AIAA - 2000-3791

DESIGN AND TESTING OF DEMONSTRATION UNIT FOR MAINTAINING ZERO CRYOGENIC PROPELLANT BOILOFF

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Abstract

Launching of cryogenic propellants into earth orbit and beyond is very expensive. Each additional pound of payload delivered to low earth orbit requires approximately 35 pounds of additional weight at liftoff. There is therefore a critical need to minimize boiloff in spacecraft long term missions/systems. Various methods have been used to date, including superinsulation and thermodynamic vents to reduce boiloff. A system was designed and tested as described herein that will totally eliminate boiloff. This system is based on a closed-loop, two-stage pulse tube refrigerator with a net refrigeration of four watts at 15K for the recovery of hydrogen propellant. It is designed to operate at 30 Hz which is an order of magnitude higher than other typical pulse tube refrigerators. This high frequency allows the use of a much smaller, lighter weight compressor. This paper describes the system design, fabrication and test results.

Background And Introduction

The need for a means of propellant boiloff recovery has been recognized almost since the beginning of the U.S. space program. In fact, Dr. Werner Von Braun, famous rocket scientist and former Director of NASA/MSFC was personally interested in and funded work in this area. These efforts were of particular interest to him with respect to a Mars mission. Some of those early efforts are documented in Reference 1. Since that time, significance improvements have been made in spacecraft cryogenic refrigeration technology which we have incorporated into our current efforts. Spacecraft Cryogenic propellant tanks are normally vented periodically to prevent excessive pressure buildup. This pressure buildup is caused by the boiling off of propellant due to heat leaks through the tank walls, insulation and various structural penetrations. In the real world, this heat leak can never be entirely eliminated no matter how good the insulation and design. This venting obviously wastes vitally needed propellants. Each pound of propellant delivered to low earth orbit requires about 35 pounds of weight at launch. For other higher orbits and interplanetary missions the launched-weight-to-payload ratio is of course even higher. There is therefore a critical need for a means to recover this propellant boiloff and return it to its tank. The efforts described in this paper are aimed at meeting this need.

This work was performed under a NASA/MSFC Small Business Technology Transfer Contract. Dean Applied Technology Co., Inc. (DATCO) was the Small Business Concern with the University Of Alabama (UAH) in Huntsville as the partnering Research Institution.

Summary Of Work Accomplished

This effort was accomplished in two phases. Phase 1 investigated a range of possible methods for boiloff recovery, selected the best approach and determined the feasibility of this approach. The pulse tube refrigerator was selected as the best solution. In Phase 2, this system was designed, fabricated and tested.

During Phase 1, the following methods were considered for boiloff recovery:

- *A total of twelve Active Refrigeration cycles*
- *A direct - condensing space radiator*
- *Use para-to-ortho hydrogen heat of conversion as a low temperature heat sink in a condensing heat exchanger*
- *Hydrogen boiloff combustion as a power source to drive an active refrigeration system, thus eliminating the need for spacecraft electrical power.*

During Phase 1, it was conclusively shown that it is feasible to use Pulse Tube Refrigeration Technology to reliquefy cryogenic propellant boiloff and that this technology has the following advantages:

- *Uses a simple tube known as a "pulse tube", an orifice and a reservoir rather than a moving expander at the cold end to produce refrigeration*
- *No vibration produced at cold end*
- *Can be flight qualified on the ground, thereby eliminating expensive flight testing*
- *Dynamics are significantly simpler than other cycles*
- *Fewer moving parts - hence*
  - *Longer life*
  - *Higher reliability*
  - *Lower cost*
- *Uses no CFC or HCFC refrigerants*
- *Requires no phase change in zero-g, hence no boiling or condensing heat transfer or phase separators*
- *There is currently significant interest and efforts to commercialize this technology in Japan, China and USA.*

As a result of Phase 1 efforts, it was concluded and recommended that a pulse tube refrigerator should be built and tested.

Requirements/Groundrules

The first step of Phase 1 was to establish groundrules and requirements. Our boiloff rates were based on the results of Reference 1 design which used the following:

- *Tank diameter: 3.9m*
- *Tank length: 9.1m*
- *Insulation: 4 MLI blankets each 2.54 cm thick*

This resulted in a hydrogen boiloff rate of .028 Kg/hr (.062 lbm/hr) at a pressure of 101K-N/m^2 (14.7 Psia). This resulted in a heat load of approximately four Watts. We selected a temperature of 15K at which this four Watts of refrigeration must be produced. This gives us a temperature drop of 5K across the condensing heat exchanger for liquefying hydrogen at 20K at the given pressure.

Active Refrigeration Cycles Considered

A trade study was performed which included seven recuperative refrigeration cycles and five regenerative refrigeration cycles. An investigation was performed on each of these 12 cycles which provided information on history/background, how it works, current applications, advantages and disadvantages and methods for application to our propellant boiloff problem. This trade study employed the Kepner-Tregoe Analysis (KTA) Trade Methodology which provides an explicit, quantitative comparison of the candidate, alternative systems and results in numerical scoring of each alternative. This method accounts for varying importance of evaluation criteria by assigning weighting factors to each criteria. Each cycles was scored with respect to 15 criteria. The results yielded a score of 9330 points for the pulse tube compared to 7130 for its nearest competitor and 1845 for the cycle with the lowest score. Therefore the pulse tube was selected as the system to be employed for this application. Using our best engineering judgment based on experience in dealing with numerous space programs, the following weighting factors as shown in Table 1 were assigned to each of the 15 evaluation criteria.

Criteria	Weighting Factor
Weight	10
Volume	4
Power	10
Cost	10
Life	9
Vibration	3
Complexity	9
Cold-End Moving Parts	6
Compression Ratio	6
Staging Capability	6
Single Compressor to Drive Multiple Stages	6
Experience Base	4
On-going Supporting Research/Development by Others	2
Technology Readiness Level	3
Flight Qualification on Ground	7

Table 1: Weighting Factors Assigned To Each Evaluation Criteria

The following 12 refrigeration cycles were considered:

- *Precooled Linde-Hampson*
- *Claude*
- *Helium-Refrigerated*
- *Cascade*
- *Kapitza*
- *Heylandt*
- *Dual-Pressure Claude*
- *Stirling*
- *Pulse Tube*
- *Vuilleumier*
- *Gifford - McMahon*
- *Duplex Stirling*

Space Radiator Considerations

Any time that thermal energy has to be rejected from a spacecraft, the first and most obvious method that comes to mind is to use a radiator. Deep space provides a virtually infinite heat sink if it can be made compatible with other spacecraft requirements. Various thermal radiators have been used successfully on spacecraft for years. We therefore investigated the feasibility of using a radiator to perform the heat rejection necessary to meet the stated requirements of this boiloff recovery system.

A thermal analysis model was set up consisting of 1,000 nodes radiating to an input sink temperature. A mass flow rate of hydrogen flowing through the radiator of .06257 $lb_{/hr}$ was used in all cases per the established requirements. Specific heat, view factor to space, emissivity, and initial temperature were input as variables. A varying incremented area of each node was input. These inputs were then used to calculate outlet temperature, total surface area and total heat rejected. The temperature step between nodes was not allowed to exceed approximately 1.8 degrees K. Itterations were performed until the area needed to meet the needed outlet temperature and/or heat rejection were attained.

A total of six scenarios or cases were analyzed as described in Table 2:

Case No.	Case Description
1A	Boiloff circulated directly through radiator
1B	Circulate boiloff through vapor shields before entering radiator, remove superheat added by vapor shields
2A	Compressor used to raise temperature of boiloff before entering radiator - no vapor shields
2B	Use J-T valve to return pressure to 14.7 Psia after existing the radiator - vapor shields used
3A	Use recuperative heat exchangers before compressor and before JT valve - no vapor shields
3B	Same as 3A but with vapor shields

Table 2: Description Of Cases Analyzed In Space Radiator Considerations

The conclusion of these efforts was that this system was not found to be feasible because of the high pressure, large radiator areas, and low effective sink temperature required.

Investigation Of The Use Of Para-Ortho Conversion Of Liquid Hydrogen As A Heat Sink

Hydrogen can exist in either of two states, Parahydrogen (p-H_2) and Orthohydrogen (o-H_2). The difference in these two forms is in the relative orientation of the nuclear spins of the two atoms composing the diatomic molecule. In the ortho form, the nuclear spins of the two atoms in the molecule are parallel, that is in the same direction. In the para form, the spins are antiparallel. There is a significant difference in energy levels of these two states. The heat of conversion can be up to 702 J/gm (302 BTU/lbm). This is significant when compared to the heat of vaporization which is 446 J/gm (192 BTU/lbm). The equilibrium ratio of para-to-ortho form depends upon temperature. "Normal" hydrogen (n-H_2) refers to hydrogen at ambient and higher temperatures, and consists of 75% ortho and 25% para. At the boiling point of 20K and one-atmosphere pressure, the equilibrium composition is essentially 100% para. The equilibrium mixture of hydrogen at any temperature is known as "equilibrium" hydrogen (e-H_2). If liquid hydrogen is produced in the ortho state, it will gradually and spontaneously convert to its equilibrium state of 100% para with the corresponding release of the heat of conversion. Since the heat of conversion is greater than the heat of vaporization, the liquid hydrogen will vaporize a major portion of itself even without the addition of any heat from outside sources such as conduction through the tank walls and insulation. Thus, liquid orthohydrogen is unstable and will result in loss of about 70% of its mass due to spontaneous conversion over a period of time. Therefore for storage reasons, liquid hydrogen is produced commercially in the 100% para form. The conversion from ortho to para is carried out as an integral part of the liquefaction process using heat exchangers and catalytic converters. Two basic types of catalysts have been developed, namely iron-oxide and nickel-silicate catalysts. The iron oxides are cheaper but less effective, i.e. less active than the nickel-silicates. During this contract we investigated the possibility of using this conversion process energy change as a heat sink for the reliquefier system. The ortho to para conversion is exothermic, releasing heat, while the para to ortho conversion is endothermic, requiring heat to be added thus having the potential of serving as a heat sink in one side of a heat exchanger. This concept was previously mentioned in Reference 1, but was not thoroughly investigated. Operation of this concept is as follows (quoted from Reference 1): "---- parahydrogen from the storage tank is heated in a counter-flow heat exchanger to approximately 125R. At this temperature, it is passed through a catalyst to promote the conversion of a portion of the parahydrogen to orthohydrogen. The energy absorbed in the conversion process is employed as the heat sink for a reliquefier. The partially converted hydrogen stream is then cooled in the opposite side of the counter-flow heat exchanger and returned to the storage tank. The net effect of this system is that the heat transfer through the tank insulation and the energy supplied to the reliquefier compressor are stored in the hydrogen. The orthohydrogen so produced will, of course, spontaneously but slowly convert back to parahydrogen and liberate the stored energy. However, for up to the order of 1000 hours, the forward conversion rate can be made to exceed the back conversion rate and no-boiloff-loss conditions can be maintained."

Our concept differs from that of Reference 1, in that we want to use hydrogen in the liquid state whereas Reference 1 proposed to use vapor phase. In Reference 1, the temperature of the vapor would be raised to 70K where the equilibrium composition was other than 100% para. The addition of heat in the presence of a catalyst would force the mixture to the equilibrium composition with the accompanying absorption of heat from the portion of the stream to be cooled. The converted portion would be vented carrying with it the absorbed heat. Thus a system could be designed and operated that could at best only recover some fraction of the boiloff.

Our system as conceived would provide a heat sink at the liquid hydrogen temperature of 20K and a heat of conversion of 702 J/g (302 BTU/lb) which would be a real "gold mine" breakthrough in dealing with the boiloff recovery problem. This converted liquid could be stored separetely until used in the next engine burn.

However, in order for this concept to work, the liquid must be converted out of its equilibrium state "backward" to some non-equilibrium condition. When dealing with the initial liquid hydrogen storage problem years ago, the developers were working with getting the mixture to progress "forward" toward equilibrium which they were able to speed up by the use of catalysts. When they found a solution to their storage problem, there was no further commercial interest in the "backward" conversion problem.

We thoroughly searched the literature for clues on how to do a para to ortho conversion. A small portion of the early work dealt with conversion in the liquid state. However, the practical solution was to do the conversion at higher temperatures in the vapor state. Therefore there is little information or experience available in the literature for conversion in the liquid state. We were unsuccessful in finding a way of producing this process. Therefore we were unsuccessful in utilizing this concept for our recovery problem.

Preliminary Design

As a result of these Phase 1 efforts, the pulse tube concept was selected as the best method for this application. A preliminary design was developed as shown on Figure 1. This is a three-stage concept which lifts four Watts of cooling from 15K to 290K. Heat is assumed to be rejected to the central spacecraft thermal control system such as a coolant fluid loop. This design is based on the orifice pulse tube concept whose operation is described below. Each of the three stages consists of a pulse tube, heat exchangers, and regenerator with orifice and reservoir which provide the phase lag between pressure and mass flow in the pulse tube which is required to produce cooling. This design also utilizes the double inlet concept to further enhance the phase lag and provide additional cooling capacity. The first stage lifts heat from 195K to 290K, the second stage from 51K to 195K and the third stage from 15K to 51K. Gaseous hydrogen propellant boiloff is passed through the cold-end heat exchanger at 15K where it is condensed into liquid.

Pulse Tube Refrigeration History And Background

The Pulse Tube Refrigeration Cycle is a relative newcomer compared to other refrigeration cycles. In 1963, Professor Gifford of Syracuse University and his graduate student, R. Longsworth, noticed that blanked off plumbing lines connected to gas compressors became hot at the closed end. By connecting the plumbing line to a compressor through a regenerator, cooling was achieved at one end and heating at the other, thus the birth of the Basic Pulse Tube Refrigerator. The original one-stage cooler was reported to have achieved 150K while a two-stage device achieved 120K. After a few years, the coolers had reached 120K and 85K, respectively. Although the cooler raised much curiosity, its efficiency was disappointing and by the end of the 1960's, its use as a practical cooler was not being pursued. In the late 1970's, Dr. J. Wheatley of the Department of Energy's Los Alamos National Laboratory became interested in a related technology - thermoacoustic engines and coolers. These devices are driven to operate at acoustic resonance with a loudspeaker and have no moving parts other than the working fluid and the loudspeaker. However, its efficiency was lower than the Basic Pulse Tube and it did not see practical applications.

In 1981, Dr. Pete Kittle of NASA Ames Research Center, who is the source of much of this information on the history of Pulse Tube Refrigeration, heard a talk by Dr. Wheatley and recognized that a cooler with a single moving part had much potential for space applications. The single moving piston in the Pulse Tube gave it the ability to be more reliable, simpler to control, and cheaper than the Stirling Cycle Cooler. In addition, since the Pulse Tube has no moving parts at the cold end, it offers longer lifetime in cryogenic applications and eliminates vibration at the cold end which is great for detector applications.

Shortly thereafter, Dr. Kittle joined with Dr. Ray Radebaugh of the National Institute of Standards and Technology (NIST) and began developing Pulse Tubes as cryocoolers. In 1983, they made a breakthrough when, without adding any moving components, they were able to improve the efficiency of the Pulse Tube Refrigerator by increasing the phase shift between the pressure and the mass flow. This was done by connecting an orifice and a reservoir to the hot end of the Pulse Tube. This improvement became the standard Pulse Tube Refrigerator configuration and became known as the Orifice Pulse Tube. Single-stage Orifice Pulse Tube (OPT) Refrigerators, have reached 30K while a three-stage OPT has reached 3.6K.

By the late 1980's, Pulse Tube development had begun at many laboratories around the world - including the United States, China, Japan, France, and Germany. A group led by Professor Matsubara of the Nihon University in Japan, developed a moving plug or hot piston Pulse Tube. In this configuration, efficiency was increased by adding a second moving component, but it was not at the cold end.

According to Dr. Kittel, the most important development has been the innovation of the double inlet Pulse Tube by Dr. Zhu, et al. of Xi'an Jiaotong University in China and the subsequent refinement into the multiple bypass Pulse Tube by Dr. Zhou of the Academia Sincia in China. This innovation offered additional efficiency improvements.

How The Orifice Pulse Tube Refrigerator Operates

The operation of the Orifice Pulse Tube is as follows. First, the gas is compressed in the compressor. It then flows through the compressor aftercooler, where heat is rejected to the surroundings or a cooling water loop. Next, the gas flows through the regenerator which is basically an "economizer", or "thermal sponge", conserving cooling from one cycle to the next. The gas then enters the cold-end heat exchanger where heat is added to the gas from the surroundings, thus producing refrigeration. Next, the flow enters the pulse tube, orifice and reservoir. The purpose of these three components is to produce a phase shift between the mass flow and pressure. Without this phase shift, there would be no cooling. In the pulse tube the gas shuttles back and forth between the hot and cold ends rather than circulating continuously around a loop, as in some refrigeration cycles. Heat is lifted against the temperature gradient and rejected at the hot-end heat exchanger, to the surroundings or to a cooling water loop. For a more detailed description of how the pulse tube operates, see Reference 2.

Pulse Tube Cryocooler Detail Design
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More detailed analyses during Phase 2 showed that we could simply the Phase 1 preliminary design. It was shown that a two-stage design could be used rather than our original three-stage configuration. Therefore we did a detailed design and fabrication of a two-stage pulse tube configuration. These detailed analyses were done by Dr. Ray Radebaugh of the National Institute Of Standards and Technology (NIST). Dr. Radebaugh is one of the world's foremost experts on pulse tube refrigeration, having written numerous papers on the subject and designed numerous units. We were fortunate to have had his inputs to our project.

Figure 2 shows this upgraded design. This two-stage cryocooler is designed to provide four Watts of net cooling at 15K, which is 5K degrees below the 20K liquefaction temperature of hydrogen at 101 KPa (14.7 Psia) pressure. Both first and second stages employ tapered tube configurations for improved efficiency. Both tubes are made of .254mm (.010") thick, 304 stainless steel with butt welded seams. The first stage tube is 8cm (3.15") long, the second stage tube is 10 cm (3.9") long. The first and second stage regenerators are also made of .254mm (.010") thick 304-stainless steel with a butt-welded seam. The first and second stage regenerator matrix materials are 400-mesh stainless steel screens and .127mm. (005") diameter lead spheres respectively. This system is also designed with double inlet valves in both first and second stages. The first-stage primary and secondary orifice valves have flow coefficients of 0.17 and 0.19 respectively and the second stage primary and secondary valves have flow coefficients of 0.098 and 0.120 respectively. Our system is also designed with inertance tubes in both first and second stages for improved cooling capability. The peak mass flow rates for the first stage cold end and warm end heat exchangers are 7.48 and 1.95 grams/second, respectively, and 10.7 and 0.95 grams/second for the second stage cold-end and warm-end heat exchangers respectively. The first-stage regenerator inlet peak mass flow rate is 19.6 grams/second.

The first stage cold end heat exchanger is designed to operate at 70K when the second stage cold-end heat exchanger is at 15K and pumping a 4-Watt load. Heat to be removed by the 1st stage warm-end heat exchanger is 45.2 Watts and 16.6 Watts for the second-stage warm-end heat exchanger.

All heat exchanger bodies are made of copper with copper screens brazed inside. The first stage warm-end and cold-end heat exchangers and the second stage warm-end heat exchangers use 100 mesh screens with a wire diameter of .114mm (.0045"). The second stage cold-end heat exchanger uses 150 mesh screens with .066mm (.026") diameter. As seen in Figure 3, this is a linear design with the warm-ends on each end and the cold heat exchangers in the middle.

Our unit uses helium gas as the working fluid in order to get to this 15K minimum temperature. The mean operating pressure is about 1.46 MPa (212 Psi) with a peak-to- peak pressure variation of about 654 KPa (95 Psi) at the first stage regenerator inlet.

The unit operates at 30 Hz which is almost an order of magnitude faster than most pulse tube cryocoolers previously reported in the literature which typically operate at 1 to 5 Hz. This high frequency operation presents challenges in the regenerator design. However this high frequency was chosen to allow the use of a smaller, lighter-weight linear compressor in a future flight weight designs. A low-frequency linear compressor to drive this unit would be huge. Our concept is to use a small compressor but run at high speed to deliver the required mass flow rates and pressures. This can be compared to a race car engine which is run at very high RPM to get a lot of power out of a small engine.

All our warm-end heat exchangers are water-cooled. All these components are brazed together as seen in Figure 2 and mounted inside a small vacuum tank for testing. This task was fabricated by UAH. All four orifice valves are external to the test chamber to allow for orifice flow coefficient adjustment and tuning during testing.

Figure 3 shows these components mounted on one of the test vacuum chamber bulkheads and being insulated with multilayer insulation (MLI) Blankets. The test chamber is evacuated with a turbomolecular and a mechanical vacuum pump during testing. This minimizes parasitic heat leaks to the cold components.

Instrumentation for this unit consists of thermocouples and a silicon diode for temperature measurements and dynamic and static pressure measurements. Data is recorded, reduced and presented using Labview software and National Instruments
P.C.-based data acquisition boards.

Our cryocooler unit is driven by two commercially available air conditioning compressors operated in parallel. Commercially available oil separators, filters and aftercooler are also used.

The pulse tube concept requires an oscillating flow of working fluid, hence the name "pulse". In order to convert the circulating flow from a typical compressor to oscillating flow, we have designed and built a rotary valve. This valve allows flow of helium into and then back out of the cooling components. Valve ports connecting the first stage inlet to the main compressor feed lines are sequentially opened and closed. It is first connected to the high pressure line which causes flow to enter the inlet. Then the high pressure port is closed and the low pressure port is opened causing the flow to be evacuated. Plenum chambers are used on both the high pressure and low pressure sides of the rotary valve.

Figure 4 shows the completely assembled components mounted inside its test rack.

Test Results

This unit has been completely assembled, checked out and leak checked. It is now running and cooling and is being generated. There have been some problems in getting the required 654 KPa (95 Psi) peak-to-peak pressure on the pulses to the first stage heat exchanger due to leaks in the rotary valve and other operating problems. We have also had some problems with getting totally oil-free helium to our cold-end components. In spite of these problems, we have at the time of this writing, attained a temperature of 170K
(-150F).

Figure 5 shows typical measured pressure pulses to the first stage regenerator inlet. Figure 6 shows a typical pressure pulse waveform. Figure 7 shows a typical temperature versus time history for the second stage cold-end heat exchanger with no applied refrigeration load. A resistance heater is to be installed on the second stage heat exchanger to simulate refrigeration load in future tests.

At the time of this writing testing is being continued. Testing consists of a trial and error process in which the four orifice flow coefficients and other operating parameters are systematically varied and the resulting effect on temperature is observed until the minimum value is reached.

Future Work

We plan to continue this testing until our desired temperature and refrigeration load are reached. We also plan to upgrade this linear design to a folded configuration which will place all penetrations on a single bulkhead rather than two bulkheads. This will simplify our interfacing with a test chamber, dewar, propellant tank and/or propellant lines on a vehicle.

We also plan to pursue the development of an efficient, flight weight linear compressor to replace our commercial compressor and rotary valve. The linear compressor concept produces oscillating flow, or pulses, directly to the cooling end components without the need for the rotary valve.

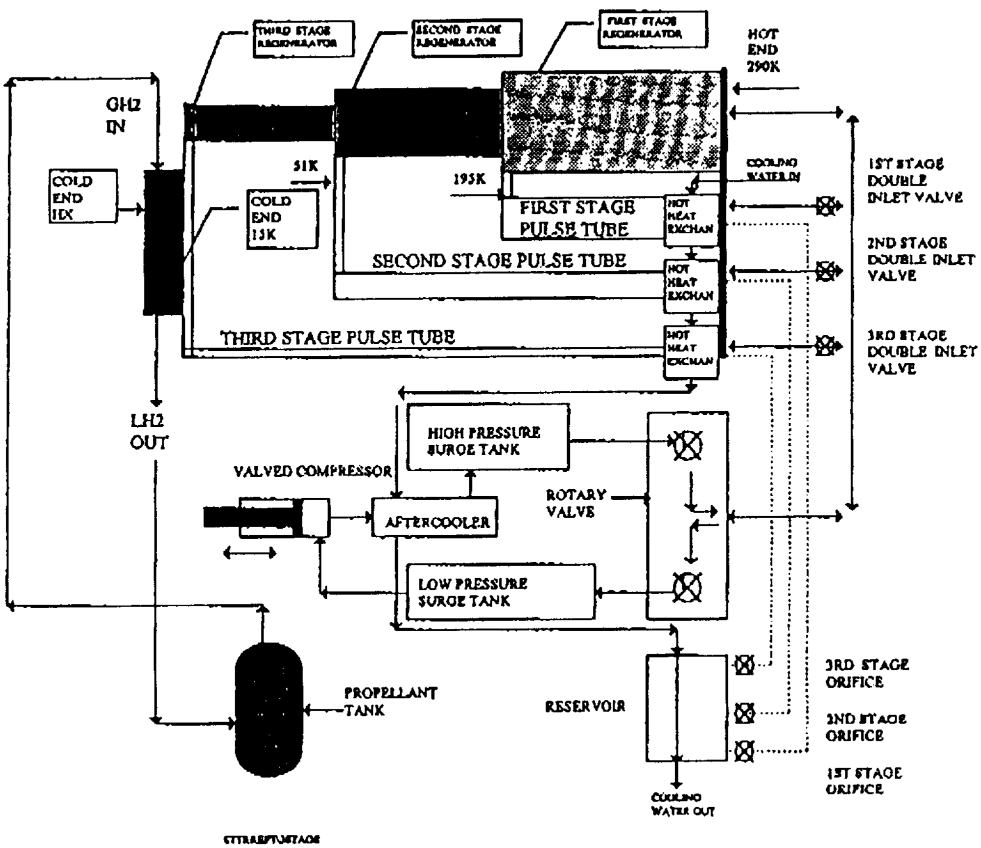

FIGURE 1: SCHEMATIC FOR THREE-STAGE PULSE TUBE BOILOFF RECOVERY SYSTEM PRELIMINARY DESIGN CONFIGURATION DEVELOPED DURING PHASE 1

FIGURE 4: DATCO PULSE TUBE CRYOCOOLER FINAL TEST CONFIGURATION ASSEMBLED INSIDE TEST RACK WITH ROTARY VALVE, VACUUM TEST CHAMBER, PLENUM TANKS, COOLING FAN, TURBOMOLECULAR AND MECHANICAL VACUUM TANKS

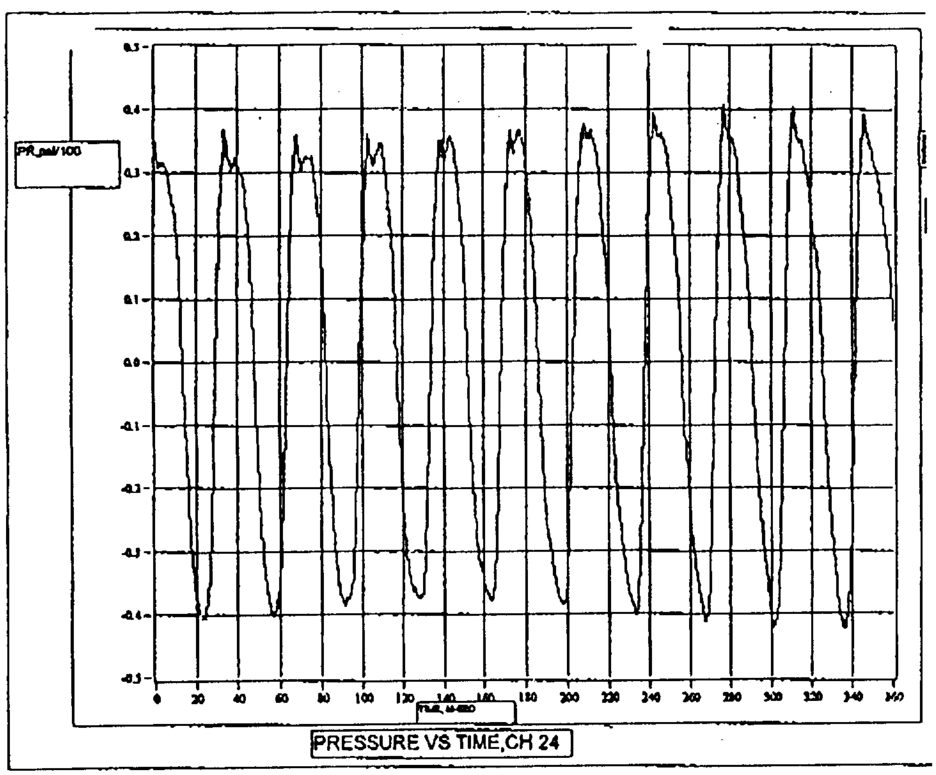

FIGURE 5: PRESSURE PULSES IN HELIUM GAS WORKING FLUID DURING A TYPICAL TEST

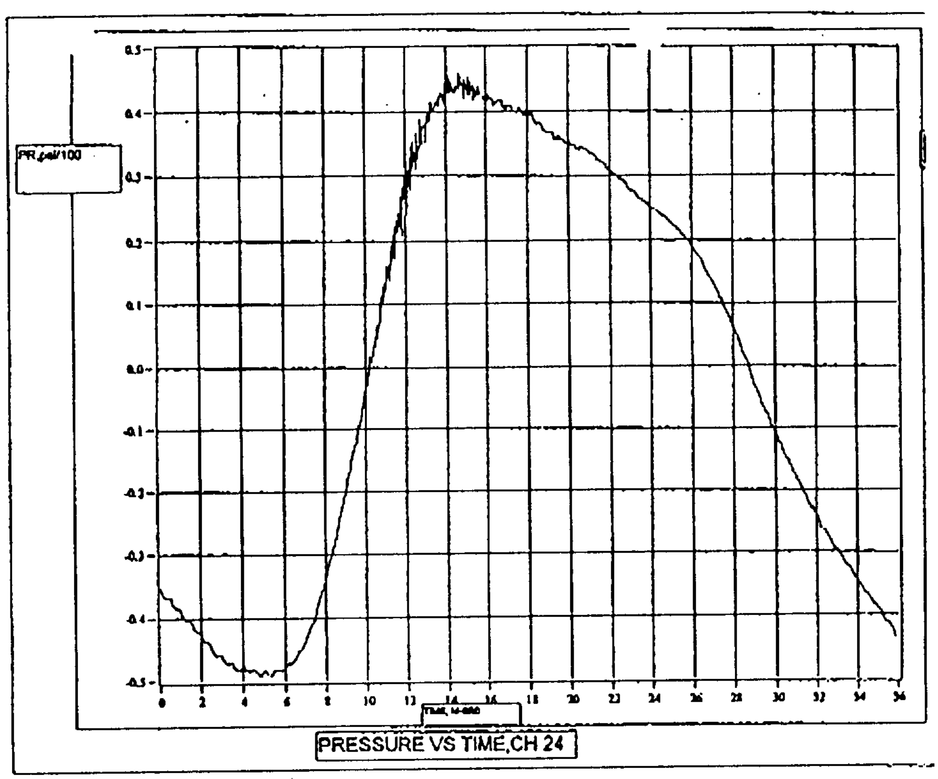

FIGURE 6: HELIUM GAS PRESSURE PULSE WAVEFORM DURING TYPICAL TEST

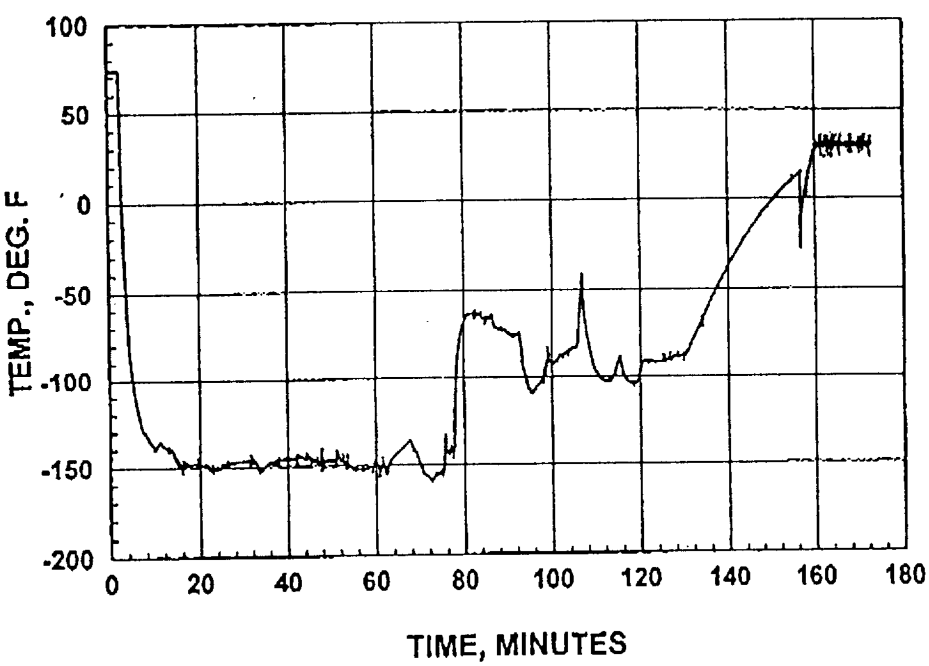

FIGURE 7: TEMPERATURE VERSUS TIME FOR SECOND STAGE COLD END HEAT EXCHANGER BODY DURING TYPICAL TEST