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NASA Technical Memorandum 104516

Autogenous Pressurization of Cryogenic Vessels Using Submerged Vapor Injection

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Prepared for the
Cryogenic Engineering Conference and International Cryogenic
Materials Conference
cosponsored by the University of Alabama in Huntsville, and NASA
Marshall Space Flight Center
Huntsville, Alabama, June 11–14, 1991

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ABSTRACT

Experimental results are reported for submerged injection pressurization and expulsion tests of a 4.89 m^3 liquid hydrogen tank. The pressurant injector was positioned near the bottom of the test vessel to simulate liquid engulfment of the pressurant gas inlet; a condition that may occur in low-gravity conditions. Results indicate a substantial reduction in pressurization efficiency, with pressurant gas requirements approximately five times greater than ideal amounts. Consequently, submerged vapor injection should be avoided as a low-gravity autogenous pressurization method whenever possible. The work presented herein validates that pressurant requirements are accurately predicted by a homogeneous thermodynamic model when the submerged injection technique is employed.

INTRODUCTION

Future space flight will require the transfer of cryogenic liquids under low-gravity (low-g) conditions for use in chemical and nuclear propulsion, life support, and thermal control. Conventional pressurization of settled cryogenic tanks utilizes hardware that diffuses the pressurant flow within the ullage in a manner which minimizes impingement of the pressurant on the liquid-vapor interface or tank walls. In the low-g environment, the distribution of liquid and vapor phases may not be well defined and it becomes difficult to ensure that the pressurant is injected directly into the tank ullage. It is possible that direct injection of the pressurant into the bulk liquid will occur during liquid reorientation, sloshing, or even static conditions. For all of these conditions, the pressurant-liquid interaction may lead to either evaporation of the liquid or condensation of the pressurant gas, depending upon the complex heat and mass transfer processes involved.

Previous studies of submerged gas injection include an experimental investigation of helium gas injection into liquid hydrogen (LH_2) by Johnson[1]. For this situation, interaction of the non-condensible helium gas and LH_2 leads to vaporization of a portion of the LH_2, which in some cases can reduce the required amount of pressurant gas. However, when a condensible pressurant is used for tank pressurization, the potential for pressurant condensation (collapse) is high. This is noted in experiments performed by DeWitt and McIntire[2] with liquid methane. When the pressurant was directly injected into the ullage, liquid sloshing increased the pressurant requirement for a condensible pressurant (methane) and decreased the pressurant requirement for non-condensible pressurants (helium and hydrogen). Finally, the interaction of the pressurant with the liquid frequently results in undesirable liquid heating.

The results reported herein are concerned with autogenous tank pressurization and expulsion of liquid hydrogen (LH_2) in a normal-g test environment. The gaseous hydrogen (GH_2) pressurant was injected into the tank well below the liquid level in order to simulate the increased interaction with the liquid cryogen that may occur in low-g applications. Data was obtained in a LH_2 tank having characteristics similar to propellent tanks of future spacecraft.

EXPERIMENTAL APPARATUS

The test facility (see Fig. 1) consists of a 7.6 m diameter vacuum chamber containing a 4.0 m diameter cylindrical shroud that in turn encloses the LH_2 test tank. The shroud was maintained at a constant temperature of 295 K by electrical resistance heating to obtain a constant heat input to the test article. This heat input includes penetration heat leaks through the insulation system and is determined from boil-off data[3]. Vacuum chamber pressure during the test series was on the order of 10^{-4} kPa. The test tank is suspended by fiberglass composite struts and all instrumentation lines and flow lines other than the pressurant line are routed though a LH_2 cold guard (not shown) to minimize conductive heat transfer to the test article. Pressurant (normal-GH_2) is supplied from outside high pressure storage bottles. A steam heat exchanger is used to heat the pressurant to 330 K, or the heat exchanger may be bypassed to provide ambient temperature gas at 275 K. Pressurant flow rate is calculated from pressure drop measurements across a square edged orifice (1.15 cm in diameter) placed in the pressurant line. The orifice is instrumented with high and low range differential pressure transducers as well as upstream pressure and temperature transducers. A tank bypass line allows the pressurant line to be thermally conditioned prior to a test when heated pressurant is required. Expelled LH_2 from the test article is returned via a transfer line to an outside storage dewar. The transfer line is instrumented with a venturi flow meter to measure expulsion flow rate. Pressure and temperature transducers, as indicated in Fig. 1, are located on the outflow line to measure the thermodynamic state of flow out of the test vessel.

The LH_2 test tank is approximately an ellipsoidal volume of revolution having a major-to-minor axis ratio of 1.2, a major diameter of 2.2 m, a volume of 4.89 m^3, and a mass of 149 kg. It is constructed of 2219 aluminum. Most of the wall is 2.08 mm thick except for the thick bolted flange and lid at the top, thickened lands for support lugs, and a thickened equatorial region. It is insulated with 34 layers of double aluminized Mylar separated by silk netting. The tank insulation, size, and lightweight construction (other than the lid) are representative of a cryogenic storage tank that could be used in future spacecraft. Pressurant is fed into the test article through a "j-tube" which directs the gas into the LH_2. The j-tube's inside diameter is 1.2 cm and it has a 0.95 cm full cone spray nozzle attached at the outlet.

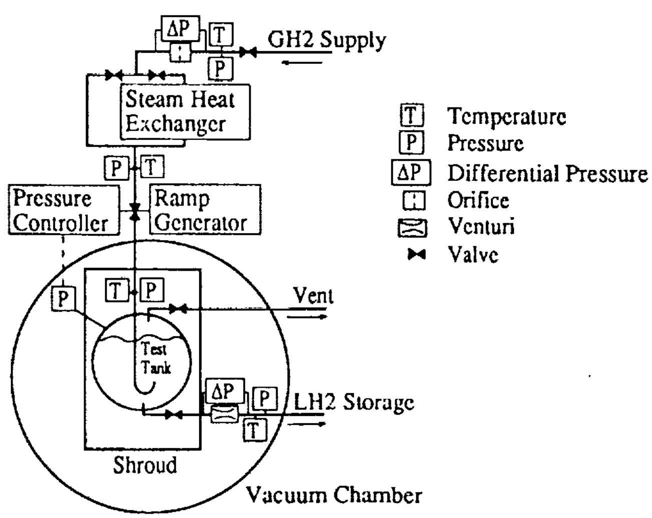

Fig. 1. Schematic diagram of test facility.

The total length of the j-tube from the attachment point at the tank lid to the outlet is approximately 2 m.

Figure 2 is a schematic diagram that indicates the location of the j-tube and various temperature sensors. The j–tube exit is approximately 25 cm from the tank bottom. Liquid fill level in the tank is measured by a capacitance probe, and liquid-vapor temperatures are measured by silicon diode transducers. The external wall temperature distribution is measured by a number of wall-mounted silicon diode transducers. Tank pressure is measured by pressure transducers in direct communication with the tank ullage. Liquid-vapor temperature measurements inside the tank are accurate to ± 0.3 K, while wall temperatures are accurate to ± 0.6 K. An in situ calibration increases the accuracy of liquid-vapor temperature measurements to ± 0.1 K by adjusting the individual sensor readings to known saturation conditions. Tank pressure measurements are accurate to ± 0.01 kPa. Capacitance probe readings are accurate to ± 1.9 cm, translating to a maximum error of ± 1.5 percent fill at the 50 percent fill level (by volume). Pressurant gas flow rate measurements have an estimated accuracy of ± 0.18 and ± 0.40 kg/hr for the large orifice using the low and high range differential pressure transducers, respectively. Liquid outflow could not be properly measured due to cavitation in the venturi; instead it was determined from liquid level change in the tank. Data is sampled by an automated data acquisition system at selected intervals (15 to 60 sec) throughout the duration of the experiments.

TEST PROCEDURE

The tank is prepared for a test by filling to the desired fill level while the tank pressure is maintained at least 15 kPa above atmospheric pressure. If heated pressurant is used, the tank bypass line is opened and the pressurant line is thermally conditioned until the temperature transducer near the j-tube inlet indicates the desired gas temperature. Next the tank is vented to the atmosphere to induce substantial bulk boiling of the tank liquid which produces nearly isothermal conditions within the tank. A venting period of approximately 15 min is necessary to obtain saturated liquid temperatures throughout the tank. A test is initiated by closing the vent line valves and opening the pressurant line valves. In the first portion of a test, a preset tank pressure ramp rate is maintained by controlling the pressurant flow control valve with an automatic ramp generator. After the maximum tank pressure is attained, a 2 min hold period follows during which control of the pressurant flow valve is switched to an automatic pressure controller. The tank pressure is kept constant by addition of pressurant during liquid expulsion. Liquid outflow is regulated by remote operation of flow control valves in the outflow line. Expulsion is stopped at a nominal 5 percent fill level. Data is automatically recorded at regular intervals throughout the duration of the test.

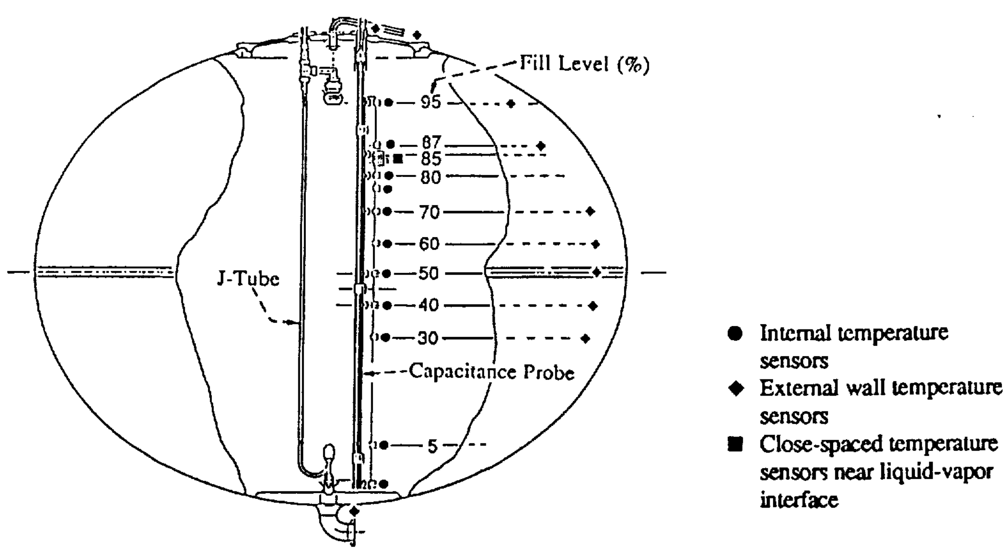

Fig. 2. Schematic diagram of test tank and instrumentation.

DATA ANALYSIS

Mass and energy balances are performed by dividing the tank interior volume and wall into horizontal segments corresponding to the internal and wall-mounted temperature sensors. At any given time, segment boundaries are adjusted as necessary to accommodate the variable location of the liquid-vapor interface. The amount of vapor condensation or liquid evaporation is determined from a mass balance performed on the ullage volume:

$$\pm M_{t,i\rightarrow f} = M_{u,f} - M_{u,i} - M_{G,i\rightarrow f} \tag{1}$$

A positive value indicates net evaporation. Initial and final ullage masses are obtained by numerical integration of the density profiles where $\rho=f(T,P)$:

$$M_{u,i} = \int_{V_{u,i}} \rho dV \cong \sum_{n=1}^{N_i} \rho_{n,i} V_{u_{i,n}} \tag{2}$$

$$M_{u,f} = \int_{V_{u,f}} \rho dV \cong \sum_{n=1}^{N_f} \rho_{n,f} V_{u_{f,n}} \tag{3}$$

The mass of injected pressurant is calculated by numerical integration of instantaneous flow rate measurements obtained from the calibrated orifice in the pressurant line.

A thermodynamic analysis was performed by applying the first law to the wall and the liquid and vapor contents of the tank. No external work is performed and if kinetic and potential energy terms are neglected, the tank energy balance is:

$$\Delta U_{T,i\rightarrow f} = \int_{t_i}^{t_f} \dot{M}_G h_G dt + \int_{t_i}^{t_f} \dot{Q} dt - \int_{t_i}^{t_f} \dot{M}_L h_L dt \tag{4}$$

Energy added to the tank consists of energy input by the pressurant plus heat leak from the environment minus energy of the liquid outflow from the tank. The various integrals on the right hand side were numerically calculated. The enthalpy of the pressurant was obtained using the measured gas temperature at the tank inlet. Enthalpy of the liquid outflow was based on average values of measured temperature near the tank outlet at the constant expulsion pressure. An average heat leak rate of 28 W times the test duration gives the total heat leak. The energy input results in thermal heating of the vapor, liquid, and tank wall:

$$\Delta U_{T,i\rightarrow f} = \Delta U_{u,i\rightarrow f} + \Delta U_{L,i\rightarrow f} + \Delta U_{w,i\rightarrow f} \tag{5}$$

The quantities on the right hand side of Eq. 5 were calculated as follows:

$$\Delta U_{u,i\rightarrow f} \cong \sum_{n=1}^{N_f} \rho_u \left(h_u - \frac{P}{\rho_u}\right) V_{u_{f,n}} - \sum_{n=1}^{N_i} \rho_u \left(h_u - \frac{P}{\rho_u}\right) V_{u_{i,n}} \tag{6}$$

$$\Delta U_{L,i\rightarrow f} \cong \sum_{n=1}^{N_f} \rho_L \left(h_L - \frac{P}{\rho_L}\right) V_{L_{f,n}} - \sum_{n=1}^{N_i} \rho_L \left(h_L - \frac{P}{\rho_L}\right) V_{L_{i,n}} - \int_{t_i}^{t_f} \dot{M}_L h_L dt \tag{7}$$

$$\Delta U_{w,i\rightarrow f} \cong \sum_{n=1}^{N_w} M_w \int_{T_{i,n}}^{T_{f,n}} C_w dT \tag{8}$$

where ρ and h are functions of temperature and pressure and C_w is the specific heat of the tank wall material. Dropping the i,f subscripts, Eq. 5 may be rearranged as:

$$1 = \frac{\Delta U_L}{\Delta U_T} + \frac{\Delta U_u}{\Delta U_T} + \frac{\Delta U_w}{\Delta U_T} \tag{9}$$

The overall energy balance is then utilized to analyze the resulting distribution of the added energy to the liquid, vapor, and tank wall regions.

Experimentally determined pressurant requirements may be compared to two simple analytical models. The first model gives the so called "worst case" pressurant requirement. It assumes that the pressurant attains thermal equilibrium with the tank contents, i.e. a homogeneous thermodynamic state. Under cryogenic conditions, the energy increase of the tank wall may be neglected. Solutions for the thermal equilibrium prediction are obtained by combining the mass and energy balances applied to the tank contents:

$$M_G = \frac{M_f U_f - M_i U_i + (M_i - M_f) h_L}{h_G - h_L} \tag{10}$$

The second model assumes no energy or mass transfer occurs between the pressurant and the tank or the initial tank contents. This model provides the so called "ideal" pressurant requirements. It is formulated assuming that the initial ullage mass is isentropically compressed during the ramp process. The remaining portion of the initial ullage volume plus the volume vacated by the liquid during expulsion is assumed to be occupied by added pressurant which undergoes an isentropic expansion from its supply condition. The ideal mass requirement for specified initial and final fill levels is:

$$M_G = \rho_{G,f} V_t \left[1 - F_f - (1 - F_i) \frac{\rho_{u,i}}{\rho_{u,f}} \right] \quad \text{where } \rho_{G,f} = f(P_f, s_G) \tag{11}$$

TEST RESULTS

A series of experiments were performed in which the effects of ramp duration, expulsion time, and pressurant gas temperature were investigated. A test summary is provided in Table 1. All tests began with the tank vented to atmospheric pressure (99 to 107 kPa) followed by a ramp pressurization to approximately 275 kPa. Initial liquid fill levels were 84 percent. Liquid expulsions were limited to 4 tests where the final tank fill level was 7 percent (5 percent for Test No. 6).

The tank pressure history for Test No. 5R is shown in Fig. 3 which consisted of a ramp pressurization process followed by a 2 min hold period and then a constant pressure expulsion of the liquid. During the ramp process, the tank pressure was increased from 99 to 275 kPa. Next was the hold period, followed by liquid expulsion from the 84 to 7 percent fill level. The expulsion occurred at constant tank pressure except for a small pressure drop (13 kPa) experienced when the liquid outflow valve is first opened. Pressure histories for the other ramp and expulsion tests listed in Table 1 are similar to that shown in Fig. 3 except for differences due to the parametric variation of the ramp and expulsion rates.

Table 1. Test Summary

Test No.	Pressurant Temperature (K)	Ramp Duration (min)	Expulsion Duration (min)	Pressurant Consumption Ramp (kg)	Pressurant Consumption Expulsion (kg)	Pressurant Consumption Total (kg)	Evaporated Mass Ramp (kg)	Evaporated Mass Expulsion (kg)
1	275	27	none	3.0	n/a	3.0	-2.4	n/a
2	275	20	none	3.0	n/a	3.0	-2.2	n/a
3	275	18	none	2.9	n/a	2.9	-2.1	n/a
5	275	21	15	3.0	1.6	4.7	-2.3	7.5
5R	275	13	15	3.1	1.7	4.8	-2.0	7.6
6	275	13	25	3.0	1.6	4.6	-1.9	6.2
9	330	12	15	2.4	1.4	3.7	-1.3	7.2

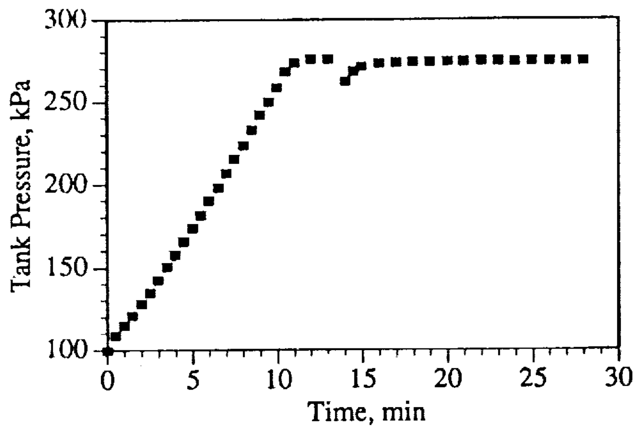

Fig. 3. Tank pressure versus time for Test No. 5R.
Ramp pressurization followed by liquid expulsion.

Representative internal tank temperatures (measured near the vertical tank axis) are shown in Fig. 4 for Test No. 5R. Three of the measurement locations were initially below the liquid level while the remaining location was at all times in the vapor region. All of the liquid temperatures are in close agreement during the ramp process; increasing with time. At the end of the ramp period the liquid temperatures are approximately 0.3 to 0.4 K less than the saturation temperature of LH_2 at 275 kPa. The uppermost temperature was slightly above the saturation temperature during the initial portion of ramp process and then rapidly increased thereafter except for a brief temperature drop attributed to the sudden pressure drop at the start of the expulsion period. Within a few minutes after outflow began, the liquid temperatures reached the saturation temperature corresponding to the expulsion pressure. Two of the temperature sensors became exposed to the ullage during the expulsion and exhibited a steady temperature rise for the remainder of the test as the surrounding vapor becomes superheated. In all of the tests, substantial liquid heating occurred due to the submerged injection of the pressurant gas.

Ramp duration, ranging from 12 to 27 min, did not have a significant effect on the pressurant energy input for the ramp pressurization tests. As shown in Table 1, for gas temperatures of 275 K, the amount of injected pressurant was approximately 3 kg, while at the hotter gas temperature of 330 K, the pressurant mass was 2.4 kg. Total energy input for all ramp tests was 11,500 kJ ± 4 percent.

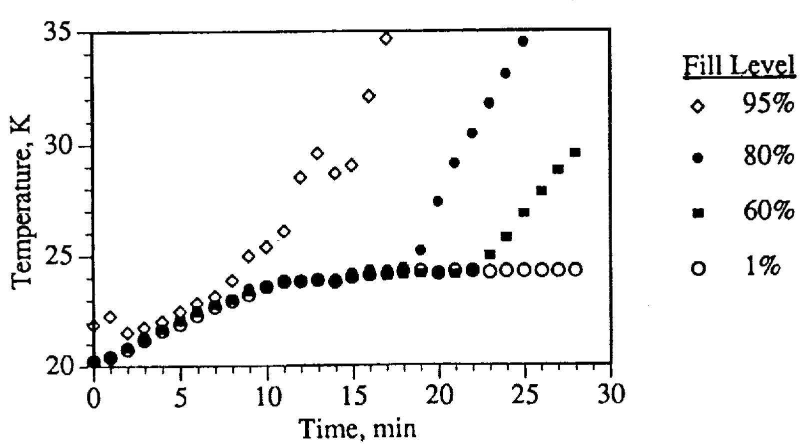

Fig. 4. Internal tank temperature histories for Test No. 5R. Ramp pressurization at 84 percent fill followed by expulsion to 7 percent fill.

Liquid expulsion time also did not have a significant effect on the pressurant energy input. Results were obtained for expulsion durations of approximately 15 and 25 min. As was the case with ramp pressurization, less pressurant mass was needed when the pressurant temperature was increased, with the energy input remaining the same for the two gas temperatures. Total energy input for the expulsion tests, including the ramp period, was 18,100 kJ ± 3 percent.

Mass balances on the vapor region indicate that 63 to 80 percent of the 275 K pressurant gas and 54 percent of the 330 K pressurant gas condenses during ramp pressurization. For the combined ramp and expulsion processes, the mass analysis indicates that net evaporation of the liquid occurs, with the amount of evaporated mass being of the same order as that of the injected pressurant.

Energy balances applied to the combined ramp and expulsion tests were found to balance to within 3 percent. Less than 0.4 percent of the total energy input (ΔU_T) was due to the tank heat leak. Using the analysis described above, it was found that approximately 89 percent of the incoming energy went into liquid heating ($\Delta U_L/\Delta U_T$), 10 percent into vapor heating ($\Delta U_u/\Delta U_T$), and 1 percent was absorbed by the tank wall ($\Delta U_w/\Delta U_T$). This distribution is in good agreement with thermal equilibrium calculations from the homogeneous model. Energy balances applied to only the ramp process were found to be in error by as much as 30 percent, with the calculated liquid heating exceeding the energy supplied by the pressurant. It is theorized that the error is due to the existence of radial temperature gradients in the liquid, with liquid heating away from the central vertical axis lagging that near the axis where measurements were obtained. The radial temperature gradients are thought to be most significant in the liquid region at the end of the ramp period.

Comparison of the experimental results with pressurant requirements predicted by the homogeneous model (Eq. 10) for the combined ramp and expulsion processes shows agreement within 6 percent, with the measured values generally exceeding predictions, as shown in Table 2. For ramp pressurization only, it is seen that the predicted pressurant mass is more than the experimentally measured values. This result is plausible if the liquid has radial temperature gradients.

The last two columns in Table 2 list the ideal pressurant requirements calculated from Eq. 11. The measured total pressurant consumption (for combined ramp and expulsion) exceeds the ideal amounts by a factor of approximately five. For other initial and/or final fill levels, this factor will vary. For the ramp pressurization process only, the factor ranges from 41 to 46. Actual values of this "collapse factor" for direct ullage pressurization fluctuate according to diffuser design and numerous other conditions. Generally, well designed direct ullage pressurization systems have collapse factors that are substantially less than five.

Table 2. Comparison of Measured and Predicted Pressurant Gas Requirements

Test No.	Pressurant Temperature (K)	Measured Ramp (kg)	Measured Total (kg)	Homogeneous Ramp (kg)	Homogeneous Total (kg)	Ideal Ramp (kg)	Ideal Total (kg)
1	275	3.0	n/a	3.2	n/a	0.066	n/a
2	275	3.0	n/a	3.2	n/a	0.065	n/a
3	275	2.9	n/a	3.2	n/a	0.068	n/a
5	275	3.0	4.7	3.2	4.5	0.066	0.90
5R	275	3.1	4.8	3.2	4.5	0.067	0.90
6	275	3.0	4.6	3.1	4.4	0.065	0.92
9	330	2.4	3.7	2.6	3.7	0.058	0.75

CONCLUSIONS

Normal-g pressurization and expulsion tests from the 84 to 7 percent fill level were conducted to simulate tank pressurization in a low-g environment. Autogenous pressurization of LH_2 by the submerged vapor injection technique produces substantial liquid heating. For ramp durations less than 30 min, the liquid heating does not appear to be radially uniform. After the expulsion process begins, liquid in the tank reaches the saturation temperature and approaches the homogeneous thermal state. Measured pressurant requirements for the combined ramp and expulsion processes were predicted by the thermal equilibrium analysis (homogeneous model) to within seven percent. Pressurant gas requirements exceed ideal requirements by a factor of approximately five. For spacecraft design, submerged pressurant injection should be avoided whenever possible due to the excessive amount of pressurant needed and the undesirable liquid heating. If submerged injection cannot be precluded, the thermal equilibrium model should be used to determine pressurant requirements.

NOMENCLATURE

C	specific heat
F	fill level (liquid volume/total volume)
h	specific enthalpy
M	mass
$\dot{M}$	mass flow rate
N	number of segments
P	pressure
$\dot{Q}$	heat leak rate
s	specific entropy
T	temperature
t	time
U	internal energy
V	volume
ρ	density

Subscripts

f	final
G	gas (pressurant)
i	initial
L	liquid
n	summation index
T	total
t	transfer due to phase change
u	ullage
w	wall

ACKNOWLEDGEMENTS

The authors would like to acknowledge the support of the many people who participated in the K-Site Phase 1B test program at Plum Brook Station.

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Report Documentation Page

1. Report No. NASA TM-104516	2. Government Accession No.	3. Recipient's Catalog No.
4. Title and Subtitle Autogenous Pressurization of Cryogenic Vessels Using Submerged Vapor Injection		5. Report Date
		6. Performing Organization Code
7. Author(s) Robert J. Stochl, Neil T. Van Dresar, and Raymond F. Lacovic		8. Performing Organization Report No. E-6386
		10. Work Unit No. 591-23
9. Performing Organization Name and Address National Aeronautics and Space Administration Lewis Research Center Cleveland, Ohio 44135-3191		11. Contract or Grant No.
		13. Type of Report and Period Covered Technical Memorandum
12. Sponsoring Agency Name and Address National Aeronautics and Space Administration Washington, D.C. 20546-0001		14. Sponsoring Agency Code

15. Supplementary Notes

Prepared for the Cryogenic Engineering Conference and International Cryogenic Materials Conference cosponsored by the University of Alabama in Huntsville, and NASA Marshall Space Flight Center, Huntsville, Alabama, June 11-14, 1991. Responsible person, Robert J. Stochl, (216) 433-5403.

16. Abstract

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17. Key Words (Suggested by Author(s))	18. Distribution Statement
Fuel tank pressurization Cryogenic fluids Reduced gravity	Unclassified - Unlimited Subject Category 28

19. Security Classif. (of the report)	20. Security Classif. (of this page)	21. No. of pages	22. Price*
Unclassified	Unclassified	10	A02

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*For sale by the National Technical Information Service, Springfield, Virginia 22161