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# THERMODYNAMIC MODELING OF THE NO-VENT FILL METHODOLOGY FOR TRANSFERRING CRYOGENS IN LOW GRAVITY

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## Abstract

The filling of tanks with cryogenics in the low-gravity environment of space poses many technical challenges. Chief among these is the inability to vent only vapor from the tank as the filling proceeds. As a potential solution to this problem, the NASA Lewis Research Center (NASA Lewis) is researching a technique known as No-Vent Fill. This technology potentially has broad application. The focus of this paper is the fueling of space based Orbital Transfer Vehicles.

This paper describes the fundamental thermodynamics of the No-Vent Fill process to develop an analytical model of No-Vent Fill. The model is then used to conduct a parametric investigation of the key parameters; initial tank wall temperature, liquid-vapor interface heat transfer rate, liquid inflow rate and inflowing liquid temperatures. Liquid inflowing temperature and the liquid-vapor interface heat transfer rate seem to be the most significant since they influence the entire fill process. The initial tank wall temperature must be sufficiently low to prevent a rapid pressure rise during the initial liquid flashing stage, but then becomes less significant.

## Nomenclature

A	area
Cv	specific heat at constant volume
h	enthalpy
$\bar{h}$	convective heat transfer coefficient
k	thermal Conductivity
L	characteristic length
M	mass
$\dot{m}$	mass flow rate
q	heat flux
Q	heat Input
T	temperature
t	time
U	total internal energy
u	internal energy per unit mass
V	volume
W	work
$\dot{W}$	rate of work

$\alpha$  thermal diffusivity

$\rho$  density

Subscripts:

cond condensate

fgas flashing gas

fliq flashing liquid

gas ullage gas

in inlet

inf interface

lg between liquid and gas

liq bulk liquid

sat saturation

sgas saturated gas

sliq saturated liquid

tank tank

wall tank wall

## Introduction

Many fluid management issues face the designer of systems for filling tanks with cryogenics on-orbit. The most important of these are the large rate of generation of vapor from the residual energy stored in cryogenic tank walls, the uncertainty of liquid and vapor distributions in a tank in low gravity, and the need to keep maximum tank pressure low to reduce tank mass. During a normal gravity fill a top vent is kept open to vent the vapor generated during the fill process thereby maintaining a low tank pressure. If the same approach is used in low gravity the ullage gas may never vent. Instead of venting vapor, large amounts of liquid may be dumped overboard. If liquid is vented from one side of a nonpropulsive vent and vapor from the other side, the spacecraft may tumble out of control. To prevent the loss of liquid, the spacecraft may be placed in an artificial gravity field by continuous thruster firing to position the ullage at a vent opening, but this may require dedicated thrusters and additional propellant. The problems of low-g fluid management and the thruster technique for liquid positioning have been described previously by Lacovic.<sup>1</sup> For the fill operation, maintaining control between the tanker spacecraft and the spacecraft being refueled while thrusting may be difficult due to potentially large shifts in the center of gravity. In some instances, such as a depot based at a space station, thrusting may be impractical due to large system size. The No-Vent Fill technique, first analyzed in detail by Merino, is a methodology whereby these problems are eliminated.<sup>2</sup>

\*Member AIAA

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The No-Vent Fill methodology is as follows: The tank wall is prechilled to a temperature sufficient to remove most of the thermal energy from it. The tank is evacuated by venting to space, and liquid is then injected through spray nozzles and/or mixing jets to promote liquid-vapor heat exchange. The tank can be filled without venting as long as the prechill is sufficient and the liquid-vapor heat exchange keeps the liquid and vapor close to thermodynamic equilibrium.

In the main body of this paper, a model for the initial flashing stage (when tank pressure is less than the liquid saturation pressure) is formulated. A model for the later stages, which preserves the temperature difference between the bulk liquid and the vapor, is also formulated. These two models are combined to provide a transient model of the No-Vent Fill process. A computer program was developed to solve the model equations, and then used to conduct a parametric study of the No-Vent Fill process. Parameters investigated include: initial tank wall temperature, interface heat transfer, liquid inflow rate, and inflowing liquid temperature.

### No-Vent Fill Theory

#### General

Following the tank chillover and evacuation, liquid inflow is started. Liquid is injected through spray nozzles and/or mixing jets to promote liquid-vapor heat exchange. The initial inflow partially flashes while entering, and vaporizes on striking the tank wall. This removes the remaining residual energy from the tank walls and raises the tank pressure to near saturation conditions. The continuing inflow of liquid either condenses or compresses the vapor allowing more liquid to enter. For a period of time condensation dominates and tank pressure remains fairly constant. Eventually vapor compression takes over. The tank pressure rises rapidly to the limit pressure, and the fill process must be stopped. It is the objective of No-Vent Fill technique to postpone the compression phase as long as possible.

To understand the No-Vent Fill concept it is helpful to examine the thermodynamics of the Fill process. It can be shown by an equilibrium analysis of the thermodynamics that, with a sufficiently low temperature of the entering liquid, the end state of a full tank should be at an acceptably low pressure. Unfortunately like all real processes, the fill process does not proceed at thermodynamic equilibrium. Major sources of nonequilibrium behavior are as follows: (1) energy transfer from the initially warm tank to the entering fluid, (2) heat flux into the tank during the fill process, (3) flashing of the liquid during the initial low pressure stages of the fill, and (4) temperature gradients within and between the tank, the vapor, and the liquid.

A model which separates the vapor and liquid phases is needed to determine the effect of temperature differences and the rate of heat transfer between the two phases. Since the initial flashing stage of the flow is controlled by different phenomena than the condensation and compression stages, the model will be divided into two parts. For simplicity, heat flux into the tank during the fill process will be assumed to be negligible.

#### Liquid Flashing Stage

The initial flashing stage is modeled as follows. The high heat transfer rates found in boiling make it unlikely that much liquid will remain in the tank until the wall is chilled to near the incoming liquid temperature. When the tank pressure is less than the saturation pressure of the incoming liquid, the liquid inflow will flash resulting in a gaseous inflow which enters the ullage vapor directly and a liquid stream which is cooled to the saturated state corresponding to the current tank pressure. This liquid stream then strikes the tank wall and is assumed to be quickly vaporized to a saturated vapor. A comparison between the heat transfer rates of film boiling of the liquid stream and gaseous conduction into the ullage gas (the comparison is included in an appendix to this report) indicates that the effects of wall-gas heat transfer are secondary as long as conduction is the predominate heat transfer mechanism. For simplicity the effects of wall-gas heat transfer will be assumed negligible. The formulation presented here only accounts for the tank wall energy removed by the vaporizing liquid stream. The equations for this stage are:

Inflow energy balance at the liquid inlet

$$\dot{m}_{in} h_{in} = \dot{m}_{fgas} h_{sgas} + \dot{m}_{fliq} h_{sliq} \quad (1)$$

Initial flashing mass balance

$$\dot{m}_{in} = \dot{m}_{fgas} + \dot{m}_{fliq} \quad (2)$$

Wall energy balance

$$-M_{wall} \frac{d(C_v T)}{dt} = \dot{m}_{fliq} (h_{sgas} - h_{sliq}) \quad (3)$$

Gas mass balance

$$\frac{dM_{gas}}{dt} = \dot{m}_{fgas} + \dot{m}_{fliq} \quad (4)$$

Gas energy balance

$$\frac{d(M_{gas} U_{gas})}{dt} = (\dot{m}_{fgas} + \dot{m}_{fliq}) h_{sgas} \quad (5)$$

Rearranging Eq. (1) and substituting Eq. (2) into Eq. (1)

$$\dot{m}_{in} h_{in} = (\dot{m}_{in} - \dot{m}_{fliq}) h_{sgas} = \dot{m}_{sliq} h_{fliq} \quad (6)$$

or

$$\dot{m}_{in} (h_{in} - h_{gas}) = \dot{m}_{fliq} (h_{sliq} - h_{sgas}) \quad (7)$$

Substituting Eq. (7) into (3)

$$-M_{wall} \frac{d(C_v T)}{dt} = \dot{m}_{in} (h_{sgas} - h_{in}) \quad (8)$$

Substituting Eq. (2) into (4)

$$\frac{dM_{gas}}{dt} = \dot{m}_{in} \quad (9)$$

and into Eq. (5)

$$\frac{d(M_{\text{gas}} u_{\text{gas}})}{dt} = \dot{m}_{\text{in}} h_{\text{sgas}} \quad (10)$$

Expanding Eq. (5) and using the results of (9) and (10):

$$M_{\text{gas}} \frac{du_{\text{gas}}}{dt} = \dot{m}_{\text{in}} (h_{\text{sgas}} - u_{\text{gas}}) \quad (11)$$

For the times when the tank pressure is above the inflowing liquid saturation pressure, Eqs. (8), (9), and (11) can be written directly as the mass and energy equations for the Fill process. Once the liquid and wall temperatures are equal the residual wall energy is negligible and the model can be switched to the condensation and compression stage equations.

#### Vapor Condensation and Compression Stage

To study the relation between gas and liquid the fluid is broken into three nodes: gas, liquid and interface. A mass and energy balance can be written for each node.

Gas mass balance:

$$\frac{dM_{\text{gas}}}{dt} = -\dot{m}_{\text{cond}} \quad (12)$$

Gas energy balance:

$$\begin{aligned} \frac{d(M_{\text{gas}} u_{\text{gas}})}{dt} &= M_{\text{gas}} \frac{du_{\text{gas}}}{dt} + \dot{m}_{\text{cond}} u_{\text{gas}} \\ &= \dot{m}_{\text{cond}} h_{\text{gas}} + \dot{W}_{\text{lg}} \end{aligned} \quad (13)$$

Liquid mass balance:

$$\frac{dM_{\text{liq}}}{dt} = \dot{m}_{\text{in}} + \dot{m}_{\text{cond}} \quad (14)$$

Liquid energy balance:

$$\begin{aligned} M_{\text{liq}} \frac{du_{\text{liq}}}{dt} + u_{\text{liq}} \frac{dM_{\text{liq}}}{dt} + q_{\text{inf}} + \dot{m}_{\text{in}} h_{\text{in}} + \dot{m}_{\text{cond}} h_{\text{iq}} \\ = \dot{W}_{\text{lg}} \end{aligned} \quad (15)$$

The interface node has some unique features. It is assumed to be a infinitesimally thin layer between the liquid and gas. Since it is infinitesimally thin, it can store neither mass nor energy. This makes the equations for the interface as follows:

Interface mass balance:

$$\frac{dM_{\text{inf}}}{dt} = 0 \quad (16)$$

Interface energy balance:

$$q_{\text{inf}} + \dot{m}_{\text{cond}} h_{\text{liq}} = \dot{m}_{\text{cond}} h_{\text{gas}} \quad (17)$$

Some additional equations are needed to solve this set of equations.  $q_{\text{inf}}$  is determined by a convective heat transfer equation. Heat transfer from the gas side can be neglected because gas thermal conductivity is much smaller than liquid thermal conductivity for the fluids of interest in

this document. The main effects of the gas is the mass transport of energy already accounted for in Eq. (17).  $q_{\text{inf}}$  is then a function of liquid side heat transfer. Writing this as a convective equation:

$$q_{\text{inf}} = \bar{h} A_{\text{inf}} (T_{\text{inf}} - T_{\text{liq}}) \quad (18)$$

It can be shown by kinetic theory that:

$$T_{\text{inf}} = T_{\text{sat}} \quad (19)$$

Combining Eqs. (17) to (19).

$$\dot{m}_{\text{cond}} = \frac{\bar{h} A_{\text{inf}} (T_{\text{sat}} - T_{\text{liq}})}{(h_{\text{gas}} - h_{\text{liq}})} \quad (20)$$

The work on the gas can be calculated from basic thermodynamics for a control volume:

$$W = PdV \quad (21)$$

If the pressure rise is not too great:

$$\dot{W}_{\text{lg}} = P_{\text{gas}} \frac{dV_{\text{gas}}}{dt} \quad (22)$$

In a tank of constant volume:

$$V_{\text{gas}} = V_{\text{liq}} = \text{constant} \quad (23)$$

From Eq. (23):

$$-\frac{dV_{\text{gas}}}{dt} = \frac{dV_{\text{liq}}}{dt} \quad (24)$$

From Eq. (24)

$$\frac{dV_{\text{liq}}}{dt} = -\frac{1}{\rho_{\text{liq}}} (\dot{m}_{\text{in}} + \dot{m}_{\text{cond}}) \quad (25)$$

So:

$$\dot{W}_{\text{lg}} = \frac{P_{\text{gas}}}{\rho_{\text{liq}}} (\dot{m}_{\text{in}} + \dot{m}_{\text{cond}}) \quad (26)$$

Equations (12) to (17), (20), and (26) along with the thermodynamic equations of state form a fairly complete model of the No-Vent Fill process. The chief source of thermodynamic inefficiency is the work term  $\dot{W}_{\text{lg}}$ . The major independent variables in this analysis are  $\dot{m}_{\text{in}}$ ,  $h_{\text{in}}$ ,  $T_{\text{wall}}$ , and  $V_{\text{tank}}$ .

#### Analysis

The equations for the two stage model are sufficiently complex that a closed form solution is not possible. A computer algorithm called NVFILL has been written to solve the equations using a finite difference approximation. The variables  $\dot{m}_{\text{in}}$ ,  $h_{\text{in}}$ ,  $h$ , and  $V_{\text{tank}}$  are inputs to the code and held constant throughout a run.  $T_{\text{wall}}$  is set at the start.  $A_{\text{inf}}$  is defined as the surface area of a sphere whose volume is that of  $V_{\text{gas}}$ , the gas volume which changes with time.

During the initial flashing stage, the computational algorithm adds the mass added to the

ullage during the timestep, calculates a new ullage density internal energy and enthalpy, and then uses the density and enthalpy to determine a new ullage pressure. In the condensation-compression stage the problem is solved by a two-step calculation procedure. For the first step, the ullage is held at constant pressure while a interfacial mass transfer rate is calculated. For the second step, a new liquid volume is calculated from the liquid inflow, interfacial mass transfer and current bulk liquid conditions. The ullage vapor is then compressed adiabatically to fill the remaining tank volume. The time step for both these processes combined is set to  $10^{-3}$  hr which is sufficiently small compared to the process rates for most cases of interest to insure a good approximation of the continuous liquid inflow, interfacial mass transfer and vapor compression processes. The computational algorithm terminates when the tank is 95 percent full with liquid or the tank pressure exceeds 60 psia.

### Results and Discussion

Many differing tank sizes and geometries exist in the literature for Orbit Transfer Vehicle (OTV) designs. To provide a comparison with previous work, the OTV tank parameters employed by DeFelice and Aydelott are used.<sup>3</sup> These OTV parameters are summarized in Table 1. The baseline OTV parameters in Table 1 are representative of an unmanned space based OTV and are expected to be the sizes used in an initial operational space based OTV. Only the hydrogen tankage is examined since previous work indicates that the fill transients in the hydrogen tankage are the most severe.<sup>2</sup>

The effect of initial tank temperature on the tank pressure, as filling proceeds, is shown in Fig. 1. The 218.3 K (393 °R) temperature was the suggested initial tank temperature from Ref. 3. It produces a large initial pressure transient and causes the tank to fill at a much higher pressure than the other initial tank temperatures investigated. The 152.8 K (275 °R) initial condition was selected as a wall temperature which will raise the pressure in the tank just above the stagnation pressure of the incoming liquid at the end of the vapor flashing stage. A temperature of 55.6 K (100 °R) was selected as a likely minimum temperature, below which little effect on tank pressure results because all the tank energy has been essentially removed. It can be seen from Fig. 1 that even though the 55.6 K (100 °R) initial tank temperature transitions from the wall chilldown to the compression/condensation stage at about 130 kPa (15 psia) bulk boiling in the liquid raises the tank fill pressure characteristics to near the 152.8 K (275 °R) line for the majority of the fill process.

Reference 3 selected the initial tank temperature based on a criteria which yielded the desired fill level at the end of the fill using a thermodynamic equilibrium analysis to predict the end state. The initial pressure transient it produces was not accounted for in the Ref. 3 analysis and may be undesirable because of the control problems it poses for tanks with strict maximum pressure requirements. The stagnation pressure requirement used to select the intermediate initial wall temperature is probably a better criteria for initial wall temperature since it minimizes the initial pressure transient. Consequently, the 152.8 K

(275 °R) wall temperature was used for the rest of the No-Vent Fill studies.

Previous work on fluid mixing give wide ranges of surface heat transfer coefficients.<sup>4-6</sup> To cover the range of expected values, heat transfer coefficients spanning the range of two decades from 5.68 to 568 W/m<sup>2</sup> K (1 to 100 Btu/ft<sup>2</sup> hr °R) were parametrically evaluated. As can be seen in Fig. 2 (data from Fig. 1 assumed 56.8 W/m<sup>2</sup> K [10 Btu/ft<sup>2</sup> hr °R]), heat transfer coefficient has a significant effect on the tank pressure response during the No-Vent Fill process. For the smallest heat transfer coefficient, tank pressure exceeds 414 kPa (60 psia) prior to completion of the fill process. This is the only analysis run which did not terminate at 95 percent full. The highest heat transfer coefficient produces hardly any pressure rise beyond the initial flashing stage.

The 56.8 W/m<sup>2</sup> K (10 Btu/ft<sup>2</sup> hr °R) heat transfer coefficient was selected as a estimate for No-Vent Fill calculations for the rest of the study. Heat Transfer rates above this are likely to be achieved only by using forced convection to promote liquid-vapor heat transfer.

Figure 3 shows the tank pressure response for No-Vent Fills at liquid inflow rates at 453.5 kg/hr (1000 lbm/hr), 907 kg/hr (2000 lbm/hr, Table 1 rate) and 1814 kg/hr (4000 lbm/hr). Although the curves are fairly close for most of the fill process there is a significant difference in the final pressure with the highest pressure corresponding to the highest flow rate.

Figure 4 shows fills at various inflowing liquid temperatures corresponding to saturation pressures of 34.5 kPa (5 psia), 103 kPa (15 psia) and 172 kPa (25 psia). Inflowing liquid temperature shows a very strong effect on the No-Vent Fill process. This is to be expected given the dependence of the condensation rate on temperature difference. For real spacecraft, subatmospheric subcooling (lowering the liquid temperature below the saturation temperature at atmospheric pressure) may be difficult to achieve so 103 kPa (15 psia) was used as the baseline for the other parametric runs. The pronounced effect of subcooling indicates that further investigation of heat exchangers to cool the liquid inflow is probably worthwhile.

### Concluding Remarks

Several areas for model improvement can be suggested as a result of this study. Most evident is the need for a more accurate correlation relating the interfacial heat transfer rate to the independent parameters of No-Vent Fill. Two different correlations are under investigation currently but are dependent on liquid injection technique; one for droplet sprays through the ullage, and a second for turbulent jet mixing of the bulk liquid. In most real transfer systems the inflow rate will not be constant but will be controlled by the pressure difference between the liquid supply and the tank being filled. A model which links the flow rate to the tank pressure variation is being developed. The effects of wall-vapor and vapor-liquid heat transfer on the initial vapor flashing stage which were neglected in the analysis of this paper are under continued investigation. Several proposed injection techniques for No-Vent Fill show the potential for promoting forced convection in

the vapor thus raising the wall vapor heat transfer rate much closer to the liquid boiling heat transfer rate. The exact heat transfer correlations for these conditions are still being determined.

The results of these parametric investigations help define areas of research where the No-Vent Fill process should be studied. Several ground test experiments are being prepared at NASA Lewis to study the controlling phenomena. One experimental apparatus will employ liquid Nitrogen as the test fluid and provide controlled flows of liquid and gas which can be used to study heat transfer and condensation on the liquid interface. A small portable experimental test rig capable of being operated in a remote area with LH<sub>2</sub> is also being fabricated. This test rig will be used to study the effect of thermal subcooling and injection technique on the No-Vent Fill process. In addition, NASA Lewis is reactivating a large cryogenic vacuum chamber to allow No-Vent Fill testing with LH<sub>2</sub> on large scale tankage. Initial testing for this facility will investigate the effects of initial tank temperatures and in flow rate, as well as provide proof of concept testing for the No-Vent Fill technique. It is hoped that the combined results from the portable and large scale tests will provide the data necessary to determine interfacial heat and mass transfer rates and verify the No-Vent Fill analytical models.

Although ground testing can be used to investigate key phenomena, it is only in the low-gravity environment where the fluid dynamics and heat transfer mechanisms combine to give a true simulation of the Low-Gravity No-Vent Fill process. To investigate the effect of a Low-Gravity environment, several approaches are either under development or being planned. NASA Lewis has had good success with Computational Fluid Dynamic codes to model low-gravity flows. An effort is underway to adapt these codes to the No-Vent Fill problems. NASA Lewis is undertaking studies for the design of a free flying spacecraft called the Cryogenic Orbiting Liquid Depot Satellite (COLD-SAT) to study fluid management issues with liquid hydrogen in space. The study of No-Vent Fill is one of the primary experiments to be conducted on the COLD-SAT. Although a LH<sub>2</sub> experiment such as COLD-SAT is required to understand the technology completely, NASA Lewis is also exploring small scale low-gravity experiments which can provide precursory data for COLD-SAT and allow for a more focused COLD-SAT design.

#### Appendix

To estimate the heat transferred from the tank wall to the ullage vapor by means of solid conduction, the vapor is approximated as a semi-infinite slab of gaseous hydrogen. Schneider gives as the solution for the total heat input for a semi-infinite slab.<sup>7</sup>

$$\frac{Q}{A} = 2k \sqrt{\frac{t}{\pi\alpha}} (T_{\text{wall}} - T_{\text{gas}}) \quad (\text{A1})$$

Gaseous Hydrogen at 101.3 kPa (15 psia) has the following property values:<sup>8</sup>

$$T_{\text{gas}} = 20.268 \text{ K } (36.6 \text{ }^\circ\text{R}) \quad (\text{A2})$$

$$k = 0.01694 \text{ W/K m } (0.00979 \text{ Btu/ft}^2\text{hr } ^\circ\text{R}) \quad (\text{A3})$$

$$\alpha = 0.00375 \text{ m}^2/\text{hr } (0.04037 \text{ ft}^2/\text{hr}) \quad (\text{A4})$$

For a typical fill

$$T_{\text{wall}} = 152.8 \text{ K } (275 \text{ }^\circ\text{R}) \quad (\text{A5})$$

$$t = 0.055 \text{ hr} \quad (\text{A6})$$

(for the duration of the flashing stage.)

Substituting these values into Eq. (A1)

$$\frac{Q}{A} = 9.69 \text{ W hr/m}^2 (3.07 \text{ Btu/ft}^2) \quad (\text{A7})$$

Dividing by the time of the flashing stage to obtain an average heat flux

$$\frac{q}{A} = 176 \text{ W/m}^2 (55.8 \text{ Btu/ft}^2 \text{ hr}) \quad (\text{A8})$$

Brentari and Smith give as the minimum film boiling heat flux for liquid hydrogen a value of approximately:<sup>9</sup>

$$\frac{q}{A} = 8.5 \times 10^3 \text{ W/m}^2 (2.69 \times 10^3 \text{ Btu/ft}^2 \text{ hr}) \quad (\text{A9})$$

The gaseous conduction heat flux is therefore only 2.1 percent of the minimum heat flux to be found in the film boiling process.

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Table 1. - Prototype OTV LH<sub>2</sub> Tankage Characteristics

Mass, kg (lbm)	256 (565)
Volume, m <sup>3</sup> (ft <sup>3</sup> )	43 (1500)
Mass to volume ratio, kg/m <sup>3</sup> (lbm/ft <sup>3</sup> )	6 (0.37)
Inflow rate, kg/hr (lbm/hr)	970 (2000)

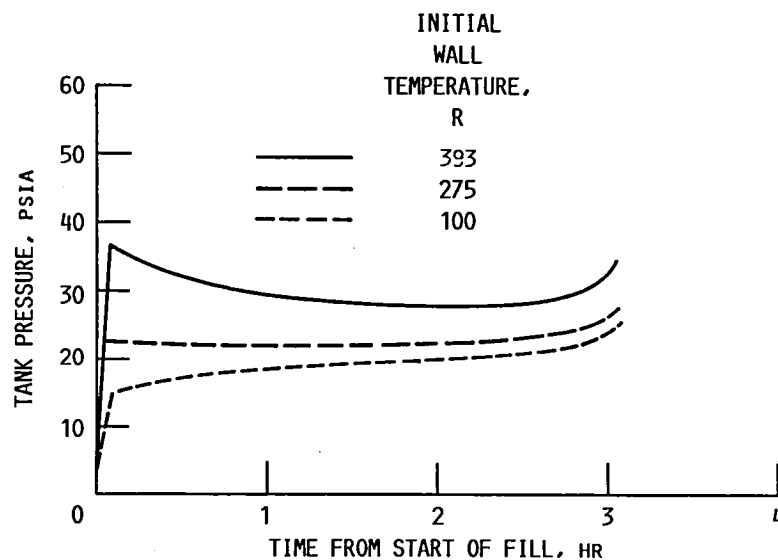


FIGURE 1. - EFFECT OF INITIAL WALL TEMPERATURE ON THE NO-VENT FILL PROCESS. PROTOTYPE OTV LH<sub>2</sub> TANKAGE; TANK VOLUME 1500 CU FT; MASS TO VOLUME, 0.3 LBM/CU FT; INFLOW, 2000 LBM/HR; INFLOW TEMPERATURE, 36.6 R; HEAT TRANSFER COEFFICIENT, 10 BTY/SQ FT-HR-R.



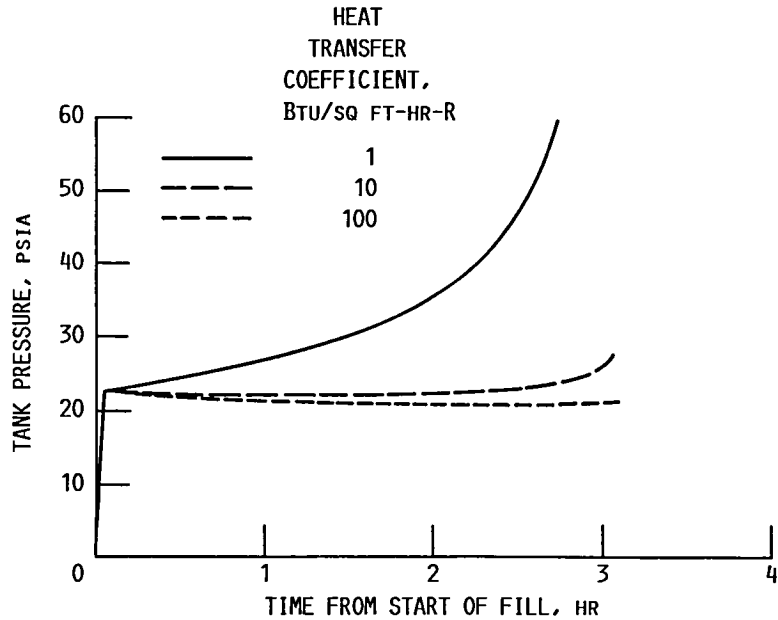


FIGURE 2. - EFFECT OF INTERFACIAL HEAT TRANSFER COEFFICIENT ON THE NO-VENT FILL PROCESS. PROTOTYPE OTV LH2 TANKAGE; TANK VOLUME 1500 CU FT; MASS TO VOLUME 0.3 LBM/CU FT; INFLOW, 2000 LBM/HR; INFLOW TEMPERATURE, 36.6 R; INITIAL TANK WALL TEMPERATURE, 275 R.

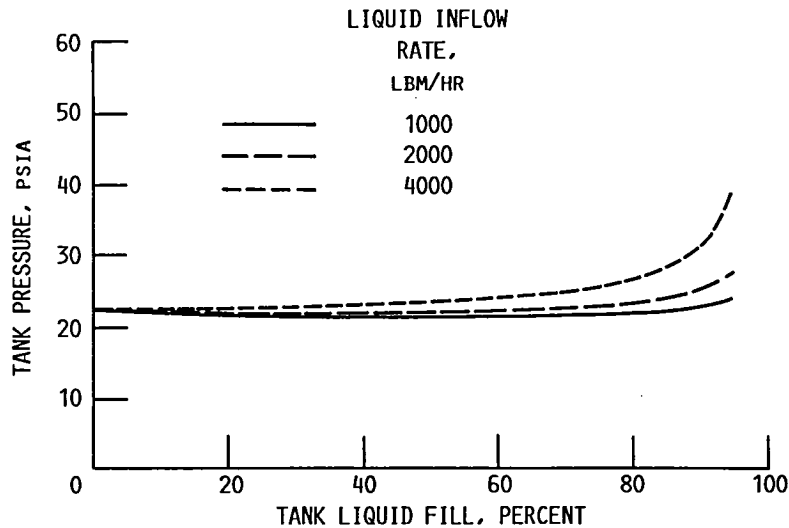


FIGURE 3. - EFFECT OF LIQUID INFLOW RATE ON THE NO-VENT FILL PROCESS. PROTOTYPE OTV LH2 TANKAGE; TANK VOLUME 1500 CU FT; MASS TO VOLUME, 0.3 LBM/CU FT; INITIAL WALL TEMPERATURE, 275 R; INFLOW TEMPERATURE, 36.6 R; HEAT TRANSFER COEFFICIENT, 10 BTU/SQ FT-HR-R.

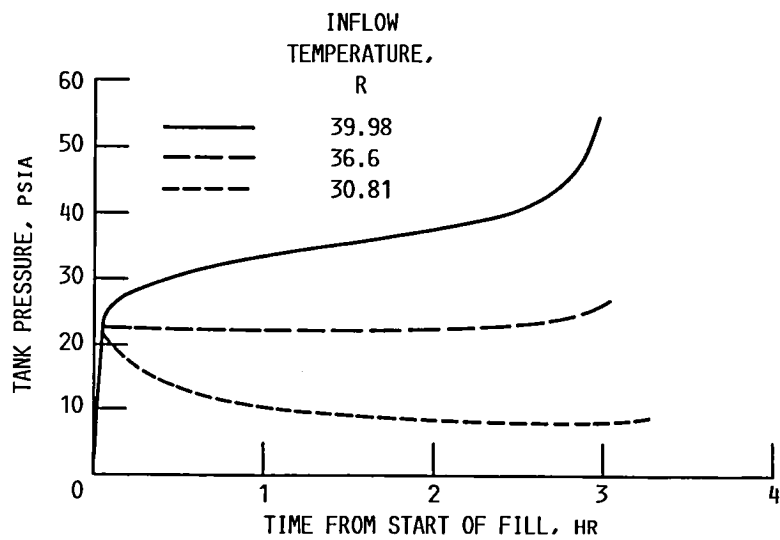


FIGURE 4. - EFFECT OF INFLOWING LIQUID TEMPERATURE ON THE NO-VENT FILL PROCESS. PROTOTYPE OTV TANKAGE; TANK VOLUME 1500 CU FT; MASS TO VOLUME, 0.3 LBM/CU FT; INFLOW, 2000 LBM/HR; INITIAL WALL TEMPERATURE, 275 R; HEAT TRANSFER COEFFICIENT, 10 BTU/SQ FT-HR-R.



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16. Abstract The filling of tanks with cryogenics in the low-gravity environment of space poses many technical challenges. Chief among these is the inability to vent only vapor from the tank as the filling proceeds. As a potential solution to this problem, the NASA Lewis Research Center (NASA Lewis) is researching a technique known as No-Vent Fill. This technology potentially has broad application. The focus of this paper is the fueling of space based Orbital Transfer Vehicles. This paper describes the fundamental thermodynamics of the No-Vent Fill process to develop an analytical model of No-Vent Fill. The model is then used to conduct a parametric investigation of the key parameters; initial tank wall temperature, liquid-vapor interface heat transfer rate, liquid inflow rate and inflowing liquid temperatures. Liquid inflowing temperature and the liquid-vapor interface heat transfer rate seem to be the most significant since they influence the entire fill process. The initial tank wall temperature must be sufficiently low to prevent a rapid pressure rise during the initial liquid flashing stage, but then becomes less significant.					
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