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Neil T. Van Dresar
Lewis Research Center
Cleveland, Ohio

and

Mark S. Haberbusch
Ohio Aerospace Institute
Brook Park, Ohio

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Neil T. Van Dresar
National Aeronautics and Space Administration
Lewis Research Center
Cleveland, Ohio 44135

and

Mark S. Haberbush
Ohio Aerospace Institute
Brook Park, Ohio 44142

ABSTRACT

Thermodynamic models have been formulated to predict lower and upper bounds for the mass of pressurant gas required to pressurize a cryogenic tank and then expel liquid from the tank. Limiting conditions are based on either thermal equilibrium or zero energy exchange between the pressurant gas and initial tank contents. The models are independent of gravity level and allow specification of autogenous or non-condensable pressurants. Partial liquid fill levels may be specified for initial and final conditions. Model predictions are shown to successfully bound results from limited normal-gravity tests with condensable and non-condensable pressurant gases. Representative maximum collapse factor maps are presented for liquid hydrogen to show the effects of initial and final fill level on the range of pressurant gas requirements. Maximum collapse factors occur for partial expulsions with large final liquid fill fractions.

INTRODUCTION

Future space exploration missions will require large quantities of cryogenic liquids and the ability to transfer these fluids between earth-to-orbit tankage vehicles, orbiting depots, and space transportation vehicles. Such transfer operations will require supply tank pressurization and pressurized expulsion to achieve the transfer between spacecraft. The technique involves the introduction of a gaseous pressurant into a cryogenic supply tank. When the tank pressure becomes sufficiently high, liquid cryogen is expelled from the tank via a transfer line to the destination point. The principal objective of pressurization analysis is the determination of the required amount of pressurizing gas for transfer operations. An accurate estimate of the amount of pressurant gas required for a transfer operation is highly desirable; having excess pressurant carries a substantial weight penalty, while an insufficient amount will result in incomplete cryogen transfer.

Pressurization of a vessel containing cryogenic fluid by the addition of a relatively warm pressurant gas introduces a number of complex energy transfer processes into the system. Condensation of the pressurant may occur at the liquid-vapor and vapor-wall interfaces. Evaporation of liquid at the liquid-vapor interface may also occur. Heat and mass exchange rates at the liquid-vapor and vapor-wall interfaces are dependent upon widely varying conditions. For example, low gravity performance will differ from that in normal-gravity due to differences in liquid orientation and thermal stratification.

A logical starting point for calculating the amount of pressurant required for a transfer operation is to determine the extreme case requirements, thus bounding the analysis. This approach has been reported by Moore *et al*¹. The limitations of the Moore analysis are that

the models assume an initially full tank which is entirely emptied and that only condensible pressurant gas is considered. This paper describes an extension of the earlier analysis that allows calculations for various initial and final tank fill levels using either autogenous (condensible) or non-condensible pressurant gases. In these models, the tank is first pressurized to a higher pressure (the ramp process) and then the liquid is expelled at constant tank pressure. The analyses are applicable to tank pressurization and expulsion in low or normal gravity environments. Comparisons with experimental data are provided as well as example calculations for hypothetical space applications.

THERMODYNAMIC ANALYSES

Two models will be described. The first is for an ideal process, which leads to relatively low pressurant mass requirements, while the second imposes a thermal equilibrium condition that leads to significantly higher requirements. The idealized situation is characterized by a complete absence of energy and interfacial mass transfer. The equilibrium model may be considered as an extreme case representative of conditions such as lengthy transfer times coupled with vigorous mixing of the tank fluid and pressurant. The models may not establish absolute minimum and maximum pressurant needs, but are expected to sufficiently bound actual requirements.

Heat exchange with the tank wall is not allowed in the ideal model, and is neglected in the thermal equilibrium model due to the resulting narrow wall temperature range and low specific heat of typical wall materials if the wall is assumed to be in thermal equilibrium with the tank contents. The validity of ignoring the wall energy increase in the thermal equilibrium model is illustrated by Fig. 1 which shows the amount of total energy input absorbed by wall heating for self-pressurization of a representative lightweight tank for spaceflight under homogeneous, thermal equilibrium conditions. The assumption is clearly valid for liquid hydrogen and is acceptable for nitrogen and oxygen except when the liquid fill level approaches zero. In applications with tanks of substantial heat capacity, energy exchange with the wall becomes significant—thus inclusion of wall heating would be necessary.

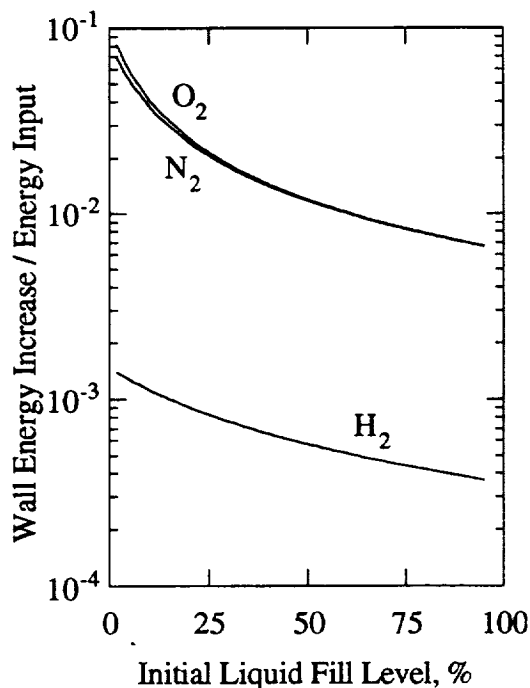


Figure 1 - Wall Energy Increase as a Fraction of Total Energy Input. Self-Pressurization Under Thermal Equilibrium Conditions, 100 to 200 kPa. 150 kg Aluminum Alloy Tank, 5 m³.

Potential and kinetic energy terms are neglected. The tank (control volume) is assumed to be rigid, therefore work due to volume expansion or contraction is zero. Furthermore, the tank is assumed to be well-insulated, and heat transfer from the surroundings is neglected. When a noncondensable pressurant is used, it is also assumed that the gas is insoluble in the liquid. (This is usually, but not always, a valid assumption.) In the analyses that follow, the initial fluid state prior to pressurization is saturated at the starting pressure and free of any noncondensable component.

Ideal Model

Thermal isolation of the liquid, initial ullage, and added pressurant components is assumed. It is assumed that the initial ullage mass undergoes an isentropic compression during ramp pressurization. For isentropic compression over the pressure range of interest (100 to 350 kPa), the liquid volume is constant to within one-half of one percent. Thus, it is unnecessary to include the liquid contents in the analysis. The added pressurant gas fills the remaining ullage volume during pressurization and occupies the volume of the displaced liquid during expulsion. It is assumed that the pressurant gas remains at its inlet temperature after entering the tank. The pressurant requirement is calculated from a simple volumetric analysis

$$m_p = \rho_{b,2} V \left[1 - F_2 - \frac{\rho_{a,1}}{\rho_{a,2}} (1 - F_1) \right] \quad (1)$$

where the required gas and vapor densities are obtained from thermodynamic tables.

$$\rho_{a,1} = \rho_{sat}(P_1) \quad (2a)$$

$$\rho_{a,2} = \rho(P_2, s) \quad \text{with } s = s_{sat}(P_1) \quad (2b)$$

$$\rho_{b,2} = \rho(P_2, T_{in}) \quad (2c)$$

The pressurant gas may be either condensible or noncondensable.

Thermal Equilibrium Model-Autogenous Case

Thermal equilibrium is assumed to exist between the tank contents and the added pressurant—the pressurant gas, vapor, and liquid are all at a uniform temperature. It is further assumed that equilibrium is maintained throughout both the ramp and expulsion processes. The ramp ends at an increased liquid fill level due to thermal expansion of the liquid and condensation of autogenous pressurant gas. Consequently, the specified initial liquid fill fraction must be sufficiently less than unity (approximately ≤ 0.95) to allow adequate space for thermal expansion of the liquid during the process.

Governing equations are applied to a control volume containing the tank fluid contents. Pressurant mass and energy enter the control volume and liquid mass and energy exit the control volume during expulsion. Two governing equations are the conservation of mass and energy. For the autogenous case, the final fluid state is saturated at the final tank pressure. If final fill level is specified, there are two unknowns—pressurant mass and expelled liquid mass. Simultaneous solution of the mass and energy conservation equations is sufficient. If total liquid outflow mass is specified (0 for ramp), there are three unknowns—pressurant mass, final vapor mass, and final liquid mass. A third equation—a volume balance at the final state—is required to obtain the solution.

Conservation of mass - autogenous case:

$$m_{\ell,1} + m_{a,1} + m_p = m_{\ell,2} + m_{a,2} + m_o \quad (3)$$

Conservation of energy - autogenous case:

$$\begin{aligned} m_{\ell,1}u_{\ell,1} + m_{a,1}u_{a,1} + m_p h_{in} \\ = m_{\ell,2}u_{\ell,2} + m_{a,2}u_{a,2} + m_o h_o \end{aligned} \quad (4)$$

Volume constraint - autogenous case:

$$\frac{m_{\ell,2}}{\rho_{\ell,2}} + \frac{m_{a,2}}{\rho_{a,2}} = V \quad (5)$$

Thermal Equilibrium Model-Noncondensable Case

When the equilibrium model is applied to processes involving a noncondensable pressurant, the modeling becomes more complex. The two component ullage is modeled by the Amagat Law—each component is at the mixture temperature and pressure, and the sum of the component volumes equals the total ullage volume. It is not known beforehand whether the fluid temperature is saturated or subcooled. If it is saturated, the temperature is specified and the ullage contains both vapor and the noncondensable gas. If the temperature is less than the saturation temperature at the final pressure (subcooled condition), then the temperature is an unknown, and the ullage contains only noncondensable gas (since the vapor cannot exist at this state). The same governing equations apply with additional terms for the noncondensable ullage component. When the temperature is below saturation, the vapor component is excluded.

Conservation of mass - noncondensable case:

$$m_{\ell,1} + m_{a,1} + m_{b,1} + m_p = m_{\ell,2} + m_{a,2} + m_{b,2} + m_o \quad (6)$$

Conservation of energy - noncondensable case:

$$\begin{aligned} m_{\ell,1}u_{\ell,1} + m_{a,1}u_{a,1} + m_{b,1}u_{b,1} + m_p h_{in} \\ = m_{\ell,2}u_{\ell,2} + m_{a,2}u_{a,2} + m_{b,2}u_{b,2} + m_o h_o \end{aligned} \quad (7)$$

Volume constraint - noncondensable case:

$$\frac{m_{\ell,2}}{\rho_{\ell,2}} + \frac{m_{a,2}}{\rho_{a,2}} + \frac{m_{b,2}}{\rho_{b,2}} = V \quad (8)$$

The ramp process is solved first. Saturation is assumed, and the above equations yield a solution for the three unknowns— m_{ℓ} , $m_{a,2}$, and m_p ($= m_{b,2}$ for ramp only). If a meaningful solution cannot be obtained, the liquid is subcooled. An iterative technique is then employed to determine the unknowns— m_{ℓ} , m_p , and the final temperature.

If the ramp ends at the saturated state, the expulsion will also occur under saturated conditions with saturated liquid outflow. There are four unknowns—

$m_{a,2}$, $m_{b,2}$, m_p , and $m_{\ell,2}$ or m_o —and a fourth governing equation, conservation of the noncondensable component only, is required to obtain a solution.

Conservation of noncondensable mass:

$$m_{b,1} + m_p = m_{b,2} \quad (9)$$

However, if the expulsion starts at a subcooled liquid state, energy from the pressurant heats the liquid thus causing the temperature to rise and the outflow enthalpy to increase. Under this condition, the expulsion process is solved incrementally and requires solving for four unknowns— $m_{b,2}$, m_p , $m_{\ell,2}$ or m_o , and the final temperature at each expulsion increment.

As the expulsion proceeds, it is possible to reach the saturation temperature—at this point the solution procedure must switch to the saturation analysis discussed earlier.

Thermal Equilibrium Model-Parahydrogen/Normal Hydrogen

Pressurization of liquid parahydrogen with normal hydrogen gas is a special case requiring a separate thermal equilibrium analysis since the available sensible heat of normal hydrogen (a mixture of para- and orthohydrogen) is less than for pure parahydrogen. This model uses an approach similar to the autogenous equilibrium model with internal energies and outflow enthalpy based on mixture properties. It is assumed that no ortho- to parahydrogen conversion occurs and that the mass fraction of normal hydrogen is uniform throughout the control volume. An iterative procedure is required to obtain the normal hydrogen mass fraction, x .

Liquid specific internal energy:

$$u_{\ell} = xu_{\ell,n} + (1-x)u_{\ell,p} \quad (10)$$

Vapor specific internal energy:

$$u_a = xu_{a,n} + (1-x)u_{a,p} \quad (11)$$

Outflow specific enthalpy:

$$h_o = xh_{o,n} + (1-x)h_{o,p} \quad (12)$$

A normal hydrogen mass conservation equation is used to determine convergence of the normal hydrogen mass fraction:

$$x_2 = \frac{m_p + x_1(m_{\ell,1} + m_{a,1} - m_o/2)}{m_{\ell,2} + m_{a,2} + m_o/2} \quad (13)$$

For ramp calculations, m_o and x_1 are zero. During expulsion, the mass fraction of normal hydrogen increases, therefore an incremental expulsion analysis is necessary to update the vapor and liquid specific energies and outflow enthalpy.

COMPARISON WITH EXPERIMENTAL DATA

The ideal and thermal equilibrium models are compared in Fig. 2 to normal-gravity data^{2,3} where liquid parahydrogen was expelled using normal hydrogen pressurant gas. In ref. 2, the pressurant gas was introduced through a diffuser located in the ullage region, while in ref. 3, the pressurant was injected directly into the liquid near the bottom of the tank. Total pressurant mass is the amount required for the ramp plus expulsion. The models adequately bound the experimental data and show increasing pressurant requirements with increasing expulsion pressure. The maximum collapse factor, defined as

$$CF_{\max} = \frac{\text{thermal equilibrium pressurant mass}}{\text{ideal pressurant mass}} \quad (14)$$

increases with increasing expulsion pressure as well. Since normal hydrogen gas was used in the experiments, the data cannot be strictly classified as autogenous pressurization. In the figure, the predictions of the autogenous thermal equilibrium model are also shown. It is apparent that this model predicts pressurant mass values lower than measured values and thus does not provide a sufficient upper bound for pressurant requirements.

Figure 3 shows a comparison of the models with experimental data⁴ where a noncondensable pressurant gas, helium, was used to expel liquid methane. The differences in measured pressurant requirements are due to static vs. liquid sloshing conditions in the tank, the presence or absence of anti-slosh baffles in the tank, and variations in expulsion time. Again, the models show sufficient bounding of experimental results. Both models exhibit decreasing pressurant requirements with increasing inlet pressurant gas temperature. The ideal model predictions follow the inverse temperature dependence for ideal gas behavior, while the thermal equilibrium model is relatively insensitive to pressurant temperature because of the large heat capacity of the liquid methane. It can be determined from the model curves that the maximum collapse factor increases with pressurant temperature.

EXAMPLE CALCULATIONS

The models were used to generate maximum collapse factor maps for pressurization and expulsion of liquid parahydrogen as functions of the initial and final fill fractions. The tank pressure range and inlet gas temperature were selected as representative values for space applications. Figure 4 is a contour plot for the autogenous case, while Fig. 5 provides results for pressurization using gaseous helium, a noncondensable. The contours are similar, and exhibit approximately the same range of maximum collapse factors except in the upper right corner of the maps where the maximum values are higher for the autogenous case. At large final fill fractions (approximately greater than 0.5), the ratio of maximum-to-minimum pressurant mass is a factor of 10 or more. This result indicates the importance of accurate predictive capabilities for pressurant requirements in initially full systems undergoing partial expulsions. Conversely, as the initial and final fill levels approach zero, the maximum collapse factor decreases and the use of simple predictive models may suffice.

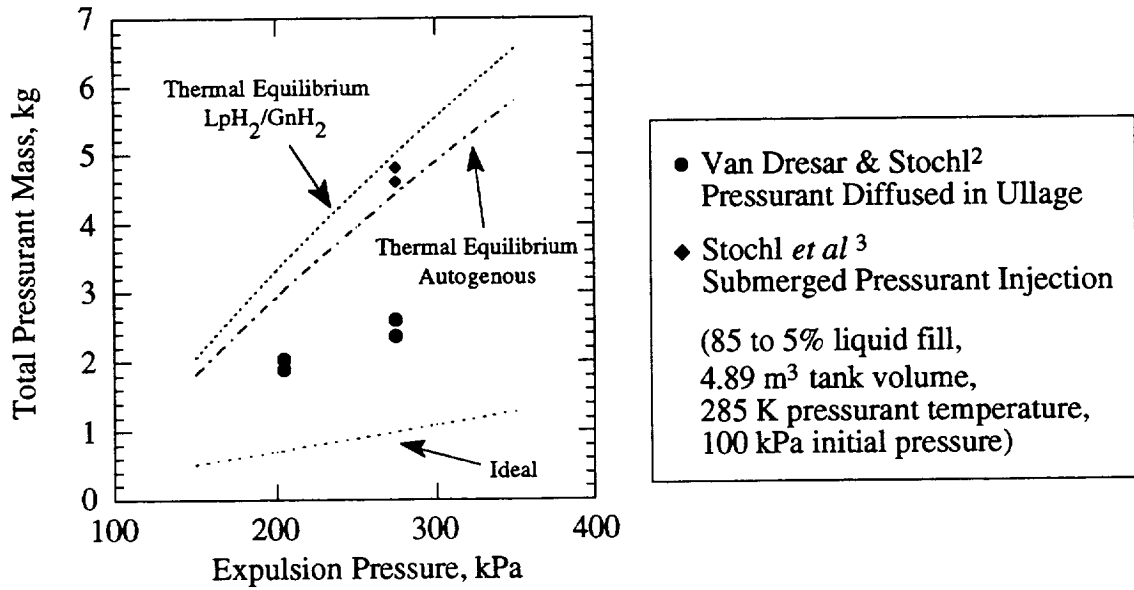


Figure 2 - Comparison of Ideal and Thermal Equilibrium Models with Experimental Data for Liquid Parahydrogen Expulsion Using Normal Hydrogen Pressurant.

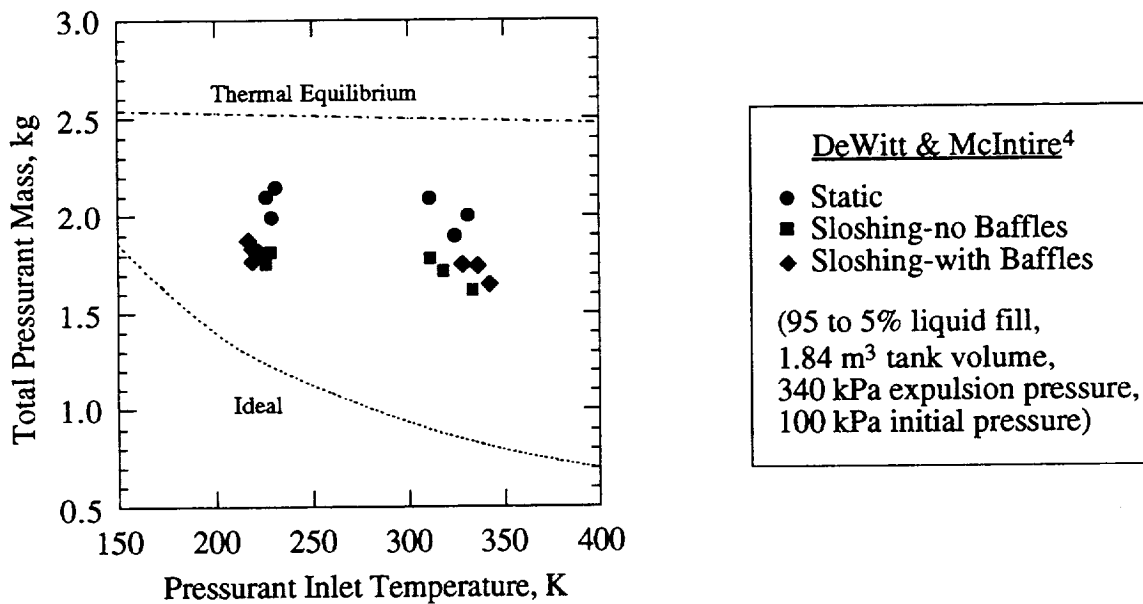


Figure 3 - Comparison of Ideal and Thermal Equilibrium Models with Experimental Data for Liquid Methane Expulsion Using Helium Pressurant.

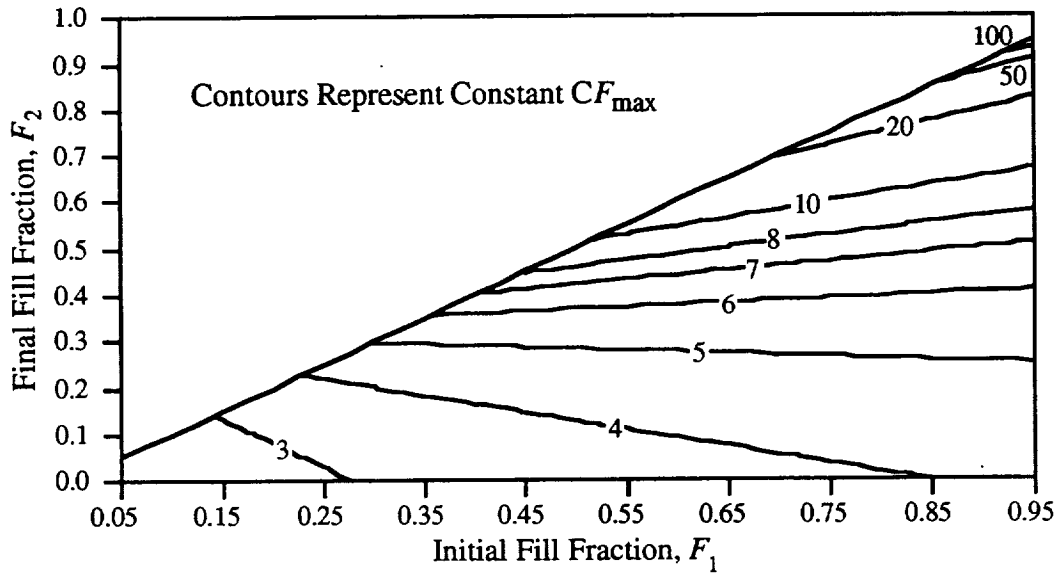


Figure 4 - Maximum Collapse Factor Map for Liquid Parahydrogen Pressurized with Gaseous Parahydrogen (Autogenous Pressurant). 100 to 200 kPa, 300 K inlet gas temperature.

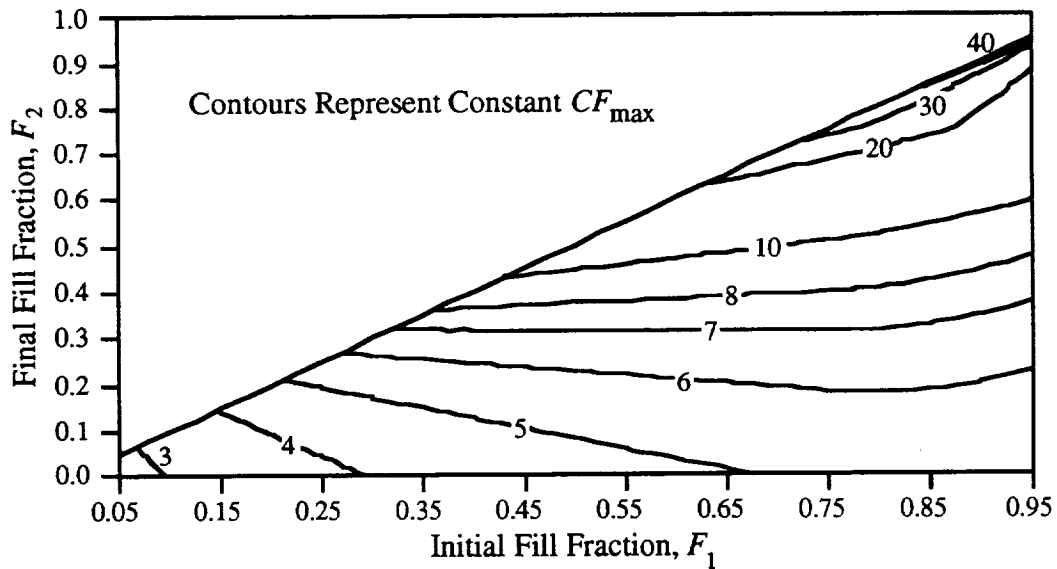


Figure 5 - Maximum Collapse Factor Map for Liquid Parahydrogen Pressurized with Gaseous Helium (Noncondensable Pressurant). 100 to 200 kPa, 300 K inlet gas temperature.

CONCLUSIONS

The use of the ideal and thermal equilibrium models to bound pressurant gas requirements has been extended to partially filled tanks and to the use of noncondensable pressurant gases. Pressurization of liquid parahydrogen with gaseous normal hydrogen has been treated as a special case of the thermal equilibrium model. The models have been validated with limited normal-gravity data and allow the determination of the possible range of pressurant mass requirements in preliminary design calculations. The magnitude of the maximum collapse factor allows one to evaluate the need for improved predictive capabilities.

NOMENCLATURE

CF	collapse factor
F	liquid fill fraction (by volume)
h	specific enthalpy
m	mass
P	pressure
s	specific entropy
T	temperature
u	specific internal energy
V	total tank volume
x	normal hydrogen mass fraction
ρ	density

subscripts

a	initial vapor component (ideal model) or vapor (t.e. model)
b	pressurant component (ideal model) or noncondensable (t. e. model)
in	pressurant inlet
l	liquid
max	maximum
n	normal hydrogen
o	outflow
p	pressurant or parahydrogen
sat	saturated
1	starting condition
2	ending condition

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