O.	O.	O.
V-205	N.A.	N.A.	N.A.	O.	O.	O.	O.	O.	O.
V-206	N.A.	N.A.	N.A.	C.	C.	O.	O.C.	C.	C.
V-207	N.A.	N.A.	N.A.	D.O.A.	D.O.A.	E.C.A.	E.C.A. D.O.A.*	D.O.A.	D.O.A.
V-208	N.A.	N.A.	N.A.	E.O.	E.O.	D.C.	D.C. E.O.	D.C.	D.C.
V-209	N.A.	N.A.	N.A.	C.	C.	C.	C.	C.	O.
V-210	N.A.	N.A.	N.A.	D.O.A.	D.O.A.	D.O.A.	D.O.A.	D.O.A.	E.C.A.
V-211	N.A.	N.A.	N.A.	O.	O.	O.	O.	O.	O.
V-212	N.A.	N.A.	N.A.	C.	C.	C.	C.	C.	O.
V-213	N.A.	N.A.	N.A.	D.O.A.	D.O.A.	D.O.A.	D.O.A.	D.O.A.	E.C.A.

Abbreviations

O.	-	Open	Disc.	-	Disconnected
C.	-	Closed	E.O.	-	Energized and Open
E.	-	Energized	D.C.	-	Closed and De-energized
De.	-	De-energized	D.O.A.	-	De-energized and open to atmosphere
Conn.	-	Connected	E.C.A.	-	Energized and closed to atmosphere
			N.A.	-	Not Applicable

*Note: These two states allow hydrogen to be drawn from either of the two vessels.

Conditions (5,6) - When the differential pressure between the units reaches the present value, the operator will partially open valve V-108 to initiate the cool-down operation. The transfer and shut-off will then be performed automatically. When the transfer operation starts, V-108 should be closed.

Condition (7) - After the transfer operation has been completed, the lines must be purged. The operator will open V-206 by flicking a switch on the primary and will connect C-105B to C-105. The hydrogen gas entering the line will blow the liquid from the line into the secondary storage vessel. When the liquid has been removed from the line, as indicated by a pressure rise in the secondary inner vessel, V-206 will be closed, and C-105B will be disconnected. V-208 will then be opened by flicking a switch and C-105A is to be connected to C-105. The nitrogen gas entering the line will displace the hydrogen gas by blowing it through the warm cool-down gas by-pass line to the atmosphere. When a signal at the control console indicates that the hydrogen gas has been expelled, V-208 will be closed and C-105A, C-101, C-102, C-104, and EC-102 are to be disconnected in that order.

Condition (8) - When the operator parks the secondary unit, he will merely turn off the drive unit (similar to turning off an automobile engine). No manual corrections need to be made to the system.

Condition (9) - To increase the flow of hydrogen gas to the fuel cell, the operator need only depress the accelerator lever of the drive unit.

APPENDIX K

INNER SHELL STUDY FOR THE
50-GALLON LIGHTWEIGHT VESSEL

I. INTRODUCTION AND SUMMARY

It is the purpose of this appendix to summarize the results of the inner shell study and to present information generated but not previously published under this contract.

In the First Quarterly Report⁽⁴⁾ three materials were recommended as promising candidates for lightweight inner vessels for liquid hydrogen storage for the Energy Depot. These were aluminum alloys 2219-T87 and 7039-T6, and 18% nickel maraging steel. Since publication of that report, aluminum alloy X7006-T6 has been added to the list. The inner shell study was initiated to establish the following necessary but otherwise unavailable information.

- . Weldability - by conventional commercial processes suitable for production, if possible;
- . Weldment Characteristics - Notched and unnotched tensile strengths and Charpy-V notch impact resistance determined at room and cryogenic temperatures.

The results of this study are summarized as follows:

- . Aluminum alloy 2219 appears to be the most promising inner shell material for lightweight Energy Depot liquid hydrogen vessels. With this material it is possible to reproducibly achieve the highest strength weldments which are not brittle at liquid hydrogen temperature. The use of this alloy is recommended for future vessel studies for Energy Depot.
- . Aluminum alloy 7039 is not recommended for use at this time for the present application. As is documented in the last section of this appendix, considerable difficulty was experienced in obtaining x-ray-quality weldments in thin sections on any consistent basis. The formulations of the alloy and the recommended filler metal must be developed further, as must techniques for reducing heat input to the weldment to reduce the crack sensitivity of the deposited weld metal for thin-walled vessels.
- . Aluminum alloy X7006 exhibited the lowest weld joint strength, a large scatter in tensile strength data, and the lowest percentage of weld elongation. The alloy has recently been removed from the commercial market.
- . 18% nickel maraging steel is not recommended for this use. Low impact test results and lack of ductility indicated by the absence of lateral contraction at the notch of welded specimens indicate this material to be quite notch sensitive at -423°F. Cryogenic tensile tests were not performed as the samples exhibited delayed cracking effects.

These conclusions are substantiated by information presented in the following sections of this appendix.

II. CRYOGENIC NOTCHED TENSILE TESTS

Samples of weldments of the four candidate materials were sent to General Dynamics/Astronautics, California for notched and unnotched tensile testing at -320°F and -423°F . The following summarizes the results of these tests.

2219 T87 W/2319 Filler, MIG Process, Post Weld Aging Treatment

Under normal production processing, welded specimens of 2219 alloy exhibit both high strength and low notch sensitivity at temperatures as low as -423°F . Because of the low notch sensitivity advantage can be taken of the high strength property at low temperatures. This alloy therefore appears most promising to application to lightweight liquid hydrogen vessels.

X7006-T6 W/5180 Filler, MIG Process, Post Weld Aging Treatment

Although unnotched specimens failed prematurely as a result of fusion zone defects, notched strengths were high at both -320°F and -423°F and the indication is that fairly high notched/unnotched ratios would be obtained under more closely controlled conditions, in spite of very low elongation results.

7039 T6 W/5039 Filler, MIG Process, Post Weld Aging Treatment

Consistently low notched ratios, obtained in this group of tests, compared with other more optimistic results from other investigations, indicate that the 7039 material may be quite sensitive, in its cryogenic properties, to specific welding processes or, more accurately, to heat input during welding. The excessively large grain structure and weld zone intergranular failures observed tend to substantiate this possibility.

18% Nickel Maraging Steel, Metal Arc Process, Post Weld Aging Treatment

Machined and aged test specimens were not tested because visual examination prior to test indicated the existence of small cracks in the weld zone.

The following is the complete report submitted by General Dynamics/Astronautics to Air Products and Chemicals, Inc. in fulfillment of the sub-contract to perform these tests(17).

Test Report of
General Dynamics/Astronautics, Inc.

TO: Distribution

FROM: MATERIALS RESEARCH GROUP, Dept. 592-1

SUBJECT: Cryogenic Tensile Tests Conducted on Aluminum Welds for
Air Products and Chemicals, Inc., Contract Number PO-03-06260

ABSTRACT: The mechanical properties of inert gas metal-arc (MIG) weld
in 7006, 7039, and 2219 aluminum alloys were determined at
liquid nitrogen (-320°F) and liquid hydrogen (-423°F) test
temperature. Tests were performed on smooth and notched
tensile specimen. Data obtained included ultimate tensile
strength of smooth and notched specimens as well as load-
extension curves and percent elongation for the unnotched
specimens.

Prepared by: R. W. Tryon
Senior Engineering Metallurgist

APPROVED BY: A. Hurlich
Chief of Materials and Structures Research

Test Report of
General Dynamics/Astronautics, Inc.

TEST PROGRAM

The purpose of this contract was to determine the mechanical properties on smooth and notched aluminum alloy and nickel maraging steel specimens at liquid nitrogen (-320°F) and liquid hydrogen (-423°F) test temperatures. Air Products and Chemicals, Inc. supplied 32 "finished" machined tensile specimens consisting of eight welded specimens of each of the following alloys - 7006, 7039, 2219, and 18% nickel maraging steel. The word "finished" is in quotes since General Dynamics/Astronautics agreed to machine the notch as well as modify the specimens to fit existing cryostats or cryogenic test fixtures. The contract required the Materials Research Group to conduct tensile tests at -320°F and -423°F on two smooth and two notched specimens for each material, determine the ultimate strength of both types of specimens, and obtain percent elongations and load-extension curves on all smooth specimens. In addition, General Dynamics/Astronautics contracted to design and machine all notched specimens to provide a stress concentration factor (K_t) equal to approximately 17 and calculated from the equation $K_t = \frac{a}{r}$,

where a equals $1/2$ the width between notches, and r the radii of the notches. It was understood by APCI that GD/A would not guarantee notch radii of less than 0.001 inches. It was also agreed that GD/A would visually and dimensionally inspect all test specimens supplied by APCI.

PROCEDURES

Visual Examination: Weld cracks were visually observed in two maraging steel specimens, subsequently, all maraging steel specimens were inspected by the fluorescent penetrant method to detect the presence of and determine the extent of all defects. Penetrant inspection revealed the presence of microcracks in all eight specimens, apparently the result of machining and/or welding operations performed by APCI. GD/A notified APCI immediately that the maraging steel specimens were unsuitable for testing because the presence of such microcracks would undoubtedly lead to premature failure and, thus, erroneous test data. APCI requested that all maraging steel specimens be returned untested at the completion of the test program. Additional machining on the remaining 24 aluminum test specimens was required to accommodate existing cryogenic test fixtures. Modifications of both the smooth and notched specimens included a reduction in length of the test specimen from 12 inches to 9 inches and two $1/2$ inch holes drilled approximately $3 \frac{5}{8}$ inches from the center-line of the weld cross section (see attached sketch).

All welds were machined flush with the base material by APCI and, therefore, each specimen was micro-etched in a modified Tucker's reagent to define the weld and determine its exact location. The micro-etch served two purposes, (1) a positive means of locating the centerline of the cross section of the weld and (2), assurance that the entire root of each machined notch would be located in deposited weld metal. It should be noted that the centerline of

Test Report of
General Dynamics/Astronautics, Inc.

face or root welds does not necessarily correspond to the centerline of the cross section of the same weld, particularly, if the welding process used is inert gas metal-arc, (MIG).

Dimensional Inspection: Areas of all specimens machined by APCI were determined and approximate ultimate tensile loads were estimated. The load requirements were found to be well within the capabilities of the 30,000 pound tensile machines.

The dimension considered most important to the overall design of the notch was the width of the specimen at the center of the weld. It is at this point on the specimen that edge notches, designed with a 60° included angle and root radii of 0.0010", are introduced by machining to a standard depth of at least 30 percent but not greater than 40 percent of this measured width. For the 7006 and 2219 specimens, this width was found to vary between 0.624" and 0.695" while the width of the 7039 specimens did not exceed $0.934 \pm .002$ ". The attached sketch of APCI's notched tensile specimen shows the width between notches (2a) which GD/A had to machine for each alloy in order to obtain maximum stress concentration factors. Dimensions pertinent to the notched tensile strength of the deposited filler metal for all notched specimens are shown in tabular form in Table I. Note that a K_t of 18 was attainable only for 7039 aluminum alloy weldments while the K_t value for all other notched specimens was approximately 14.5.

Mechanical Testing: Test apparatus consisted of universal testing machines equipped with continuous stress-strain recorders and strain pacers. Specially designed cryoextensometers were used for the low temperature tests. All tensile tests were performed in liquid nitrogen (-320°F) and liquid hydrogen (-423°F) cryostats mounted on the universal test machines. Tests were conducted after each immersed specimen had reached equilibrium temperature (approximately 5 minutes) as determined by copper constantan thermocouples. The smooth tensile specimens were tested at a strain rate of 0.005 inches/inch/minute until yield, then followed by 0.15 inches/minute head speed until failure. Notched specimens were tested at 0.15 inches/minute head speed until failure. Yield strengths were determined from the stress-strain curves by the 0.2% off-set method. Elongations were reported as total elongations over a one inch gauge length.

Results and Discussion: The mechanical properties of APCI's aluminum MIG weldments at -320°F and -423°F test temperatures are shown in Table II. The results show that in all cases the ultimate tensile strengths, yield strengths, and notched tensile strength increased as the temperature is decreased. Percent elongation, as measured over a one inch gauge length, remained fairly constant or decreased slightly as the test temperature was lowered from -320°F to -423°F. The following general observations are also made with respect to the weld joint properties of each alloy.

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2219 Alloy: Highest weld joint strengths were obtained in these weldments at both test temperatures, i.e., 60.8 ksi at -320°F and 68.5 ksi at -423°F . No significant change was noted in the ductility of these welds as measured by percent elongation. In all cases, specimens failed in cleavage through the weld, and thus, a valid comparison can be made with respect to the cryogenic notched-unnotched tensile ratios of the deposited filler metal. These ratios were 0.80 and 0.71 for -320°F and -423°F , respectively, and is a qualitative indication that the deposited filler metal in the weldments becomes more notch sensitive as temperature is reduced.

7006 Alloy: These weldments exhibited the lowest joint strength, a large scatter in test data, and lowest percentage of weld elongation. The lack of fusion was identified on the fractured surfaces of two smooth tensile specimens and this defect is believed largely responsible for the unexpected low test values. It is interesting to note, however, that the notched tensile strength of the deposited filler metal in the 7006 alloy is higher than 2219 alloy weldments tested under very similar conditions. Only one smooth tensile specimen in this group (Specimen #2) failed in the weld and an analysis, therefore, of the relative sensitivity or insensitivity of the deposited filler metal to notches cannot be made from the available test data. The notched-unnotched tensile strength ratio at -423°F of Specimen #2, i.e. 55.0 ksi/54.8 ksi, is equal to unity and suggests that the deposited filler metal in the 7006 weldments is relatively tough and insensitive to the presence of stress raisers ($K_t = 14.5$) at liquid hydrogen temperature. It is highly recommended that additional testing be conducted on these weldments (or the new 7106 alloy with a zirconium addition) to substantiate the insensitivity of the deposited filler metal (5080) to sharp notches.

7039 Alloy: These weldments exhibited the greatest amount of ductility as measured by percent elongation in a one inch gauge length. Ultimate tensile strengths of the smooth specimens were somewhat inconsistent and varied as much as plus or minus 6 percent of the average test value. This variation, even on a limited number of test specimens, is considered too large for automatic welds. An examination of the fractured surfaces of both smooth and notched tensile specimens revealed an excessively large equiaxed grain structure and intergranular type failures in the weld metal. The type of failure and grain structure found on all specimens is believed responsible (in part) for the large variation in ultimate tensile strength of the welds as well as the resultant low notched tensile strength.

The Materials Research Group at GD/A recently completed an extensive program on the weldability studies of several high strength aluminum alloys for advanced space boosters. Table III represents the typical mechanical properties obtained for 7006, 7039, and 2219 aluminum alloy plate and their weldments. These test data are included in this report for comparative purposes and to provide additional information in the evaluation of the alloys now under investigation at APCI.

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Table I
Dimensions of Notch Specimens with
Resulting Stress Concentration Factors

<u>Specimen Number</u> ⁽¹⁾	<u>Thickness</u> <u>Inches</u> ⁽¹⁾	<u>Width</u> <u>Inches</u> ⁽¹⁾	<u>a,</u> <u>Inches</u> ⁽²⁾	<u>K_t</u> ⁽³⁾
7039 - 1	0.1051	0.9324	0.3286	18.1
4	0.1049	0.9325	0.3235	18.0
5	0.1068	0.9323	0.3280	18.1
7	0.1082	0.9327	0.3275	18.1
5425 - 2	0.4638	0.6487	0.2142	14.6
5	0.4588	0.6291	0.2098	14.5
6	0.4480	0.6861	0.2150	14.7
8	0.4555	0.6826	0.2121	14.6
7006 - 3	0.4251	0.6260	0.2144	14.6
4	0.4374	0.6241	0.2067	14.4
5	0.4355	0.6253	0.2133	14.6
6	0.4425	0.6237	0.2093	14.5

(1) Specimens identified and machined by APCI

(2) Machined by GD/A. All notch radii 0.0010 with notch angles equal to 59° 58'.

(3) $K_t = \frac{a}{r}$ where a = 1/2 of the width between the notches and r the radii of the notches.

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Table II
Mechanical Properties of Aluminum MIG
Weldments⁽¹⁾ at Cryogenic Temperatures (APCI)

Alloy	Temp. °F	Specimen No.	F _{ty} ksi	F _{tu} ksi	%e in 1"	N.T.S. ksi ⁽²⁾	Fracture Location ⁽⁵⁾
2219	-320	1	37.8	60.9	6.0		WM
		4	39.2	60.7	6.0		WM
		2				48.4(14.6)	
		5				47.2(14.5)	
		AVE	38.5	60.8	6.0	47.8	
	-423	3	43.7	67.3	5.0		WM
		7	42.5	69.7	6.0		WM
		6				47.3(14.7)	
		8				49.2(14.6)	
		AVE	43.1	68.5	5.5	48.3	
	-320	1	32.6	35.5	2.0		FZ
		7	—	38.2 ⁽³⁾	1.0		FZ
		3				46.1(14.6)	
		5				54.9(14.6)	
		AVE	32.6	36.9	1.5	50.5	
	-423	2	52.6	54.8	1.5		WM
		8	—	29.3 ⁽³⁾	2.0		FZ
		4				54.8(14.4)	
		6				55.2(14.5)	
		AVE	52.6	— ⁽⁴⁾	1.7	55.0	
7039	-320	3	28.4	57.7	13.0		FZ
		8	25.4	49.6	8.0		FZ
		4				20.6(18.0)	
		7				25.4(18.1)	
		AVE	26.9	53.7	10.5	23.0	
	-423	2	35.7	60.2	7.5		WM
		6	34.9	54.5	7.0		FZ
		1				36.6(18.1)	
		5				38.3(18.1)	
		AVE	35.3	57.4	7.2	37.5	

- (1) Bead-off condition
 (2) Stress concentration factor (K_t) in parentheses
 (3) Lack of fusion in weld
 (4) Variation too large to average
 (5) WM—weld metal; FZ — fusion zone

Mechanical Properties of Aluminum Plate Materials and
Their Weldments at Room and Cryogenic Temperatures (GD/A)⁽¹⁾

<u>Alloy</u>	<u>Gauge</u>	<u>Test Temp., °F</u>	<u>Grain Dir.</u>	<u>F_{ty} ksi</u>	<u>F_{tu} ksi</u>	<u>%e in 2"</u>	<u>Notched/Unnotched Tensile Ratio (K_t = 6.3)</u>	<u>Weld Strength ksi⁽²⁾</u>
7006-T6	0.375	75	L	54	59	20	1.33	47
			T	53	58	19	1.32	
		-320	L	64	76	18	1.13	54
			T	61	75	17	1.11	
		-423	L	69	90	19	0.94	49
			T	69	92	18	0.95	
7039-T6	0.500	75	L	54	62	18	1.35	50
			T	54	63	17	1.31	
		-320	L	65	79	21	1.15	58
			T	65	80	17	1.08	
		-423	L	71	94	22	0.97	55
			T	70	95	13	0.93	
2219-T87	0.500	75	L	54	67	13	1.13	42
			T	54	67	11	1.09	
		-320	L	65	84	15	1.07	56
			T	66	85	11	1.03	
		-423	L	72	100	13	1.00	63
			T	73	102	14	0.96	

(1) Data presented represent the average of at least three tests for each condition.

(2) Bead intact; MIG weld process using 5080 filler for 7006-T6, 5556 filler for 7039-T6, and 2319 filler for 2219-T87; as welded without post heat treatment.

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Air Products and Chemicals
INC

$\frac{0.502}{0.504}$ TYP. 2 PLACES

NOTE:

ALLOY	DIMENSION
2219	0.215
7006	0.215
7039	0.325

$\left. \begin{array}{l} +.000 \\ -.010 \end{array} \right\}$

REDRAWN BY D. BUSS 8/11/64

SKETCH OF APCI NOTCHED TENSILE SPECIMEN

This completes the report of General Dynamics/Astronautics.

III. CRYOGENIC IMPACT TESTS

Charpy-V notch impact tests were performed at -320°F and -423°F by Air Products on samples of the candidate inner vessel materials. The following is the summary of the test results.

2219 T87 W/2319 Filler, MIG Process, Post Weld Aging Treatment

The observed slight drop in impact strength with decreasing temperature is consistent with the observed high notch tensile strength.

7039-T6 W/5039 Filler, MIG Process, Post Weld Aging Treatment

Weld zone impact results dropped severely with decreasing temperatures, again substantiating notch tensile results and indicating a sensitivity to welding process.

X 7006-T6

No impact tests were conducted because of the removal of the 7006 alloy from commercial markets.

18% Nickel Maraging Steel, Welded, Post Weld Aged

Low impact results and lack of ductility indicated by the absence of lateral contraction at the notch of welded specimens indicate this material to be quite notch sensitive at -423°F .

Table K-1 presents the test data.

IV. WELDING EXPERIMENTS - 7039 ALUMINUM ALLOY PRESSURE VESSEL

As is concluded in the first section of the appendix, the 2219 aluminum alloy is recommended as the most promising material at this time for lightweight inner shells for Energy Depot liquid hydrogen vessels. This result is based on careful analysis of data taken during the last months of the program. It should be understood that approximately half-way through the program the inner shell material for the lightweight experimental 50-gallon vessel was selected. Of the promising candidate inner vessel materials⁽⁴⁾, only 7039 was then available in the thickness range required. Therefore the AEC approved Air Products' recommendation that the experimental vessel be made from the 7039 alloy. Published material data for the 7039 alloy looked favorable. Preliminary weldability studies using the MIG (Metal-arc Inert Gas) process and experimental X5039 aluminum alloy filler metal on plate thickness of 1/4-inch, 3/8-inch, and 1/2-inch indicated no extraordinary crack sensitivity for

TABLE K-1

CRYOGENIC IMPACT TEST DATA ON CANDIDATE

INNER VESSEL MATERIALS

Material	Condition	Location	Temp.	*Results
2219	T87, 2319 filler, MIG process post weld aging treatment	Base Metal	-320°F -423	4,4,4 ft.-lbs. 4,4,5,5
		Heat Affected Zone	-320 -423	5,6,7 4,5,5,5
		Weld Zone	-320 -423	4,4,5,4.5 3,3,3,5
7039	T6, 5039 filler, MIG process, post weld aging treatment	Base Metal	-320 -423	4,4,4.5 4,4,4
		Heat Affected Zone	-320 -423	 3,3,3
		Weld Zone	-320 -423	1 1,1,1
18% Nickel Steel	18% Nickel welding electrode, post weld aging treatment to 220,000 psi yield strength approximate	Base Metal	-320 -423	6.5,7,7 5.5,6.5,10**
		Heat Affected Zone	-320 -423	 3,4,5.5**
		Weld Zone	-320 -423	2,3,2**

*Charpy-V notch, 10 mm x 10 mm

**These fractures were accompanied by a complete lack of lateral contraction at the base of the notch.

the 7039 aluminum alloy. Because of these encouraging results with conventional, commercial welding techniques, no further welding development effort was recommended or performed prior to the attempted fabrication of the experimental lightweight vessel.

Vessel construction initially was to consist of two 0.045-inch flanged and dished 20-inch diameter heads butt-welded to a 0.045-inch shell section 31-inches long. Heads were spun and heat treated to the T6 temper. Material availability limitations required the heads to be formed from 0.092-inch material, and the subsequent spinning operation reduced the wall thickness at the flange to approximately 0.075-inch. Shell material consisted of 0.063-inch material, chemically milled except at the longitudinal and circumferential weld joints to 0.045-inch. A one-inch unmilled edge was left along each of the two anticipated longitudinal joints as well as the two circumferential joints.

Initial experimental longitudinal welds were made using an automated MIG process, 0.035-inch diameter wire, square butt-weld preparation, and a copper chill bar. Hold-down "fingers" were located approximately 1/2-inch away from the prospective weld joint. Moderate success was encountered with this setup, although a minor amount of burn-through together with porosity was encountered. Mechanical properties of the heat treated joints were slightly lower than had been anticipated from results on heavier sections.

Welding of the longitudinal tank joints using identical welding variables resulted in major burn-through, major porosity, and minor transverse cracking of the weld deposit. Repairs using the TIG (Tungsten Inert Gas) process without filler metal addition were found to result in severe transverse cracking. Repairs using the TIG process with a filler metal addition (X5039) also resulted in transverse cracking.

Additional experimental tests on comparable thickness material with some alterations in welding variables did not cure the existing problems. It was considered that excessive cooling rates in the weld deposit might contribute to weld cracking as well as to porosity. The hold-down "fingers" were therefore moved back from the weld joint to a distance of 3/4-inch and the copper backup bar was heated to approximately 150°F before welding. Neither of these approaches produced good results. Possible excessive restraint produced by the hold-down jig was reduced by the insertion of a high durometer rubber mat between the hold-down "fingers" and the piece, again with no change in results.

An alteration in the hold-down jig at this time did eliminate the burn-through problem observed previously. The alteration was purely mechanical in nature and produced better and more consistent contact between the piece and the copper backup bar.

A change in backup bar material, from copper to stainless steel, produced no appreciable change in results.

During this period it was established that inclusions embedded in the weld wire were contributing seriously to the porosity observed in the weld deposit. Additional wire obtained from the manufacturer was essentially no better in quality. It was decided to continue testing, because of the occurrence of both porosity and cracking, with a modified weld wire composition (X5180) produced by another manufacturer (Alcoa). Although porosity was eliminated, no decrease in crack sensitivity was observed.

Intermediate thickness material (0.125-inch) was chosen for additional tests using the X5180 weld wire composition. Sample results were excellent, and shell material was prepared of comparable thickness material using, again, a square butt-weld preparation, copper backup bar, MIG consumable wire process, and 1/16-inch diameter weld wire. Shell welds, while regular in contour, bead reinforcement, etc., again produced transverse weld bead cracking at extremely inconsistent intervals. It was found, however, that repairs could usually be made by running full-length MIG passes over existing welds. No starts or stops could be tolerated in the weld bead. In a few cases, additional cracking occurred during this repair procedure which did not yield the necessary 62 linear inches of consistent, high quality weld metal.

Initial results utilizing a fully automatic TIG process in place of the MIG process used for most of the experimental work in this project do not indicate that a change in process will materially affect the results obtained.

Base material and filler material are both suspect at this stage of the investigation, and it appears likely that weld metal dilution by the parent metal may play an important part in the problem. The material manufacturer (Kaiser Aluminum Corp.) with whom Air Products has been in close contact during the course of this investigation, is pursuing this problem.

In order to produce an experimental vessel in time to allow testing, even a vessel with much lower strength characteristics, a 4043 aluminum-silicon weld wire was finally used for butt joints between a 7039 head and a 5083 aluminum alloy shell, and the 5083 alloy shell was constructed with a longitudinal weld joint utilizing 5183 filler metal.

APPENDIX L

ALUMINUM-STAINLESS STEEL WELDING TRANSITION PIECE STUDY

I. INTRODUCTION

The conceptual vessel design established during this contract involves an aluminum shell and support mandrel for the inner vessel and stainless steel piping. The necessity of mechanically or metallurgically joining stainless steel and aluminum components for cryogenic and/or vacuum uses makes an analysis of the methods by which this can be accomplished essential. Mechanically joined pieces are generally considered undesirable because of the possibility of eventual leakage. This type of joint is rarely used since repairs in many instances are impossible. Metallurgically bonded piping joints, ranging in size from 1/2 inch to about 12 inches, have been on the market for several years. However, information about the specific applications and the success of the joints in actual use is not readily available.

Five types of samples were selected for testing under a high degree of restraint. These consisted of: two existing types of transition pieces; and three combinations of materials from which it appeared that usable transition pieces could be made or from which they had been made. These samples are identified below.

- Bi-Braze Corporation, Brooklyn, New York. This type of joint, which has been in use for several years, is presently undergoing tests at several other installations, and utilizes a mechanical locking arrangement together with a brazed aluminum-stainless steel interface.
- Reynolds Metals Company. A relatively new flash butt-welded transition piece, presently being evaluated by several users, is composed of essentially an aluminum-to-copper, and a copper-to-stainless steel flash butt-welded metallurgical joint. No operating performance has been accumulated on this joint up to the present study.
- Aluminum Company of America. This welded transition piece utilizes Duranel, a roll bonded sandwich material composed of alternate layers of aluminum and stainless steel. This study represents the first usage of this material for a cryogenic joint.
- Silver sleeve transition utilizing a screwed mechanical joint and weld joining an external silver sleeve to both the aluminum and stainless steel pipes. A history of apparently successful performance has been accumulated on this type of joint, although no published information is available on the subject.
- All-State Welding Alloys Inc. Aluminum-stainless steel soldered joint which utilizes a 509°F melting point solder. All-State has felt that their solder would satisfactorily wet both aluminum and stainless steel, but were unaware of other efforts to utilize the solder for a joint.

Other cast and brazed transition assemblies which are available on the market may have value. However, the above five chosen assemblies offer (a) a wide range of possible successful manufacturing techniques; (b) the potentially simplest types of manufactured joints; and (c) joints which in some cases have no available history.

II. TEST

A test involved rigidly restraining the test specimen and temperature cycling it between ambient and -320°F . The original intent of the test was to test each joint to failure. The number of cycles to failure was to be the criterion for differentiating between samples as to usefulness. It is true that cycling to -420°F would have been somewhat more desirable, but an evaluation of the cost of this type of testing as compared to the possible information to be gained indicated that liquid nitrogen testing would be satisfactory. Due to the variation of the coefficient of thermal expansion with temperature, the stress on a restrained sample cycled to liquid hydrogen temperature is only slightly more than that incurred during cycling to liquid nitrogen temperature. The severity of test conditions is not likely to be met in actual service, but the intent of the test was, again, to acquire a gradation of usefulness rather than specifically an acceptability of the product for a given application.

Transition joint test samples were chosen close to 7/8-inch pipe size to approximate the joint size to be utilized in the lightweight 50-gallon vessel.

Eight-inch long transition pieces were used for testing. Each piece had welded to each end a 2" x 2" x 1/2" machined block with a screwed coupling in one end block for helium leak check purposes. Transition pieces were placed in a rigid assembly constructed of 9% nickel steel. Figure L-1 is a photograph of the assembly with a joint in the test position. Sheet metal dams were placed around each joint to be tested for liquid nitrogen immersion. Dial gauges were utilized for determination of assembly shrinkage during test, since the test rig itself was found to cool to about -40°F during testing. Because of the high stress level imposed on joints during testing, all joints eventually elongated substantially and shim stock was used ahead of the machined blocks to provide restraint. Helium leak tests were performed at approximately five cycle intervals. In order to determine the net stress imposed on all pieces, each of the pieces was cycled unrestrained for one cycle. The total shrinkage, less the assembly shrinkage, was utilized for calculation of imposed stress.

Figure L-1. TRANSITION PIECE TEST FIXTURE

III. RESULTS

- Bi-Braze Corporation Transition

Stress Imposed: 13,000 psi min.
Cycles to Failure: 15
Failure Location: Braze

- Reynolds Metals Company Transition

Stress Imposed: 8,900 psi min.
Cycles to Failure: Two samples - did not fail during 150 cycles

- Aluminum Company of America, Duranel

Stress Imposed: 9,400 psi min.
Cycles to Failure: Sample did not fail during 150 cycles

- Silver Sleeve Transition

Stress Imposed: 10,000 psi min.
Cycles to Failure: Sample did not fail during 150 cycles

- Soldered Transition, All-State

No testing was accomplished on this joint. Several attempts to provide adequate solder flow and wetting were unsuccessful. Therefore, no suitable test specimen was fabricated.

IV. DISCUSSION

The yielding of the aluminum piping introduced some inaccuracies regarding the actual stress level imposed on the transition joint. However, comparison of these data with expected yield strengths at -320°F indicates that limiting stresses were the actual yield strengths of the aluminum piping.

Three test joints are still functional after 150 temperature cycles. The indication is that, at least in small diameters, several types of transition joints will perform satisfactorily under severe thermal cycling and high stress conditions. The Reynolds flash butt-welded joint was utilized successfully for the experimental lightweight 50-gallon vessel tested under this contract.

APPENDIX M
VESSEL SUPPORT DISC DEVELOPMENT STUDY

I. INTRODUCTION

The data from the first several test runs with the boiler-plate vessel⁽⁶⁾ led to considerable doubt that the predicted heat leak value for the support discs was actually being achieved. For this reason, a small calorimeter, shown in Figure M-1, was built to permit heat leak testing of the support discs. The construction of the calorimeter permitted the testing of single discs under no-load conditions or of a pair of discs with varying degrees of radial loading.

II. EXPERIMENTAL STUDY

The test data obtained with the two pairs of support discs used in the boiler-plate vessel have already been reported⁽⁶⁾. One pair of discs, identified as A and B in the data, were constructed by winding a 1/2-inch wide strip of 0.0015-inch thick stainless steel on a center mandrel to the desired outside diameter. The wound strip was held in place by a grooved outer ring. A small, U-shaped deformation in the center of the stainless steel strip provided axial rigidity to the assembly. The second pair of discs (C and D) were similarly constructed except that a fine layer of 200-mesh manganese dioxide dust was utilized as a spacer between the layers of stainless steel strip, and the strip was wound flat, i.e. with no deformation. The calorimeter test results of 10.5 Btu/hr heat leak for the two undusted discs and 6.6 Btu/hr for the two dusted discs (at 200 pounds total load) confirmed that the actual heat leak through the discs was very much higher than the expected value of 1.8 Btu/hr. The designed value of the heat leak was calculated from test results presented by Mikesell and Scott⁽¹⁸⁾ for similar discs which were 1-inch in width.

The four 1/2-inch wide support discs were fabricated by an outside vendor, and complete information on the details of the assembly procedure was not made available to Air Products.

A further study of the support discs was therefore undertaken by Air Products to investigate some of the variables involved in the heat leak characteristics of this type of support member. A winding apparatus, shown in Figure M-2, was constructed. This apparatus allowed the tension on the stainless steel strip to be controlled throughout the winding process. The support mandrel was held between two 1-inch thick transparent discs which guided the layers of stainless steel into position on the mandrel. Provision was also made for dusting of the strip during the winding process.

A width of 1/4-inch was selected for the new support discs, based on the assumption that the reduction in width would provide some reduction in heat leak. Literature data⁽¹⁸⁾ indicated that the load-bearing capability

Figure M-1. SKETCH OF CALORIMETER

Figure M-2. SUPPORT DISC WINDING APPARATUS

of the 1/4-inch wide discs would be entirely satisfactory for the lightweight vessel. A spring temper grade of stainless steel (Rockwell "C" hardness of 42) was substituted for the fully annealed material (Rockwell "C" hardness of 11) which had been used in the four 1/2-inch wide discs. It was believed that the harder material would deform less around the spacing layer of manganese dioxide dust, providing poorer contact, hence greater thermal resistance. The stainless steel strip was drawn between a stainless steel wheel having a recessed, V-shaped groove and a mating steel tool bit. This imparted a slight V-shaped peak into the strip before winding. The purpose of the V-shape was to provide increased rigidity in the axial direction. Previous experience with the pair of dusted, flat-strip discs (C and D) had shown that the axial rigidity of that type of assembly was unacceptable.

Figure M-3 compares the measured heat leak values for several pairs of 1/4-inch wide support discs with those of the original 1/2-inch wide discs. These data represent the net heat leak due to the support discs alone. A correction of 2.3 Btu/hr has been subtracted from the gross calorimeter heat leak measurements to compensate for "background" heat leak. The value of 2.3 Btu/hr was obtained in separate tests of the "empty" calorimeter and corresponds to additional heat leak due to the insulation, piping, and loading rod. Table M-I shows the pertinent assembly data for each disc which was tested for heat leak performance. The first pair of discs (E and F) tested in this series was wound with 19-pounds tension on the stainless steel strip and was dusted lightly with 200-mesh manganese dioxide dust. One of these is shown just prior to final welding of the outer ring as Figure M-4. The heat leak of these discs, measured at 200-pounds load was 20 Btu/hr, which was 2 to 3 times that of the 1/2-inch wide discs.

Several more discs (G, H, I,) were wound with lesser tensions on the strip. The measured heat leak was reduced to 6 Btu/hr for a pair of discs wound at 5- and 8-pounds tension. However, at 5-pounds tension, the assembled support disc had poor axial rigidity, in spite of the V-shape in the strip.

It was found that the electrical resistance of the support discs, measured between the center mandrel and the outer ring, gave a qualitative measure of the degree of metal-to-metal contact between the layers of the assembled discs. This provided a rough estimate of the amount of heat leak to be expected with each support disc. The amount of contact between layers could be estimated from the electrical resistance of the assembled disc, based on the total length of the strip and the measured longitudinal resistance of 1.1 ohms per foot. Several support discs, not reported in the table, were discarded without heat leak testing due to poor structural characteristics or low electrical resistance (less than 1 ohm) since testing appeared to be of no value.

Figure M-3. SUPPORT DISCS TEST RESULTS

TABLE M-I

ASSEMBLY DATA FOR DEVELOPMENTAL SUPPORT DISCS

<u>Disc</u>	<u>Width (Inches)</u>	<u>Metal Layers</u>	<u>Winding Tension (lbs.)</u>	<u>Electrical Resistance (ohms)</u>	<u>Spacing Material</u>
A	1/2	?	20	1.2	None
B	1/2	?	20	11	None
C	1/2	792	20	12	MnO ₂
D	1/2	782	20	75	MnO ₂
E	1/4	890	19	0	MnO ₂
F	1/4	895	19	1.0	MnO ₂
G	1/4	860	12	2.3	MnO ₂
H	1/4	785	8	58	MnO ₂
I	1/4	725	5	90	MnO ₂
J	1/4	246	10	225	0.004" tk. paper
K	1/4	172	11	180	0.009" tk. nylon
L	1/4	169	11	180	0.009" tk. nylon

Figure M-4. 1/4-INCH WIDE, DUSTED, ALL METAL
SUPPORT DISC

No structurally satisfactory assembly method utilizing the 1/4-inch wide, dusted strip could be visualized which would achieve the originally predicted heat leak value of 1.8 Btu/hr. It was concluded that the inverted V-shape of the strip prevented a dust layer from forming at the peak during fabrication, resulting in a high degree of metal-to-metal contact at this point.

The final assembly method attempted utilized a fabric strip wound between the V-shaped stainless steel strip. This method insured a complete separation between the layers of metal, which was confirmed by the electrical resistance of the assembled discs. The first disc made by this method (J) utilized a spacer of 1/4-inch wide, 0.004-inch thick, coarse-grained paper of the type used for adding machine rolls. The assembled disc, tested in combination with an all-metal, dusted disc (I), exhibited good heat leak characteristics. The heat leak through the paper disc alone was estimated to be about 1 Btu/hr at 200-pounds load. However, the structural characteristics of the disc were totally unsatisfactory with respect to both the radial and axial rigidity. The low strength of the paper prevented it from being wound under tension, which would have compressed the material and increased the rigidity significantly.

A pair of support discs (K and L) were then made utilizing a 1/4-inch wide, 0.009-inch thick strip of Nylon woven from stranded thread (see Figure M-5). The Nylon strip was compressed to about 0.007-inch by applying tension to both the steel and Nylon strips during the winding process. The V-shape of the stainless steel strip was retained for rigidity. Each of the assembled discs had an electrical resistance of 180 ohms. This value corresponded closely to the resistance of the total length of strip used, indicating that no electrical shorting between layers had occurred.

The structural quality of the two Nylon discs was satisfactory with respect to both the radial and axial rigidity. These discs were tested in the calorimeter at several loadings, as is shown in Figure M-3. The heat leak of the pair at 200-pounds load, corresponding to the full weight of the lightweight 50-gallon vessel containing hydrogen, was 2.3 Btu/hr. This value was within 30 percent of the originally predicted heat leak of the support discs for this vessel. The Nylon support discs also affected a considerable weight saving over the all-metal type discs used in the boiler-plate vessel study. The 1/4-inch wide Nylon discs weighed 10 ounces each, compared to 56 ounces each for the 1/2-inch wide, all-metal discs. The Nylon discs were judged fully acceptable for use in the 50-gallon lightweight vessel and were incorporated into that vessel.

Figure M-5. 1/4-INCH WIDE, COMPOSITE NYLON-
STEEL SUPPORT DISC

III. RESULTS

The results of the support disc study are summarized as follows:

- The most satisfactory method of construction for the support discs, from the standpoint of both good structural rigidity and low heat leak, utilized a Nylon strip wound between stainless steel layers. Tension of about 11 pounds was applied to both the Nylon and metal strip during winding.
- A pair of Nylon-stainless steel discs had a heat leak of 2.3 Btu/hr at 200-pounds loading. This was 35 percent of the heat leak of the best pair of all-metal discs.
- The 1/4-inch wide Nylon-stainless steel discs weighed 10 ounces each, or 18 percent of the weight of 1/2-inch wide all-metal discs.

APPENDIX N

50-GALLON LIGHTWEIGHT VESSEL SPECIFICATIONS

I. INTRODUCTION

This appendix consists of the general specifications for a 50-gallon liquid hydrogen vessel designed according to the concepts evolved during this contract. The purpose of this appendix is the documentation of the vessel design as it exists at the end of this development contract.

II. GENERAL VESSEL DESCRIPTION

The general features of the vessel criteria and design concept was discussed in Section VI of this report. Such a discussion is pertinent to this appendix but will not be duplicated here. Reference should also be made to Figure 3 which is part of Section IIB, which is a schematic representation of the vessel design and aids in the understanding of the relationship of the various components to the vessel as a whole.

III. GENERAL VESSEL SPECIFICATIONS

- A. Capacity (gross): 55-gallons
(liquid-nominal): 50-gallons
- B. Liquid to be Handled: Hydrogen
- C. Shape of Inner Vessel: Cylindrical with formed heads;
L/D = 2 (nominal)
- D. Pressure
 - 1. Inner Vessel: internal design pressure at 70°F - 35 psi*
 - hydrostatic test pressure at 70°F - 70 psi*
 - 2. Outer Vessel: external working pressure - 15 psi
 - collapse pressure - 30 psi
- E. Ambient Temperature: + 125°F to -65°F
- F. Liquid Fill Rate: 50 gallons/minute
- G. Vent System: Capable of safe venting during condition of loss of insulation vacuum.
- H. Method of Gas Transfer: Pressurization through vaporization of liquid by means of an electric heater.

*These values are to be considered tentative. Future system studies may prove the desirability of operating the vessel at a higher pressure to conserve fuel. The analysis will relate the maximum operating pressure to the minimum wall thickness dictated by manufacturing requirements. The choice between chemically-milled and non-milled shells will be made partly on the basis of weight.

- I. Liquid Loss Rate: 25% of original full load after seven days storage time.
- J. Vessel Shock Loading: Designed to withstand 10-g loading in any direction.
- K. Vessel Orientation: Horizontal, although with minor piping design changes, the design can be utilized for vertical vessels.

IV. MATERIALS AND FABRICATION SPECIFICATIONS

- A. Outer Vessel:
 - 1. The cylindrical section is to be of bonded 6061-T6 aluminum honeycomb construction or its equivalent as approved by the Contracting Officer.
 - 2. The heads sections are to be 6061-T6 aluminum formed heads.
 - 3. All bonded seams are to be leak free and to be designed for minimum outgassing.
 - 4. The cylindrical and head sections are to be welded by the inert gas shielded tungsten arc process (TIG).
 - 5. During welding, the temperature of adjacent bonded material must be kept as low as possible with 212°F as an upper limit.
- B. Inner Vessel:
 - 1. Material of construction is to be 2219-T87 aluminum.
 - 2. The inner vessel is to be of all welded construction. Welding will be performed by an inert gas shielded metallic arc (MIG) process. All welds are to be 100% x-rayed.
 - 3. The maximum working stress in the inner shell shall be less than one-half of the room temperature yield stress of the vessel material.
 - 4. The welded vessel is to be artificially aged at 325°F for 24 hours.

C. Internal Support System

1. Radial Support for the Inner Vessels:

- a. The full weight of the inner vessel and its contents is to be borne by a hollow support tube running longitudinally through its center and extending beyond it. The tube will be of 6061-T6 aluminum.
- b. The mandrel for the inner vessel will be supported at each end by a composite disc in compression. The most satisfactory disc developed to date involves a tightly wound coil of 0.0015-inch thick, 0.25-inch wide ribbon of stainless steel interspaced with a continuous 0.25-inch wide, 0.009-inch thick strip of Nylon woven from stranded thread.
- c. The steel strip is to be wound under about 11-pounds tension. A slight V-shape should be imparted to the steel strip to assure longitudinal rigidity of the disc.
- d. The inner ring for the disc is to be tack-welded to the inner vessel support mandrel. The outer ring is to be grooved to slide longitudinally on four small pins. This feature will allow the inner vessel and support tube to contract freely and will prevent rotation of the inner assembly about its longitudinal axis.

2. Longitudinal Support for the Inner Vessel

- a. An Invar rod in tension will restrain the inner vessel axially within the outer vacuum shell.
- b. The support rod will extend through the center support tube. The Invar rod will be attached to a cold spider assembly near the center of the support mandrel. The rod will extend longitudinally through both ends of the mandrel and will be attached to spider assemblies at room temperature.

D. Insulation

1. The primary insulation will consist of multiple-radiation barriers in a vacuum space.
2. The gas pressure to be attained between the layers of insulation should be about 10^{-5} mm-Hg or lower.
3. The insulation is to be about two inches thick and must demonstrate a heat flux of approximately 0.2 Btu/hr-ft² or less.

4. Silver-plated copper mirrors are to be positioned to inhibit radiative heat transfer down the interior of the support mandrel.

E. Internal Piping

1. The vent and fill lines will penetrate the vessel through the ends of the support mandrel. Near the longitudinal mid-plane of the vessel, the vent line will extend upward and terminate about 1-1/2 inches from the top interior of the vessel. Similarly, the liquid fill line will extend downward and will terminate about 1/2 inch from the bottom interior of the vessel.
2. The vent and fill lines are to be of thin walled stainless steel wrapped with a few layers of aluminized Mylar. The entire piping assembly will be of all welded or silver soldered construction.
3. In the region where the vent and fill lines penetrate the walls of the support mandrel, the aluminum and stainless steel pipes must be joined by means of transition pieces. These joints must be leak free.
4. A flash butt-welded transition piece manufactured by Reynolds Metal Co., a Duranel roll bonded sandwich joint manufactured by Alcoa, and a silver sleeve-type joint are presently considered suitable for this application.
5. The stainless steel vent and fill lines will leave the vessel through the stainless steel cover plate for an outer shell end cap. Longitudinal contraction of the lines will be taken up with stainless steel bellows.
6. External attachments to the vent and fill lines can be made by any desired technique. External piping can be welded directly to the vessel piping or attached through any convenient coupling suitable for hydrogen use.

F. Vacuum Maintenance and Measurement

1. Connection to a suitable vacuum pumping station will be made from a pipe extending radially from one end cap. The connection for vacuum measurement will be made at the other end cap.
2. An annular shaped cylindrical basket containing activated charcoal is to be located in the central support mandrel. Good thermal contact between the charcoal and the mandrel is necessary to insure that the absorber operates at as low a temperature as possible.

3. Static vacuum of 10^{-5} mm or better must be maintained for at least 5000 hours when vessel contains liquid hydrogen.

G. Gas Delivery System

1. Hydrogen gas will be generated for delivery by vaporizing liquid hydrogen by means of an electric heater.
2. The present experimental vessel contains two internal 450-watt strip heaters. Power is fed to these heaters through solid conductor, Teflon-insulated 18-gauge wire.
3. It is likely that prototype vessels will employ an external and replaceable electric vaporizer for increased system reliability. In this case, the vessel design will be modified for liquid delivery from the vessel.

H. External Vessel Support System

The full weight of the inner vessel, mandrel, and the liquid contents is not borne by the outer shell but is transmitted directly to the end caps. The appropriate external support location is on the end caps close to the outer heads, at a point adjacent to the support discs. Any convenient collar or harness arrangement is suitable. The external support system should supply radial, rotational, and longitudinal restraint to the vessel.

APPENDIX P
50-GALLON BOILER-PLATE VESSEL STUDY:
FINAL EXPERIMENTAL RESULTS

I. INTRODUCTION

The initial tests with the boiler-plate 50-gallon vessel have previously been reported in the Third Quarterly Report⁽⁶⁾. This appendix documents the program and the results for the final test series performed during the period January 18 through February 14, 1964.

II. EXPERIMENTAL STUDY

A. Heat Leak Determination for the No-Flow Condition

For the final experimental heat leak tests performed with the boiler-plate vessel, the vessel was modified by lining both end caps of the outer shell with aluminum foil to reduce the emissivity of these portions of the warm wall. The two copper mirrors at each end were also covered with aluminum foil. The vessel was then re-evacuated and filled with liquid nitrogen according to the usual cool-down procedure.

The heat leak into the boiler-plate vessel was first measured under no-flow conditions by observing the pressure buildup in the closed tank. Data were obtained over a 40.75 hour period, as shown in Figure P-1. During this test period, the tank pressure increased from 38.35 psia (-303.9°F) to 53.35 psia (-297.2°F). This corresponds to an average heat leak of 22.9 Btu/hr, corrected for the heat absorbed by the cold components of the vessel during warmup. The heat leak under no-flow conditions was calculated from the change in internal energy of the vapor-liquid mixture during the test period. The volumetric fractions of vapor and liquid at any time can be found from the equation:

$$\theta_L V_T \rho_L + (1 - \theta_L) V_T \rho_V = W_T, \quad (1)$$

where θ_L = volumetric liquid fraction in vessel;

V_T = internal volume of vessel, ft³;

ρ_L = saturated liquid density, lb/ft³;

ρ_V = saturated vapor density, lb/ft³; and

W_T = weight of vapor-liquid mixture in vessel, lb.

The total enthalpy (H_T) of the vapor-liquid mixture at any time is obtained from the equation:

$$H_T = \theta_L V_T \rho_L H_L + (1 - \theta_L) V_T \rho_V H_V, \text{ Btu} \quad (2)$$

where H_L = saturated liquid enthalpy, Btu/lb; and

H_V = saturated vapor enthalpy, Btu/lb.

Figure P-1. BOILER-PLATE VESSEL TEST RESULTS: January 18 to January 19, 1964

The total heat leak (q_{NF}) into the vessel under no-flow conditions is then calculated from the equation:

$$q_{NF} = \frac{H_{Tf} - H_{Ti} + \sum W_i C_{pi} \Delta T - (\Delta P) V/J}{\Delta t} \quad \frac{\text{Btu}}{\text{hour}} \quad (3)$$

where H_{Tf} = final enthalpy of vapor-liquid mixture, Btu;
 H_{Ti} = initial enthalpy of vapor-liquid mixture, Btu;
 W_i = weight of individual cold component of vessel, lb ;
 C_{pi} = specific heat of individual cold components of vessel, Btu/lb°F;
 ΔT = increase in temperature of cold components, °F;
 Δt = time interval, hours;
 ΔP = change in pressure, psi;
 V = gross volume capacity of vessel, ft³; and
 J = 778 ft lb/Btu

The vessel components included in the heat leak calculation were the inner vessel, the support tube and rod, the charcoal adsorber, the support discs, and the fill and vent lines. The heat leak absorbed by these components was equivalent to 1.3 Btu/hr, or 5.5 per cent of the total heat leak. That is, 21.6 Btu/hr were absorbed by the liquid nitrogen and an additional 1.3 Btu/hr were absorbed by the vessel as it warmed up.

The average temperature of the outer shell of the vessel was maintained constant within 3°F during the test run (69 to 72°F). The deviation in heat leak with shell temperature was estimated to be less than 2.5 percent, based on the assumption that the heat leak will vary with the fourth power of the outer shell temperature if all of the heat leak were due to radiation. Since some of the heat leak, possibly one-half, was by conduction, the actual variation in heat leak should have been closer to 1 per cent. The value of 22.9 Btu/hr obtained under no-flow conditions can be assumed constant within ± 0.5 per cent over the test period of 40.75 hours.

B. Heat Leak Determination under Venting Conditions

At the completion of the no-flow heat leak test, the vessel was refilled with liquid nitrogen at atmospheric pressure for heat leak testing under venting conditions. Heat leak data were obtained

over a 24-day period (573.5 hours), as shown in Figure P-2. The instantaneous heat leak values, indicated by the data points in Figure P-2, were calculated by making periodic measurements of the rate at which the venting nitrogen gas passed through a wet-test meter. The instantaneous heat leak (q_1) was calculated from the equation:

$$q_1 = \frac{2370}{t} \times \frac{530^\circ\text{R}}{T_a}, \text{ Btu/hr} \quad (4)$$

where t = time required for 3 liters of venting nitrogen to pass through wet-test meter, seconds; and
 T_a = temperature of gas at wet-test meter, $^\circ\text{R}$.

The constant (2370) in Equation 4 is a conversion factor calculated from the expression:

$$\begin{aligned} & \frac{3 \text{ liters}}{\text{rev.}} \times \frac{.0353 \text{ ft}^3}{\text{liter}} \times \frac{.07245 \text{ lb N}_2 \text{ gas}}{\text{std. ft}^3} \times \frac{2.405 \text{ Btu/lb-mole}}{28.01 \text{ lb/lb-mole}} \\ & \times \frac{3600 \text{ sec.}}{\text{hr}} = 2370 \frac{\text{Btu sec}}{\text{hr rev.}} \end{aligned} \quad (5)$$

The wet-test meter was calibrated with known gas flow rates at 70°F .

Average heat leak values were determined for the time increments between each set of instantaneous measurements. The average heat leak (q_a) was calculated from the equation:

$$q_a = \frac{116.4 (\Delta V)}{T_a' (\Delta t)} \text{ Btu/hr}, \quad (6)$$

where ΔV = volume of nitrogen passing through wet-test meter, liters;

Δt = time required for above volume of nitrogen to pass through meter, hours; and

T_a' = average gas temperature at meter, $^\circ\text{R}$.

Figure P-2. BOILER-PLATE VESSEL TEST DATA: January 20 to February 14, 1964
(Continued on Next Page)

Figure P-2. Continued BOILER-PLATE VESSEL TEST DATA: January 20 to February 14, 1964

The constant 116.4 is a conversion factor determined by the expression:

$$\frac{0.0724 \text{ lb /std. ft}^3 \times 85.9 \text{ Btu/lb} \times 530^\circ\text{R}}{28.32 \text{ liter/std. ft}^3} = 116.4 \frac{\text{Btu } ^\circ\text{R}}{\text{liter}} \quad (7)$$

The overall heat leak was also calculated from the weight loss of the vessel during the same period, shown in Figure P-3. The heat leak by weight loss (q_{WL}) is given by the equation:

$$q_{WL} = \frac{\Delta W \lambda}{\Delta t} \text{ Btu/hr,} \quad (8)$$

where ΔW = weight loss of vessel during test period, lb;

λ = latent heat of vaporization of nitrogen, Btu/lb; and

Δt = duration of test period, hours.

It should be noted that in Equations 4, 6, and 8, the density correction term, k , shown in Equation 29 of Appendix R, is omitted because it is insignificant (less than 1%) in the cases of interest.

For the 24-day test period, the overall average heat leak, determined from the total volume of nitrogen passing through the wet-test meter, was 19.2 Btu/hr. Over the same time period, the average heat leak determined from the weight loss of the vessel was 18.5 Btu/hr, a difference of 4 per cent. In each case, the heat leak is based only on the amount of heat required to vaporize liquid nitrogen; no allowance for sensible heat is included in the calculations. For the purposes of this discussion, the average value of 18.9 Btu/hr will be used.

III. DISCUSSION OF RESULTS

The heat leak under venting conditions is about 4 Btu/hr lower than that obtained under no-flow conditions. This reduced heat leak reflects the fact that, when nitrogen vapor is continuously vented from the vessel, some of the heat leak into the vessel is dissipated by heating of the venting vapor. Heat equivalent to 98.5 Btu/lb of nitrogen vapor would be required to heat the gas from the vapor saturation point (-320°F) to the average venting temperature ($+70^\circ\text{F}$), as compared to 85.9 Btu/lb required for vaporization alone. This corresponds to an additional heat leak of 21.7 Btu/hr if all of the venting gas were heated to ambient temperature before leaving the vessel. Thus, a reduction of 4 Btu/hr under venting conditions is entirely reasonable in view of

Figure P-3. BOILER-PLATE VESSEL TEST RESULTS: January 20 to February 14, 1964
(Continued on next page)

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Figure P-3. Continued BOILER-PLATE VESSEL TEST RESULTS: January 20 to February 14, 1964

the total sensible heat which could be absorbed by continuous venting of warm vapor from the vessel.

The instantaneous heat leak value, shown in Figure P-2, varied from 15.8 to 26.7 Btu/hr, or 16 percent below to 41 percent above the average value of 18.9 Btu/hr. Under venting conditions, the heat leak is affected primarily by the outer shell temperature, as under no-flow conditions, and by the barometric pressure, which has no effect under no-flow conditions. As the barometric pressure rises, the equilibrium temperature of the liquid nitrogen in the vessel rises, so that a significant amount of heat can be absorbed by the liquid without vaporization. For example, with 200 pounds of liquid nitrogen in the vessel, 120 Btu could be absorbed by the liquid without vaporization after a barometric pressure increase of one inch of mercury. Conversely, as the barometric pressure decreases, the equilibrium temperature of the liquid nitrogen decreases, resulting in an increased rate of vaporization as the liquid loses its superheat. The most rapid change in barometric pressure observed during the test period was 0.1 inches of mercury in a period of one hour. This would result in a change in the instantaneous heat leak of 1.0 to 1.8 Btu/hr during this time increment, depending upon the amount of liquid nitrogen in the vessel (170 to 293 pounds) during the test period. The maximum variation in barometric pressure was 1.5 inches of mercury over a 72-hour period (February 3 to February 6). This rate of change in pressure over this period could produce a change of 4 Btu/hr in the measured heat leak, which is about one-half of the maximum variation which was measured during this time increment.

The outer shell temperature also affects the heat leak, since it changes the over-all temperature difference between the outer boundary of the system and the liquid nitrogen in the vessel. That heat leak into the vessel which occurs by radiation will be significantly affected by the outer shell temperature, since it is proportional to the difference between the fourth power of the absolute temperatures of the outer shell and the liquid. The average outer shell temperature, derived from thermocouple measurements at eight locations, varied between 60 and 78°F during the test period. This variation would produce a total variation of 15 percent in that portion of the heat leak resulting from radiation. The remainder of the heat leak, occurring by conduction, is less significantly affected by the shell temperature. The maximum variation in heat leak by conduction would be 3-1/2 per cent under the above temperature conditions.

A correction to the measured data to compensate for the variation in shell temperature is seriously complicated by the fact that much of the heat leak by radiation and all of the heat leak by conduction is subject to various time delays. For example, the heat leak through the insulation is a combination of radiation and conduction. A change in the outer shell temperature causes a change in the temperature profile

across the 140 layers of insulation. However, the change is not instantaneous, since the heat transfer through each individual layer is by radiation and by conduction at the areas of contact between the layers. Similarly, the heat leak through the fill and vent lines is also by combined radiation-conduction and is subject to another, probably different, time delay than that associated with the insulation. The heat capacity of each of the vessel components also introduces time delay into the response of the heat leak to changes in the temperature profile.

It appears that the variations in the barometric pressure and shell temperature could account for the fluctuations in the measured heat leak values during the test under venting conditions. Due to the lack of sufficient information concerning the interaction of the temperature-pressure effects and the various time delays involved in the heat leak mechanism, it would be of dubious value to attempt to adjust the measured values of any constant, base conditions.

The average heat leak value of 18.9 Btu/hr under venting conditions can be compared to data reported in the Third Quarterly Report⁽⁶⁾ for the same vessel without aluminum foil lining in the end caps. The original heat leak test yielded an average value of 21.0 Btu/hr during an 11-day period at similar ambient conditions. Since no other improvements were made in the vessel, it was concluded that the improvement of about 3 Btu/hr was due entirely to the reduction in the emissivity of the inner walls of the end caps in the modified vessel. This improvement was therefore incorporated into the lightweight tank design and extended to include silver plating of the fill and vent lines in addition to the mirrors. The end caps themselves and the spider supports would be foil lined, due to the difficulty involved in plating these aluminum components.

IV. SUMMARY OF RESULTS

The results of the heat leak tests of the boiler-plate vessel during the final quarter of the program can be summarized as follows:

1. The average heat leak with liquid nitrogen for a period of 40.75 hours under no-flow conditions (pressure buildup) was 22.9 Btu/hr. During the test period, the average temperature of the outer shell of the vessel varied between 69 and 72°F.
2. The average heat leak for a period of 24 days with continuous venting of nitrogen gas from the vessel was 18.9 Btu/hr. A maximum instantaneous value of 26.7 Btu/hr and a minimum instantaneous value of 15.8 Btu/hr were measured. During the test period, the average temperature of the outer shell of the vessel varied between 60 and 78°F, and the barometric pressure varied between 28.6 and 30.1 inches of mercury.

3. The reduction in heat leak under venting conditions is due to the dissipation of some heat into partial warming of the nitrogen vapor above the saturation temperature before it leaves the vessel. Both tests were performed with liquid nitrogen in the vessel.

APPENDIX Q
50-GALLON LIGHTWEIGHT VESSEL STUDY:
EXPERIMENTAL RESULTS

I. INTRODUCTION

This appendix presents the test program and results of the experimental study of the lightweight 50-gallon vessel. This information forms the basis for the summary of the vessel test study presented as Section VII-C in the body of the report.

II. EXPERIMENTAL STUDY

A. Heat Leak Determination Under Venting Conditions with Liquid Nitrogen

On February 29, 1964, pumpdown of the insulation space of the lightweight 50-gallon test vessel was commenced. A leak was discovered and repaired on March 2. The pressure in the insulation space dropped slowly, relative to the rate previously experienced with the boiler-plate vessel. On the following day, when the insulation space pressure had fallen to 22 microns Hg and the pumping station registered 1.4×10^{-5} mm Hg, 242 pounds net of liquid nitrogen were placed in the vessel. The pumping station was operated and was connected to the vessel throughout this and all subsequent experiments. The insulation space pressure began falling and after 19 hours of cooldown, registered 4×10^{-5} mm Hg. The apparent heat leak indicated by the vent gas flow rate was about 20 Btu/hr.

This heat leak was far in excess of the expected value of 11.8 Btu/hr (see Section VII-D in the text). The relatively high pressure in the insulation space was considered to provide the major contribution to the excessive heat leak. For best performance the insulation space must be at pressures of about 10^{-5} mm Hg or lower. Below about 10^{-5} mm Hg, gas conduction in the insulation is negligible. Above this pressure, gas conduction becomes increasingly important and the total heat flux through the insulation increases rapidly with pressure. Figure Q-1 shows this relation of heat flux and pressure. The insulation space pressure was measured in the end cap which is open to the interior of the cold support tube containing the cold adsorber. This entire volume is relatively open to the opposite end cap to which the vacuum pumping station is connected. On this basis, it is reasonable to assume that the measured pressure is an approximation of the lowest pressure in the vacuum space but not necessarily the pressure in between the insulation layers. In addition, it is quite difficult to measure pressure in this range with high accuracy, and errors of the order of a factor of two are not uncommon. Due to the nature of the insulation, there are relatively high diffusion resistances between the interior of the insulation blanket and the volume contained within the central support tube. There is little doubt that the pressure between the layers of insulation was

Figure Q-1. EFFECT OF PRESSURE ON HEAT FLUX FOR VACUUM INSULATION

higher than the pressure at the gage. From the relationship between heat flux and gas pressure in this pressure range, the conclusion was drawn that the relatively poor vacuum was probably largely responsible for the unexpectedly high heat leak.

By enclosing the entire vessel in an atmosphere of helium and testing mass spectrographically for helium in the vacuum station inlet, it was determined that no leak existed between the outside of the vessel and the insulation space. On this basis, it was suspected that the bonded circumferential seam between the honeycomb and the end closure had opened up during welding of the honeycomb cylinder to the outer head. It should be mentioned that the vendor's recommendation, that the bonded material adjacent to the welding area be bathed in water to maintain its temperature at or below 212°F, was followed. However, after welding was completed it was observed that shrinkage caused by welding had produced cracks around the outside bonded seam. These cracks were sealed with an additional coating of epoxy resin. It was quite reasonable to assume that similar cracks existed along the internal circumferential bonded seam, which was now inaccessible for repair. If the interior of the outer shell was exposed through small cracks to the vacuum space, then the outgassing of the bonded structure would present an essentially infinite supply of gas through these cracks. This would make the achievement of high vacuum in the insulation difficult, if not impossible.

It should be stated that the design and the fabrication techniques for the honeycomb shell were not fully developed for this application under this contract. Goodyear Aerospace Corp., the vendor who fabricated the shell, considered that the present state-of-the-art of bonded aluminum honeycomb was sufficiently advanced so that they could supply a shell which could be satisfactorily welded to head sections to demonstrate the basic feasibility of using honeycomb for this application. Goodyear's preliminary time and cost estimates for fully developing a design which they would guarantee to meet the desired performance characteristics were far in excess of the time and funds which were allotted for the procurement of the outer shell under this contract.

Based on the very reasonable assumption that the interior of the bonded honeycomb was open to the vacuum space, a modification was made to the vessel. A 3/16-inch diameter hole was drilled into the honeycomb space through each end closure. A 3/16-inch diameter stainless steel tube was inserted into each of the holes which were then sealed with epoxy resin. The two tubes were joined externally to the vessel and were connected through a bed of activated charcoal at -320°F to the pumping station. A thermocouple-type vacuum gage was tapped into the line upstream of the charcoal bed. The purpose of this modification was an attempt to evacuate the honeycomb

externally in order to reduce the gas flow rate into the insulation space. By this short-circuiting of the gas evolved in the honeycomb past the main vacuum space, it was reasoned that the tank vacuum could be reduced.

On March 6, pumpdown of both the insulation space and the outer shell was commenced. After 25 hours of pumping, the pressure in the insulation space had reached 10 microns Hg while the shell pressure registered 400 microns Hg. Two hundred-forty pounds of liquid nitrogen were transferred to the vessel.

The results of the heat leak testing of the vessel as modified by measuring the boiloff of liquid nitrogen at one atmosphere are shown in Figures Q-2 and Q-3. The parameters affecting the results, viz. atmospheric pressure, shell temperature and insulation space pressure are shown in Figure Q-2, while the results of the three different methods of measurement of heat leak are shown in Figure Q-3. Heat leak values under venting conditions were calculated by techniques discussed in Appendix P. For the first 120 hours after fill the ambient temperature and the shell temperature were usually maintained at about 65°F to 70°F and the insulation space pressure slowly dropped to about 7×10^{-6} mm Hg. The average heat leak during the first 75 hours was about 20 Btu/hr. Note that at 70 hours a sharp peak occurred in the boil-off flow rate. This was undoubtedly caused by an unusual decrease in atmospheric pressure. The effect of rapid changes in atmospheric pressure probably extended up until the 100-hour mark, thus masking the effects of ambient temperature and insulation space pressure. At 120 hours from fill, the ambient temperature was reduced in an effort to improve the vacuum by decreasing the outgassing rate in the shell. Shell temperature was maintained at about 55°F from 140 to 220 hours which did reduce the insulation space pressure to about 4×10^{-6} mm Hg. Average boil-off rate during this period corresponded to a heat leak of 16.5 Btu/hr. Ambient and shell temperature were then reduced to about 40°F by circulating unheated outside air overnight through the laboratory. The insulation space pressure dropped somewhat. It was impossible to maintain this temperature long enough to obtain accurate boil-off data at this condition. At 240 hours from fill the ambient temperature was returned to normal and the shell temperature held at about 65°F for the duration of this test. Insulation space pressure appeared to follow minor fluctuations with shell temperature. Average heat leak during this interval was about 17.1 Btu/hr.

B. Heat Leak Under No-Flow Conditions with Liquid Nitrogen

On March 23, 1964, the valve in the vent line was closed and the pressure in the vessel was allowed to increase. A mercury manometer connected to the vapor space was read regularly for 94.5 hours, the

Figure Q-2. LIGHTWEIGHT VESSEL TEST DATA: March 7 to March 23, 1964

Figure Q-3. LIGHTWEIGHT VESSEL TEST RESULTS: March 7 to March 23, 1964

length of time required for the pressure to rise from 14.7 psia to 53.2 psia. Figure Q-4 is a plot of the vessel pressure as a function of time. Room and shell temperatures were held at $70^{\circ}\text{F} \pm 2^{\circ}\text{F}$ during this interval. The procedure for calculating heat leak from pressure rise data is discussed in Appendix P in relation to the boiler-plate vessel study. These data indicate that the heat leak into the non-venting lightweight vessel is 21.0 Btu/hr with liquid nitrogen.

The heat leak under venting conditions was about 17.1 Btu/hr, or 82% of that obtained under no-flow conditions. This reduction is well within that which could be expected due to heat dissipation into warming of the venting vapor. This sensible heat effect was discussed in more detail in Appendix P.

C. Heat Leak Determinations with Liquid Hydrogen

On March 31, 1964 approximately 30 pounds of liquid hydrogen were placed in the vessel. Figure Q-5 is a plot of the instantaneous heat leak indicated from vent gas flow rate, the barometric pressure and the average outer shell temperatures as a function of time. Vessel pressure was maintained at one atmosphere. During all the hydrogen testing the insulation space pressure was between 1 and 2×10^{-6} mm Hg. Because this pressure held relatively constant, these readings are not presented in Figure Q-5. Figure Q-6 is a plot of the instantaneous heat leak, the integrated boil-off volume, and the liquid weight as a function of time period following liquid hydrogen cooldown. As is shown in Figure Q-6, the integrated boil-off and the weight loss techniques indicate heat leaks of 17.8 Btu/hr and 17.6 Btu/hr, respectively. The instantaneous heat leak values were steady at 17.5 to 18.0 Btu/hr except for the period between 50 and 80 hours. During this period barometric pressure decreased rapidly and shell temperature increased, both tending to raise the boil-off rate. The average total heat leak of the vessel for liquid hydrogen under flow condition is 17.7 Btu/hr.

On April 6, 1964, the valve in the vent lines was closed. The vessel pressure rose from 24.8 psia to 43.55 psia in a span of 7.87 hours. The indicated heat leak for this vessel is 26.3 Btu/hr for liquid hydrogen under no-vent conditions. This data is presented as Figure Q-7.

SUMMARY OF RESULTS

The results of the heat leak tests of the lightweight vessel after modification to improve the vacuum are summarized as follows:

1. The average heat leak with liquid nitrogen during the last 120 hour period of continuous venting at atmospheric pressure was 17.1 Btu/hr. During this period the recorded insulation space pressure varied between 8 and 3×10^{-6} mm Hg. Average outer shell temperature varied between 55 and 75°F .

Figure Q-4. LIGHTWEIGHT VESSEL TEST RESULTS: March 23 to March 27, 1964

Figure Q-5. LIGHTWEIGHT VESSEL TEST DATA: March 31 to April 6, 1964

Figure Q-6. LIGHTWEIGHT VESSEL TEST RESULTS: March 31 to April 6, 1964

Figure Q-7. LIGHTWEIGHT VESSEL TEST RESULTS: April 6, 1964

2. The average heat leak with liquid nitrogen during a 95-hour period of non-venting was 21.0 Btu/hr. Average outer shell temperature varied between 67 and 71°F. The recorded insulation space pressure varied between 7 and 3×10^{-6} mm Hg.
3. The average heat leak with liquid hydrogen during the 145-hour period of continuous venting at one atmosphere pressure was 17.7 Btu/hr. The average outer shell temperature varied between 55 and 78°F. The recorded insulation space pressure varied between 3 and 1×10^{-6} mm Hg.
4. The average heat leak with liquid hydrogen during an eight-hour period of non-venting was 26.3 Btu/hr. The average outer shell temperature was about 73°F during this period. The recorded insulation space pressure was about 2×10^{-6} mm Hg.

A comparison of this data with the expected heat leak is presented in Section VIID of this report.

APPENDIX R
THERMODYNAMIC ANALYSIS OF LIQUID HYDROGEN STORAGE

In Section IIB of the First Quarterly Report⁽⁴⁾, a thermodynamic analysis of liquid hydrogen storage was presented. This analysis included a derivation of the expression for the heat added to a vessel which is venting at a constant pressure. The expression so derived is correct. The present discussion is to clarify the derivation of the expression.

Nomenclature

M = mass of fluid in vessel, lb
 E = internal energy, Btu/lb
 α = fluid vented, lb
 P_o = atmospheric pressure, psia
 P_2 = venting pressure, psia
 v = specific volume, ft³/lb
 ρ = density, lb/ft³
 Q_2 = heat added to vessel, Btu/lb
 V = vessel volume, ft³
 H = enthalpy, Btu/lb
 λ = latent heat of vaporization, Btu/lb
 $\psi = \frac{\alpha}{M_2}$
 $k = \frac{\rho_l}{\rho_l - \rho_v}$

subscripts:

o = atmospheric conditions
 v = vapor
 l = liquid
 2 = during or at start of step 2
 none = conditions inside tank

This analysis applies to a two-phase system in equilibrium, venting saturated vapor at vent pressure, P_2 , to the atmosphere at pressure P_o , through a throttling valve (i.e. no enthalpy change in fluid stream across valve).

The change of the internal energy of the stored fluid is:

$$\Delta (ME) \quad (1)$$

The flow from the vessel is $-\alpha$; therefore the energy removed from the vessel by the venting vapor is:

$$- E_O \alpha \quad (2)$$

The amount of work this stream does in the atmosphere is:

$$- P_O v_O \alpha \quad (3)$$

In addition, energy is added to the tank by the heat leak:

$$M_2 Q_2 \quad (4)$$

An energy balance on the system yields energy accumulation equal to:

$$(\text{energy in} - \text{energy out}) \quad (5)$$

Substituting (1) through (4):

$$\Delta(ME) = M_2 Q_2 + E_O \alpha + P_O v_O \alpha , \quad (6)$$

$$\text{or} \quad \Delta(ME) = M_2 Q_2 + H_O \alpha . \quad (7)$$

Also:

$$M = M_V + M_1 , \quad (8)$$

$$\text{or} \quad ME = M_V E_V + M_1 E_1 , \quad (9)$$

$$V = v_V M_V + v_1 M_1 . \quad (10)$$

Substituting (9) into (7), one obtains:

$$\Delta (M_V E_V + M_1 E_1) = M_2 Q_2 + H_O \alpha , \quad (11)$$

or

$$[E_V \Delta M_V + M_V \Delta E_V + E_1 \Delta M_1 + M_1 \Delta E_1] = M_2 Q_2 + H_O \alpha , \quad (12)$$

since $P, T = \text{constant}$, $\Delta E_v = \Delta E_l = 0$

and
$$E_v \Delta M_v + E_l \Delta M_l = M_2 Q_2 + H_o \alpha. \quad (13)$$

Also, note that:

$$\Delta M_v + \Delta M_l = \alpha, \quad (14)$$

and from (10),

$$\Delta V = 0 = M_v \Delta v_v + v_v \Delta M_v + M_l \Delta v_l + v_l \Delta M_l. \quad (15)$$

Since:

$$\Delta v_v = \Delta v_l = 0, \quad (16)$$

then:

$$v_v \Delta M_v + v_l \Delta M_l = 0. \quad (17)$$

Therefore, through a combination of (13), (14), and (17) one obtains:

$$\left[\frac{E_l - E_v}{1 - \frac{\rho_v}{\rho_l}} + E_v - H_o \right] \alpha = M_2 Q_2 \quad (18)$$

or:

$$\frac{M_2 Q_2}{\alpha} = k (E_l - E_v) - (H_o - E_v). \quad (19)$$

Also:

$$\lambda = H_l - H_v = E_l + P_2 v_l - E_v - P_2 v_v \quad (20)$$

or:
$$E_l - E_v = \lambda - P_2 (v_l - v_v). \quad (21)$$

Also:

$$H_O - E_V = H_O - (H_V - P_2 v_V). \quad (22)$$

Since: $H_O = H_V$ for a throttling process,

$$H_O - E_V = \frac{P_2}{\rho_V} \quad (23)$$

By substituting Equations (21) and (23) into (19) and rearranging, one obtains:

$$\frac{M_2 Q_2}{\alpha} = k \left[\lambda + P_2 \left(\frac{\rho_1 - \rho_V}{\rho_1 \rho_V} \right) \right] - \frac{P_2}{\rho_V}. \quad (24)$$

Since:

$$k = \frac{1}{1 - \frac{\rho_V}{\rho_1}} = \frac{\rho_1}{\rho_1 - \rho_V}, \quad (25)$$

therefore,

$$\frac{M_2 Q_2}{\alpha} = k \lambda + P_2 \left(\frac{\rho_1}{\rho_1 - \rho_V} \right) \left(\frac{\rho_1 - \rho_V}{\rho_1 \rho_V} \right) - \frac{P_2}{\rho_V}, \quad (26)$$

or:

$$\frac{M_2 Q_2}{\alpha} = k \lambda. \quad (27)$$

By rearrangement, one obtains:

$$Q_2 = k \lambda \psi. \quad (28)$$

This expression for the heat added to a vessel venting at constant pressure, is the same as presented earlier as Equation (4), page 22, Reference 4. The present derivation, however, is more straight-forward than the previous derivation.

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