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Hybrid Thermal Control Testing of a Cryogenic Propellant Tank

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Hybrid Thermal Control Testing of a Cryogenic Propellant Tank

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ABSTRACT

This report presents the experimental results of a hybrid thermal control system, one that integrates a passive system (multi-layer insulation) with an active system (a mechanical cyrocooler) applied to cryogenic propellant storage. These experiments were performed on a 1.39 m diameter spherical propellant tank filled with LH2 while installed in an evacuated chamber. The tank heat transfer to the cryocooler was accomplished with a condenser installed in the ullage of the tank and mated to the second stage of the cooler, and by conduction, through copper leaves mated to the first stage of the cooler. The first hybrid system test was performed with both the condenser and the leaves, a configuration that had excess capacity to remove the heat entering the tank; the second test was performed with only the condenser, with a capacity closely matched to the tank heating rate. In both of these tests, the goal of zero boil-off was achieved.

INTRODUCTION

A fact of cryogens is that they will boil-off because of environmental heating. Planning for these losses, larger than necessary tanks and associated storage systems are used. In the case of long term missions in-space, the added mass due to the cryogen boil-off and the oversized tanks and storage systems can make the use of cryogenic propellants prohibitive.

An alternate storage option is to eliminate the boil-off and associated larger-than-necessary storage system by integrating cryocoolers into the tanks. This option has been looked at in previous studies[1,2] but has never been incorporated. Recent advances in cryocooler technology[3,4,5] however, have greatly enhanced this concept. Analysis indicates that added mass for new generation cryocoolers and associated power and heat rejection systems are less than the mass of the boil-off and larger-than-necessary storage tanks if the mission storage time is greater than roughly 45 days.[6] As the storage duration increases, the mass savings increases. The limiting factor is the cryocooler life, which the current maximum is ~10 years.

While this zero boil-off (ZBO) approach looked promising based on the analysis performed, the concept had not been tested on a cryogenic storage tank that holds propellant of the quantity approximating that of an upper stage of a launch vehicle. A test on such a tank would help determine the feasibility of the concept.

EXPERIMENTATION

The design goal of the experimentation was a simple and inexpensive integration of a cryocooler with an existing insulated propellant tank.

Facility

NASA Glenn Research Center's Supplemental Multi-layer Insulation Research Facility (SMIRF), documented in Ref. 7, is a multi-purpose test bed designed to evaluate the performance of thermal protection systems. One modification to the vacuum chamber was that the cold GN_2 shroud, used to provide a controlled thermal environment to the test article, was removed in order to accommodate the test tank

Tank

The liquid hydrogen test tank used is briefly described below and in detail in Ref. 8. This 2219 aluminum tank is spherical, 1.39 meters diameter and 1.42 cubic meters in volume. This tank has a 0.3 m diameter. opening and cover to allow access to the tank interior. The tank was suspended from a tubular, stainless-steel support ring by six stainless-steel wire support struts. The tubular support ring was, in turn, suspended from the lid of the vacuum chamber by three support cables. The tank (insulated) is shown in Fig. 1.

Insulation

The insulation installed on the test tank was two multi-layer insulation (MLI) blankets to cover and thermally protect the entire tank surface. This concept is also discussed in Ref. 8. Each blanket consisted of 15 double aluminized mylar (DAM) radiation shields alternatively spaced with double silk net spacers. This results in a total of 17 layers of mylar radiation shields, or MLI, for each blanket, for a total

Figure 1: MLI insulated tank supported by tubular support ring and hung from vacuum chamber lid. Cryocooler is located at top of tank. Flexible helium lines, which go to the scroll compressor located outside the vacuum chamber, are shown.

of 34 layers of MLI. The blankets are held together with Nylon fasteners and reinforced Mylar cover sheets. Nylon button-pin studs epoxied to the tank wall supported the blankets. Approximately 2/3rds of the tank was covered with this insulation, which was fabricated and tested in 1977 and was in storage for the last 15 years at Plum Brook Station. The missing third of the insulation, which consisted of one of the six gore sections and the top and bottom dome sections that mate to the tank, were constructed at GRC using similar materials and were fitted to the tank. Also, constructed were fitted MLI and cryoglass blankets that insulated the tank supply, vent, and drain lines as well as the cryocooler. Aluminized mylar tape was used to attach the new MLI on the tank and also to attach the MLI/cryoglass blankets around the lines and the cryocooler. The insulated tank, attached to the vacuum tank lid is shown in Fig. 1.

Cryocooler

The cryocooler was borrowed from our colleagues at NIST. It was a two-stage Gifford/McMahon cycle, commercial cryocooler. The cryocooler capacity rating of the second stage was 17.5 W at 18K while, simultaneously, providing 20 W of cooling at 35K at the first stage. The cryocooler is shown in Fig. 2. The cryocooler also included a separate helium scroll compressor (not shown), which was located outside the vacuum chamber.

Heat Exchangers

The cryocooler second stage coldhead was mated to a condenser heat exchanger, shown in Fig. 3. Indium foil was used to increase the thermal contact area between the mating surfaces. Copper No. 101 was used for the flange/rod because of its very high thermal conductivity at cryogenic temperatures. Brazed to the rod was a stainless steel sleeve, which was welded to a stainless steel flange that was bolted to the tank access cover. Stainless steel was used because of its low thermal conductivity, less than 1/1000th that of copper, and effectively stopped the heat transfer from the access cover to the cryocooler. Bolted to the bottom end of the rod was our 248 cm^2 Copper 101 condenser, which extended 12.7 cm below the top of the tank, at the 97 percent full level. Note that none of the tests had liquid levels over 95.5 percent full. This condenser was designed to remove the predicted tank environmental heating by

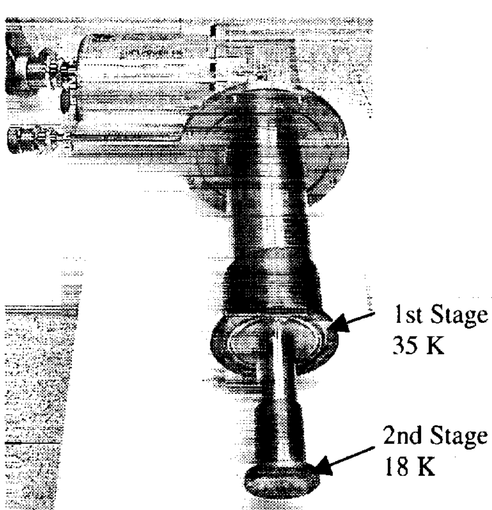

Figure 2: Two stage Gifford-McHahon cycle cryocooler used in testing. Cryocooler capacity was 17.W at 18 K and 20 W at 35 K.

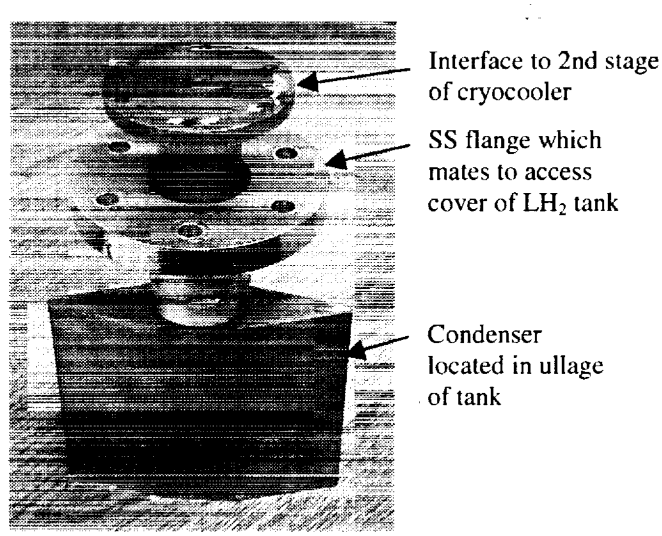

Figure 3: Second stage condenser heat exchanger.

condensing the hydrogen vapor in the ullage of the tank. Natural fluid convections of the LH2 moved the heat from the tank wall to the vapor.

In addition to the condenser, other heat exchangers were used in one of the tests to take advantage of the heat removal capacity of the cryocooler's first stage. These heat exchangers were made from 0.16 cm thick Copper No. 101 sheet. Attached to the cryocooler first stage was a hexagon plate with 19.2 cm wide sides. Attached to this plate were 2.5 cm wide straps used to conduct heat from the vent and fill lines as well as six quadrilateral plates or leaves that were inserted between the two 17 layer MLI blankets covering the tank. These leaves were trimmed to fit under the six gore sections of the insulation blankets. The purpose of the leaves was to intercept the heat entering the tank and conduct that heat to the cryocooler's first stage. The leaves were 1 m long and extended near the mid-section of the tank. The leaves plus the hexagon plate covered 2.5 m^2, or ~40 percent of the tank surface. Figure 4, shown later in the paper, is a simplified schematic that shows how the leaves fit in with cryocooler, insulation and tank.

Instrumentation

Temperatures on the tank wall, the cryocooler, and the condenser were taken with silicon diodes. The temperature sensors on the copper leaves, tank piping, MLI, and vacuum chamber were chromel-constantan thermocouples. Liquid level in the tank was determined with silicon diode point sensors and a delta pressure transducer located at the tank drain.

Capacitance-type pressure transducers were used to measure tank and facility pressures. Vacuum instrumentation with a range between atmospheric pressure and 10^{-5} torr was used. Tank boil-off was measured by one of the four calibrated mass flowmeters. These meters were calibrated with air and had a full scale of 0 to 3200, 0-400, 0-8, and 0-1 standard ft^3/hr.

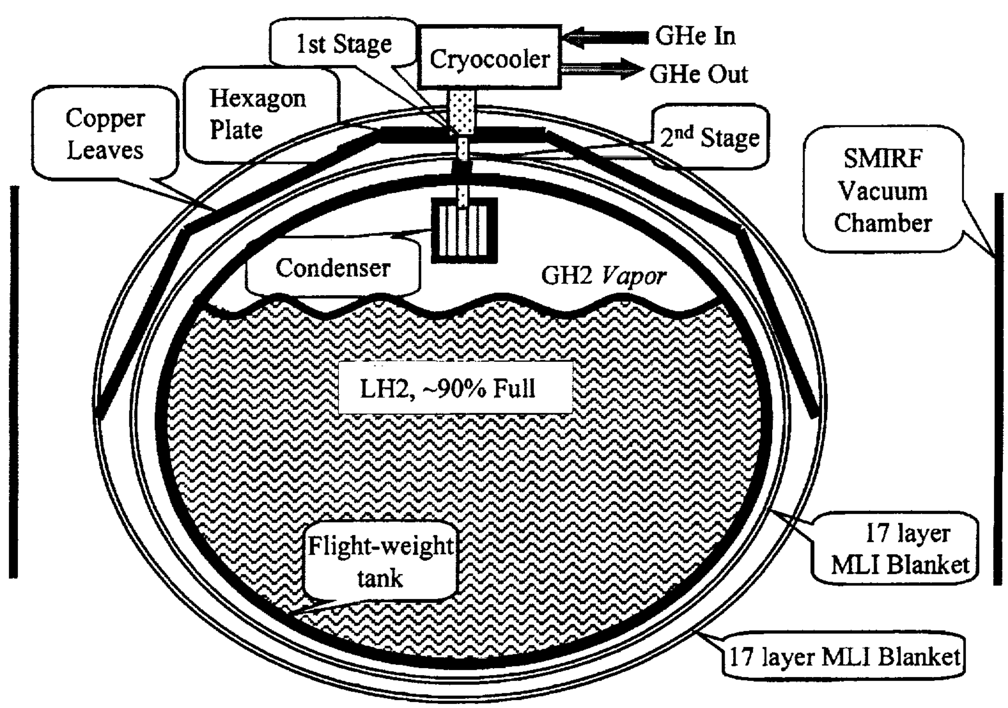

Figure 4: Schematic showing the tank, cryocooler, heat exchangers and insulation installed for the Cooler On test.

Limitations

Despite the fact that the test was performed in a vacuum chamber, space environment conditions were not simulated. The cryocooler used was not a flight-type cryocooler, thus the compressor was not integral to the cooler. Because the compressor would add heat, our test heating rates are lower than predicted for a flight cryocooler. In addition, the heat exchanger used in the tank was located in the ullage and liquid condensed off of it. This configuration would not be possible in-space.

A few minor problems were noticed during testing and afterwards. There was a small hydrogen leak in the test tank access cover which permeated the MLI, reducing its performance and causing a higher than predicted heat leak. To minimize this leak, we conducted the test at lower tank pressures than preferred. Saturation temperatures at these reduced pressures are lower, correspondingly, the cryocooler had to operate at lower temperatures where it has a lower performance. Post test, the vacuum chamber mass spectrometer measured 3.6×10^{-4} torr partial pressure of hydrogen gas. In addition, the heat entering the cryocooler second stage was not accurately measured and is not reported.

RESULTS

Baseline Conditions

The tank liquid level for all tests was ~90 percent full. The vacuum chamber pressure fluctuated for each test, with the hardest vacuum being 3×10^{-5} torr and minimum vacuum 2×10^{-4} torr. The vacuum wall temperature was not controlled and was typically about 295K.

No Cooler Test

This was a baseline test to characterize steady state heating rate of the insulated tank in the vacuum chamber. The cryocooler and associated hardware were not installed. The measured steady-state boiloff rate was ~0.12 kg/hr, equating to a heating rate of ~14.5 W. The vacuum chamber pressure was 2.8×10^{-5} torr.

Cooler On Test

This test incorporated both the first and second cryocooler stages with the test tank. The heat transfer to the first stage was via solid conduction through the copper leaves. Also, straps from this stage were used to remove heat from the tank lines. The second stage heat transfer was by condensation of the hydrogen vapor in the ullage. A schematic of this test configuration is shown in Fig. 4.

After the tank was filled with liquid hydrogen, the cryocooler was turned on. During the first few hours, the chamber vacuum gage measured a relatively soft vacuum of ~1.5×10^{-4} torr causing the boil-off rate to be about 0.3 kg/hr (35 W). Even so, the cryocooler and heat exchanger temperatures as well as the boil-off rate quickly decreased. Within 8 hr, the boil-off rate decreased to zero and the vent valve was then closed. The first stage temperature, which had a relatively low load, was at 35K and would remain at about that temperature throughout the test. The temperature at the end of the warmest copper leaf was at 43K and also remained close to that temperature through the test. At these temperatures, an estimate of the copper thermal conductivity was made and the heat transfer through the leaves was calculated to be 9 W. Besides that heat removal, the copper straps to the plumbing removed ~60 percent of the heat that had entered the tank during the No Cooler test. The second stage of the cryocooler was also effective.

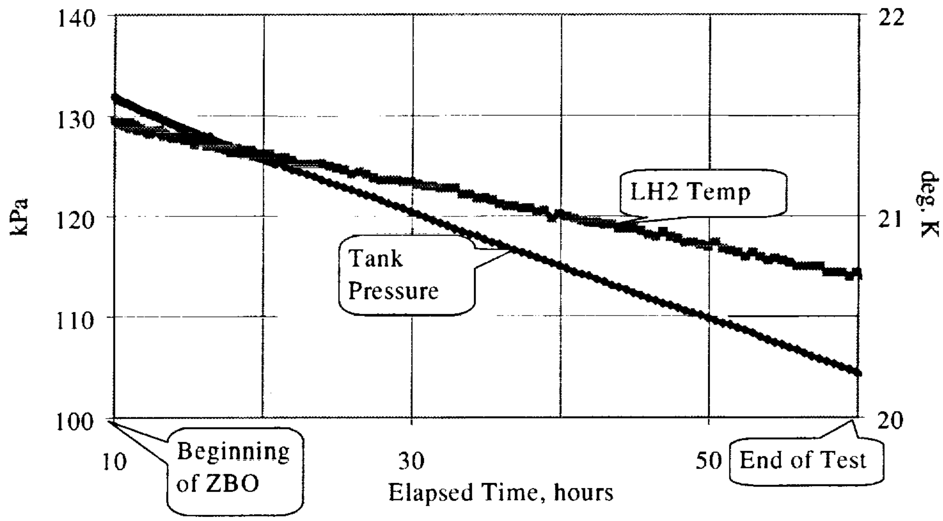

Figure 5: Cooler On test plot of tank temperature and pressure decay during testing.

After the vent valve was closed and over the next 54 hr, the condenser temperature decreased at a steady rate of 0.02 K/hr. In addition, the average LH2 tank temperature steadily decreased by 0.017 K/hr and the tank pressure dropped at a steady rate of 0.55 kPa/hr. The propellant liquid level did not change noticeably during the test. The testing was terminated when the tank pressure dropped to 103 kPa, which was 4.4 kPa above atmospheric pressure. This was to prevent air from being accidentally drawn into the tank through the vent or fill lines. The temperature and pressure data are shown in Fig. 5.

Cooler Off Test

Upon completion of the Cooler On test, the cryocooler was turned off to simulate a failed cryocooler configuration. The goal of this test was to reach steady state and compare the measured heating rate with the results from the No Cooler test. For the first 2.5 hr the boil-off rate was zero. After that, the boil-off rate quickly climbed up to 0.12 kg/hr (14.5 W), as it was for the No Cooler test, and then slightly increased for the duration of the test, ending at 0.14 kg/hr (17 W). Note that this flowmeter data was considered accurate, as the tank pressure and chamber vacuum level were consistent during this test. The condenser temperatures also increased, apparently due to the heat from the copper leaves conducting through the cooler and into the condenser. After 77 hr of testing, the test was stopped. Steady state was never reached.

Second Stage Only Test

After the Cooler Off test was completed, the tank was removed from the vacuum chamber. The copper leaves and tank line straps were removed so that the cryocooler's first stage would have zero load. The tank would thus be cooled by only the cryocooler's second stage. As stated earlier, this stage capacity was 17.5 W at 18K, a much closer match to the insulated tank's heating rate of 14.5 W determined in the No Cooler test. This test configuration resembled a one-stage cryocooler integration. Single stage cryocoolers would likely be used to cool oxygen and methane propellants in space.

During the first 55 hr of testing, the boiloff varied as did the vacuum pressure, which was not nearly as hard for this test. The average boil-off rate was 0.005 kg/hr (0.6 W), which meant that the cryocooler removed 96 percent of the heating rate experienced in the No Cooler test. At the end of 55 hr of testing, the chamber vacuum pressure finally improved somewhat, from 1×10^{-4} torr to ~8.5×10^{-5} torr, we achieved

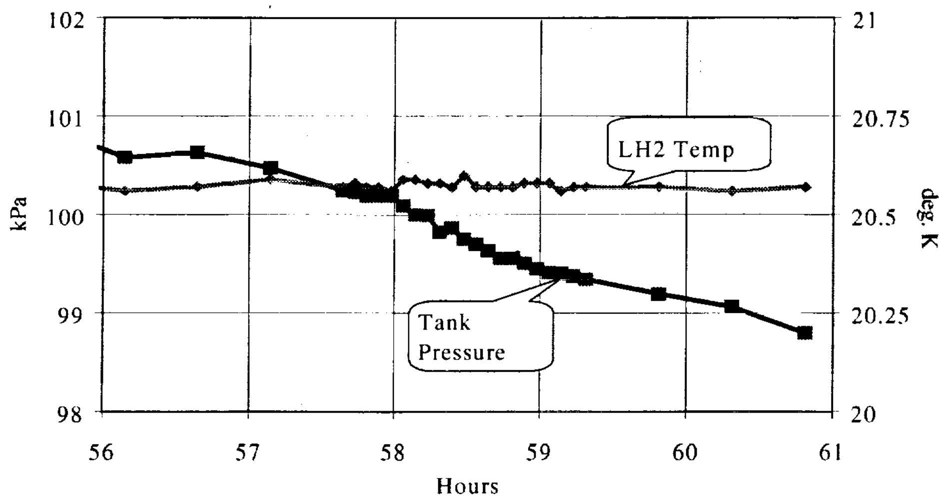

Figure 6: Second Stage Only test data showing tank pressure and LH_2 temperature versus time. This portion of the data was at the end of the test, when the boil-off rate was zero.

zero boiloff (0.00 kg/hr) and closed the vent valve. At this time, the tank pressure started decreasing very slightly. The pressure decrease rate was ~0.35 kPa/hr, while the condenser temperature remained steady. That data is shown in Fig. 6. As in the Cooler On test, testing was terminated to prevent the tank pressure from dropping below atmospheric pressure. In this case, the test was stopped when the tank was 1 kPa above atmospheric pressure.

DISCUSSION OF RESULTS

The condenser heat exchanger, attached to the second stage of the cryocooler and installed in the ullage of the tank, worked effectively. When combined with the copper leaves in the Cooler On test, it overmatched the heat entering the tank and caused the pressure and temperatures to steadily drop. In the Second Stage Only test, it closely matched the heat entering the tank and, at the end of the test, removed the heat and caused the pressure to decay, although at a lower rate (0.35 kPa/hr vs. 0.55 kPa/hr). Also, integration of the first stage was effectively accomplished with solid conduction. Despite the fact that the cooper leaves were 1 m long, the ends were only 8K warmer than the first stage cryocooler temperature. Also, the heat removed by the leaves was more than half the heat that entered the tank during the No Cooler test. Simple 2.5 cm wide straps were affixed to the plumbing and removed a majority of that heat.

Cryocooler operation, while installed on a simply supported tank, did not cause noticeable tank movement or fluid sloshing. The cryocooler with its first stage heat exchangers damped out transient facility vacuum fluctuations and reduced the operating vacuum. With the cryocooler off, slightly higher heating rates occurred and the heat exchanger temperatures steadily increased.

In the Cooler On and Second Stage Only tests, the condenser lowered the pressure because it removed the heat that entered through the tank wall and had, by natural convections, made its way through the liquid hydrogen and into the ullage. Without these convections, which are absent in low-g, the fluid behavior and heat transfer will be different. Also, in low-g, the location of the ullage would be unknown. Thus, these factors require a different heat exchanger design that would either have much greater surface area or would include a propellant mixing device, creating forced convective heat transfer.

While the first stage heat exchangers removed a majority of the tank heating during the Cooler On test, the benefit if applied to a cryogenic propellant tank in-space is undetermined. This is because the designs for a flight-type first stage heat exchanger has not been done. Lightweight, rigid designs that include fluid convection in addition to solid conduction need to be investigated.

Also needing study are two thermal issues of in-space operations. The first issue is the heat conduction from the "off" cryocooler to the tank propellants. If the cryocooler operation in-space necessitates an on-off duty cycle due to power supply availability or low pressure limitations on the storage tankage, then a thermal shunt or diode incorporated into the cryocooler-heat exchanger assembly could be necessary to prevent this undesirable heat flux. Alternatively, this problem might be mitigated if the cryocooler is sized for a greater capacity than the heat that enters the tank. This overcapacity, the second thermal issue needing study, was shown in this test to reduce the tank pressure and temperatures.

CONCLUSIONS

This zero boil-off demonstration of cryogenic propellant storage has provided an early proof of concept that, combined with advances in cryocooler technology, offer exciting possibilities for long-term cryogenic fluid storage applications in-space. The potential mass reductions for zero boil-off cryogenic storage applied to long-term space missions are substantial. The possibility to treat cryogens like space storable propellants reduces safety concerns for in-space operations, such as rendezvous and docking. Delays in those operations or other mission delays could now be tolerated because cryogen boil-off margins would not be exceeded. In addition, there is the potential for the cryocooler to damp out unexpected transient heating. Because of these benefits, zero boil-off cryogen storage offers more opportunities to utilize high-performing cryogenic propellants for long duration space missions.

While the demonstration was successful and the benefits are substantial, this effort only provides the foundation for remaining work. Analytical modeling and testing with cryocoolers and heat exchanger configurations designed for a low-g space environment are recommended. In addition, thermal effects from in-space power supply and heat rejection systems need to be included.

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