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A PRESSURE CONTROL ANALYSIS OF CRYOGENIC STORAGE SYSTEMS

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Abstract

This paper examines self-pressurization of cryogenic storage tanks due to heat leak through the thermal protection system and the performance of various pressure control technologies for application in microgravity environments. Methods of pressure control such as fluid mixing, passive thermodynamic venting, and active thermodynamic venting are analyzed using the homogeneous thermodynamic model. Simplified equations suggested in the paper may be used to characterize the performance of various pressure control systems and to design space experiments.

Nomenclature

| | |
|-------|---|
| C | coefficient defined in Eq. 9 |
| E | total system energy |
| h | specific fluid enthalpy |
| m | mass flow rate |
| P | tank pressure |
| Q | tank heating rate |
| r_1 | parameter defined in Eq. 15 |
| r_2 | parameter defined in Eq. 17 |
| t | time |
| u | specific internal energy of tank fluid |
| V | tank fluid volume not including volume occupied by internal tank hardware |
| W | power, rate of work done on control volume fluid |

| | |
|-------------|---|
| x | quality |
| ϕ | energy derivative defined in Eq. 5 |
| ρ | average tank fluid density |
| ρ^* | density ratio, $\rho_g/(\rho_f - \rho_g)$ |
| Subscripts: | |
| f | saturated liquid |
| fg | liquid-to-vapor phase change |
| g | saturated vapor |
| hsp | homogeneous self-pressurization |
| i | inlet |
| max | maximum |
| mix | mixer, mixing |
| o | outlet |
| tv | thermodynamic vent system |
| w | wall |

Introduction

The thermal environment of space produces inevitable heat transfer through the thermal protection system of a cryogenic storage tank, which if unchecked, results in a continuous tank pressure rise. The actual pressure rise rate is governed by the complex interaction of external heat leak, fluid temperature stratification, and interfacial heat and mass transfer. If the required storage duration of a space mission is longer than the period in which the tank pressure reaches its allowable maximum, an appropriate pressure control method must be applied.

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Therefore, predictions of the self-pressurization rate and performance of pressure control techniques in cryogenic tanks are required for development of cryogenic fluid long-duration storage technology and planning of future space exploration missions. Pressure control technologies being developed at NASA include high performance thermal insulation, fluid mixing, and both passive and active thermodynamic vent systems (TVS).¹⁻⁴ Schematic representations of self-pressurization and pressure control technologies for application in microgravity environments are illustrated in Fig. 1.

Advanced multilayer thermal insulation systems can greatly reduce the heat leak to a cryogenic storage tank. Vapor cooled shields are sometimes utilized to further intercept incoming thermal energy. Fluid thermal stratification generally augments the pressure rise rate of a tank subjected to a wall heat flux (Fig. 1(a)). Fluid mixing (Fig. 1(b)) can be used to circulate and mix the tank fluid destroying fluid temperature stratification. Mixing induces interfacial condensation, resulting in the reduction of tank pressure. Interface condensation plays a key role in controlling the rate of change of tank pressure. Ideally, if a tank is well mixed, the fluid temperature will be uniform and the homogeneous thermodynamic model, which assumes uniform temperature throughout the tank, may be used for the prediction of pressure change rate. Since mixer power is required to circulate the tank fluid, a certain amount of energy is added to the system. This additional energy imparted to the fluid eventually becomes heat and increases the net fluid energy. Thus, fluid mixing can only temporarily reduce tank pressure and is of interest for short-term storage. The design of the mixer influences the efficiency and performance of a fluid mixing pressure control system.

In the passive TVS concept (Fig. 1(c)) a small amount of tank liquid is withdrawn (continuously or intermittently) and passed through a Joule-Thomson device, resulting in a lower pressure and temperature two-phase fluid (since cryogenics near saturation have a positive Joule-Thomson coefficient). Depending on the design of the TVS heat exchanger, this two-phase mixture is then directed to a heat exchanger to either cool the tank fluid directly as shown in Fig. 1(c) via passive energy exchange (thermal conduction and/or free convection) and/or to intercept the heat leak (vapor cooled shields) into the tank. The fluid mixture is evaporated in the heat exchanger and the resulting vapor is then vented overboard. With this method, energy is removed from the system by sacrificing some of the liquid contents of the tank. Thus, the passive TVS may be used in a long-term storage system. However, an undesirable weight penalty may occur, due to a large heat exchanger surface area required to vaporize the vented fluid. Also, the relatively slow passive heat transfer processes on the tank side of the heat exchanger may lead to unacceptably long lag times during attempted pressure control.

The TVS and fluid mixing techniques may be combined to produce an efficient and fast response pressure control system. This approach is called the active TVS. In the specific configuration shown in Fig. 1(d), the TVS heat exchanger is designed to remove energy from the mixer flow; the resulting subcooled liquid is then circulated back to the tank through a nozzle to mix the tank fluid and induce condensation. An optimal design of the heat exchanger and mixer is quite important for proper active TVS performance.

This report presents an analysis of tank self-pressurization and the aforementioned pressure control methods. Simplified forms of the homogeneous model are given to provide a design tool for space experiments. Practical examples are also given to characterize the performance of pressure control systems.

Homogeneous Model

A general form of the model for a cryogenic storage tank can be obtained by applying the first law of thermodynamics and conservation of mass to a control volume that contains the tank liquid-vapor contents, assumed to be in a homogeneous state. Internal tank hardware is not included in the control volume. The control surface corresponds to the inside surface of the tank wall. The first law for a variable volume with one

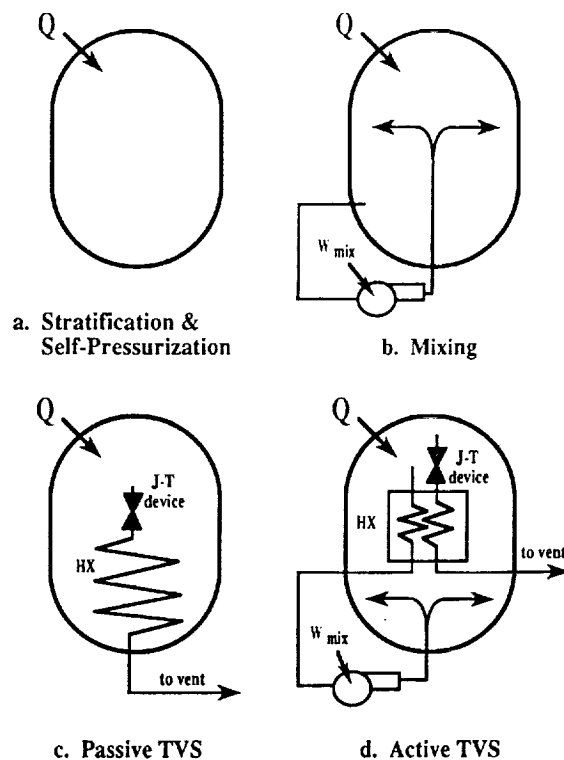


Figure 1 - Schematic representation of self-pressurization and pressure control technologies (mixer shown outside of tank for clarity).

inlet and one outlet (neglecting kinetic and potential energy terms) is:

$$\begin{aligned} \frac{dE}{dt} &\equiv \rho V \frac{du}{dt} + \rho u \frac{dV}{dt} + V u \frac{d\rho}{dt} \\ &= Q + W + m_i h_i - m_o h_o - P \frac{dV}{dt} \end{aligned} \quad (1)$$

Power input to the control volume (e.g., mixing) is accommodated using a positive value for W .

Conservation of mass for a compression or expansion process is expressed as:

$$\frac{dm}{dt} = m_i - m_o = V \frac{d\rho}{dt} + \rho \frac{dV}{dt} \quad (2)$$

If specific internal energy is considered as a function of density and pressure, $u = u(\rho, P)$, then:

$$\frac{du}{dt} = \left(\frac{\partial u}{\partial \rho} \right)_P \frac{d\rho}{dt} + \left(\frac{\partial u}{\partial P} \right)_\rho \frac{dP}{dt} \quad (3)$$

The general-purpose homogeneous equation for pressure change rate is obtained by combining the above equations and using the relationship $h = u + P/\rho$:

$$\begin{aligned} \frac{dP}{dt} &= \frac{\phi}{V} \left\{ Q + W + m_i \left[h_i - h - \rho \left(\frac{\partial h}{\partial \rho} \right)_P \right] \right. \\ &\quad \left. - m_o \left[h_o - h - \rho \left(\frac{\partial h}{\partial \rho} \right)_P \right] + \rho^2 \left(\frac{\partial h}{\partial \rho} \right)_P \frac{dV}{dt} \right\} \end{aligned} \quad (4)$$

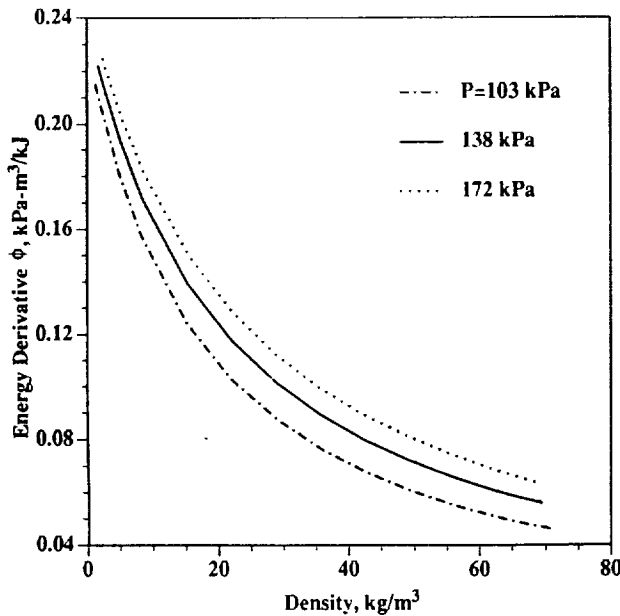


Figure 2 - Functional dependence of energy derivative on density and pressure for LH_2 .

where the energy derivative, ϕ , is defined as:

$$\phi = \frac{1}{\rho \left(\frac{\partial u}{\partial P} \right)_\rho} \quad (5)$$

The total pressure change is obtained by integrating an appropriate form of Eq. 4 using a known initial condition.

Figure 2 shows the parameter ϕ as a function of average fluid density and tank pressure for hydrogen at saturated, two-phase conditions. It is seen that ϕ decreases with increasing average fluid density, ρ , and increases with increasing tank pressure, P . Energy derivatives for other cryogenics such as nitrogen and oxygen exhibit similar behavior.

Self-Pressurization

For self-pressurization of a closed constant volume cryogenic tank experiencing heat leak only, Eq. 4 reduces to:

$$\left(\frac{dP}{dt} \right)_{hsp} = \frac{\phi Q_w}{V} \quad (6)$$

Given a fixed volumetric heating rate, Q_w/V , the homogeneous model shows that the self-pressurization rate is increasing with time because the energy derivative ϕ increases with increasing tank pressure. Usually, the homogeneous model, which assumes a uniform temperature in the tank, will give the lowest pressure rise rate of a tank. Ground tests with liquid hydrogen (LH_2) have shown that the actual pressure rise rate spans a range from approximately one⁵ to more than ten⁶ times faster than the homogeneous model prediction for well-insulated tanks. In low-gravity experiments,⁷ the pressure rise rate has been found to be slower due to reduced buoyancy and an increase in the liquid-wetted wall area. The homogeneous rate is a convenient baseline, as it represents the slowest pressure rise rate in most normal-gravity situations.

Equation 6 indicates that the pressure rise rate linearly increases with the tank volumetric heating rate, Q_w/V , and the parameter ϕ . For the purpose of illustration, a self-pressurization rate of two times the homogeneous prediction is assumed as a characteristic rate for the examples provided herein. Figure 3 shows the pressure rise of a spherical 52 m³ LH_2 tank (a typical size for future spacecraft), initially at 101 kPa, at various fill levels (by volume) and heating rates. The range of heating rate corresponds to the expected heat flux range in full scale applications. It is seen that lower liquid fill level (yielding smaller ρ and larger ϕ) and higher heating rate (yielding greater Q_w/V) result in a greater tank pressure rise rate. For 95 percent fill level, if the wall heating rate is reduced from 100 to 20 W, the mission duration can be increased from 2.1 to 10.7 days, assuming the maximum allowable tank pressure is 138 kPa. This implies that a well-designed thermal protection system can provide a significant extension of the

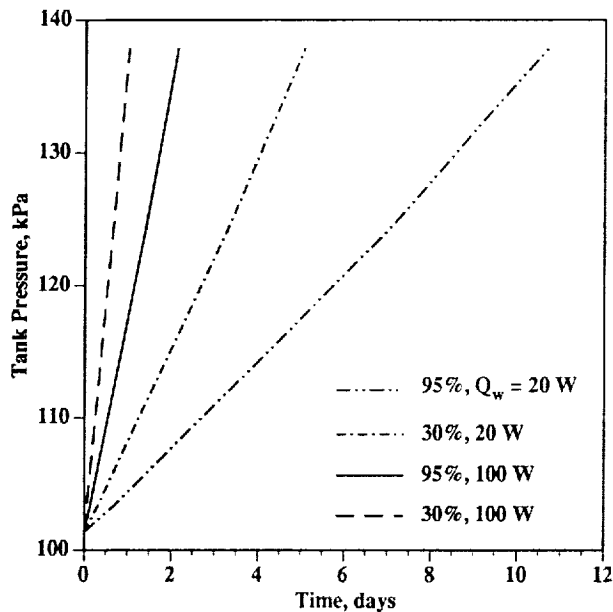


Figure 3 - Pressure rise dependence on fill level and heat leak (two times homogeneous pressure rise rate, $V=52 \text{ m}^3$).

storage duration. This benefit occurs at all fill levels and is enhanced if the allowable maximum tank pressure rise is larger.

Fluid Mixing

Mixing, such as that induced by an axial jet, may be used to extend the storage duration. As the tank fluid is mixed, thermal stratification is reduced and interfacial condensation is promoted. It is assumed that the entire tank can be completely mixed such that a homogeneous state can be achieved. When the fluid temperature becomes uniform, condensation ceases and continued operation of the mixer will not produce a further decrease in tank pressure.

Two possible modes of operation are continuous and intermittent mixing. In the continuous mixing mode, a well-mixed homogeneous state is maintained. Intermittent mixing is a cyclic process in which fluid stratification occurs until a specified operating pressure is reached followed by a mixing period that lasts until a homogeneous state is achieved. The cycle repeats until the pressure at the homogeneous state approaches the maximum allowable pressure. In either mode, operation of the mixer results in heat input due to dissipation of the mixer power that augments the pressure rise rate (as compared to the homogeneous self-pressurization rate). Thus, the mixer power, W_{mix} (corresponding to the required jet flow rate), and the mixing time, t_{mix} , are two key variables governing mixing performance. For intermittent mixing, mixing time is defined as the period beginning when the mixer is turned on to when the tank pressure reaches the value corresponding to a homogeneous state. For an efficient mixing system, the total energy input by the mixer, $W_{\text{mix}} t_{\text{mix}}$, should be kept as low as possible.

Equation 4 when applied to a mixing operation in a fixed volume tank with recirculating fluid reduces to:

$$\frac{dP}{dt} = \frac{\phi}{V}(Q_w + W_{\text{mix}}) = \frac{\phi Q_w}{V} \left(1 + \frac{W_{\text{mix}}}{Q_w}\right) \quad (7)$$

It predicts $dP/dt > (dP/dt)_{\text{hsp}}$. Therefore, if the tank is already in a homogeneous state, fluid mixing will result in an increase in pressure because of the additional heat input from the mixer. If the mixer power-to-heating rate ratio is large, the pressure rise rate with continuous mixing could be higher than that of stratified self-pressurization. A large mixer power-to-heating rate ratio could occur due to poor mixer design and/or very high performance thermal protection systems. Equation 7 assumes the recirculating fluid does not exit and re-enter the control volume. For an external pump, W_{mix} would be replaced by $m_{\text{mix}}(h_{\text{mix},i} - h_{\text{mix},o})$ in the equation. It is possible to analyze various mixing processes combined with self-pressurization periods by the combined use of Eqs. 6 and 7 as is discussed in the following example.

Several mixing situations that may occur are considered below. In each situation the tank is initially 95 percent full and saturated at 101 kPa. Homogenous self-pressurization from 101 kPa to a specified operating pressure of 138 kPa would require 21.4 days as shown in Fig. 4, and is considered the maximum possible duration for storage without venting.

(1) If the mixer is off, $W_{\text{mix}} = 0$, tank pressure rises at the assumed rate of two times $(dP/dt)_{\text{hsp}}$ and will reach the operating pressure of 138 kPa in 10.7 days as shown in Fig. 4.

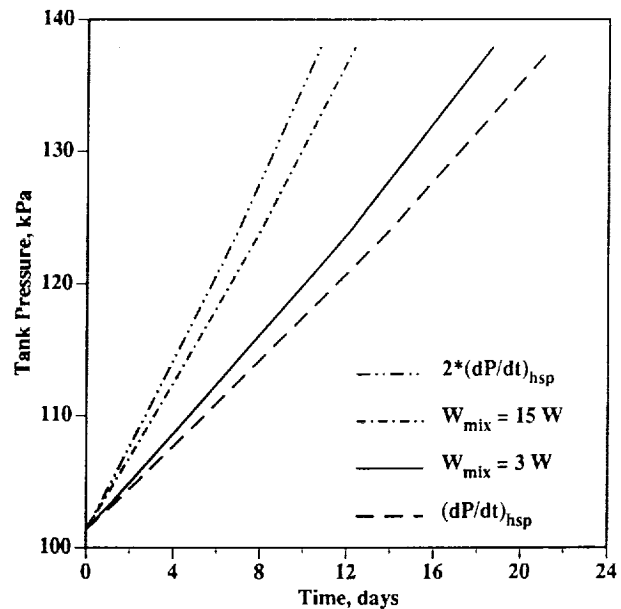


Figure 4 - Self-pressurization and the effect of continuous mixing (95% fill, $Q_w=20 \text{ W}$, $V=52 \text{ m}^3$).

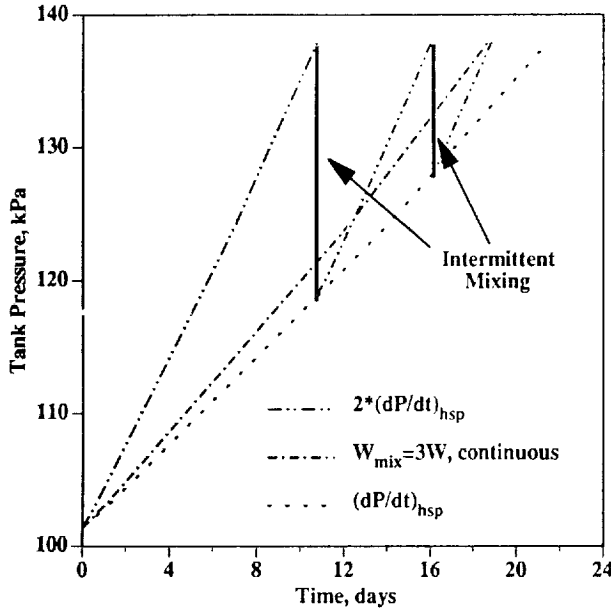


Figure 5- Comparison of intermittent and continuous mixing (95% fill, $Q_w=20$ W, $V=52\text{m}^3$).

(2) If the mixer runs continuously at $W_{\text{mix}} = 15$ W, tank pressure will reach 138 kPa in 12.3 days, as shown in Fig. 4. This is only 1.6 days longer than case (1).

(3) If the mixer runs continuously at a lower power, such as $W_{\text{mix}} = 3$ W, 18.7 days will elapse before the pressure reaches 138 kPa as shown in Fig. 4. This is 8.0 days longer than case (1). It has been assumed that homogeneous conditions can be attained at the lower mixer power.

(4) Based on available ground-based mixing time correlations,^{8,9} the mixing time for the system under consideration is less than 2 hr for a wide range of jet flow rates. Since the mixing time is quite short, the total energy input by the mixer, $W_{\text{mix}}t_{\text{mix}}$, is negligibly small compared with the tank internal energy. Thus, the tank pressure is reduced to about 119 kPa (homogeneous condition) after the first mixing cycle as shown in Fig. 5. If the mixer is run intermittently, two mixing cycles are required to achieve a similar storage duration as for case (3).

The choice of intermittent versus continuous mixing is determined by comparing the energy inputs for each method. Considering that the power required to achieve a well-mixed state from a thermally stratified state is not likely to be substantially higher than that needed to maintain homogeneous conditions, the intermittent mode is expected to be more efficient.

In reality, when the mixer is turned on, interface condensation begins and tank pressure starts to decrease. As the fluid is mixed to a certain degree, the pressure reaches a minimum value that is higher than the corresponding value for the homogeneous self-pressurization process. At this moment,

the competing effects of wall heat flux (increasing tank pressure) and induced interface condensation (decreasing tank pressure) on the pressure change become balanced. As the mixer continues to operate, the rate of tank pressure change becomes positive, due to diminishing interfacial condensation. Also, the mixer power input generates heat, inducing liquid evaporation instead of condensation as homogeneous conditions are neared. Therefore, mixing time should be established as the time period from when the mixer is turned on to when the tank pressure reaches a minimum. Over- or under-prediction of the mixing time and/or mixer power will affect mixing performance efficiency.

Passive Thermodynamic Vent System

A constant volume tank of homogeneous fluid subject to either direct venting or passive TVS operation is governed by the following form of Eq. 4:

$$\begin{aligned} \frac{dP}{dt} &= \frac{\phi}{V} \left\{ Q - m_o \left[h_o - h - \rho \left(\frac{\partial h}{\partial \rho} \right)_P \right] \right\} \\ &= \frac{\phi}{V} \left[Q_w - Q_{\text{TVS}} - m_{\text{TVS}} h_{\text{fg}} (x + \rho^*) \right] \end{aligned} \quad (8)$$

where x is the quality of the fluid entering the J-T device, $\rho^* = \rho_g / (\rho_f - \rho_g)$, and Q_{TVS} is the heat removal rate of the TVS heat exchanger. A work term is not included since the vent flow is passively driven by the tank-to-vent pressure drop. If vapor cooled shields are utilized, the vent flow will alter the heat leak rate, Q_w . If $Q_{\text{TVS}} = 0$, then Eq. 8 represents the homogeneous state of direct venting with $x = 1$ for saturated vapor only and $x = 0$ for saturated liquid only.

For normal TVS operation, liquid is withdrawn from the tank through a liquid acquisition device such that the fluid entering the J-T device has a quality $x = 0$. The heat removal rate by the lower pressure two phase flow as the fluid passes through the heat exchanger can be expressed as:

$$Q_{\text{TVS}} = m_{\text{TVS}} \Delta h_{\text{TVS}} = C m_{\text{TVS}} h_{\text{fg}} \quad (9)$$

In Eq. 9, C is set equal to $\Delta h_{\text{TVS}} / h_{\text{fg}}$ where Δh_{TVS} is the change in enthalpy in the TVS heat exchanger and h_{fg} is the latent heat of vaporization corresponding to the tank pressure. The coefficient C is a system parameter and its value depends on the performance of the TVS heat exchanger, J-T device, and the tank and vent pressures. For a properly designed space-based system typical values of C may vary from 0.8 to 1.1. Thus, Eq. 8 becomes:

$$\begin{aligned} \frac{dP}{dt} &= \frac{\phi}{V} Q_w \left[1 - \frac{m_{\text{TVS}} h_{\text{fg}}}{Q} (C + \rho^*) \right] \\ &= \left(\frac{dP}{dt} \right)_{\text{hsp}} \left[1 - \frac{m_{\text{TVS}} h_{\text{fg}}}{Q} (C + \rho^*) \right] \end{aligned} \quad (10)$$

The above equations may be used to compare the performance of a passive TVS to an ideal TVS for an isothermal fluid system. In an ideal TVS, the vapor leaves the heat exchanger at a temperature equal to the tank fluid temperature, i.e., Δh_{TVS} is at a maximum. For a LH_2 system the maximum value of C (an ideal system) can be approximated by:

$$C_{\text{max}} \equiv 1 + 4.3 \times 10^{-4} \Delta P \quad (\text{for } P < 400 \text{ kPa}) \quad (11)$$

where ΔP (in kPa) is the difference between the tank and TVS pressures.

Equation 10 is useful for specifying the range of vent flow rates required to maintain constant pressure. Figure 6 shows that for a given pressure change rate, the required TVS flow rate decreases with an increasing value of the coefficient C .

In passive TVS operation, the conduction- and/or natural convection-dominated heat transfer process leads to a slow tank pressure response to the system operation. In reality, a significant amount of TVS flow will be vented before a reduction in tank pressure occurs. In addition, the tank fluid will be thermally stratified. Thus, the TVS flow rate and the total vented mass obtained from Eq. 10 are expected to be lower than actual requirements.

Active Thermodynamic Vent System

Assuming that a constant volume propellant tank is in a homogeneous state when the active TVS pressure control method is applied, the homogeneous equation can be simplified and used to predict the rate of change in tank pressure:

$$\frac{dP}{dt} = \frac{\phi}{V} [Q_w + W_{\text{mix}} - m_{\text{TVS}} \Delta h_{\text{TVS}} - m_{\text{TVS}} h_{\text{fg}} \rho^*] \quad (12)$$

where it has been assumed that saturated liquid is removed via a liquid acquisition device. It is noted that $W_{\text{mix}} - m_{\text{TVS}} \Delta h_{\text{TVS}}$ may be replaced by $-m_{\text{mix}}(h_{\text{mix},o} - h_{\text{mix},i})$.

To maintain constant tank pressure, the required energy removal rate by the TVS heat exchanger is:

$$(m_{\text{TVS}} \Delta h_{\text{TVS}})_{dP/dt=0} = Q_w + W_{\text{mix}} - m_{\text{TVS}} h_{\text{fg}} \rho^* \quad (13)$$

Thus, Eq. 12 can be expressed as:

$$\frac{dP}{dt} = \frac{\phi}{V} Q_w \left[1 + \frac{W_{\text{mix}}}{Q_w} - \frac{m_{\text{TVS}} h_{\text{fg}} \rho^*}{Q_w} \right] (1 - r_1) \quad (14)$$

where the parameters, r_1 is defined as:

$$r_1 = \frac{m_{\text{TVS}} \Delta h_{\text{TVS}}}{(m_{\text{TVS}} \Delta h_{\text{TVS}})_{dP/dt=0}} \quad (15)$$

The parameter r_1 represents the ratio of total heat actually removed from the mixer fluid to the total heat required to be removed in order to have $dP/dt = 0$. Several observations can be made regarding Eq. 14:

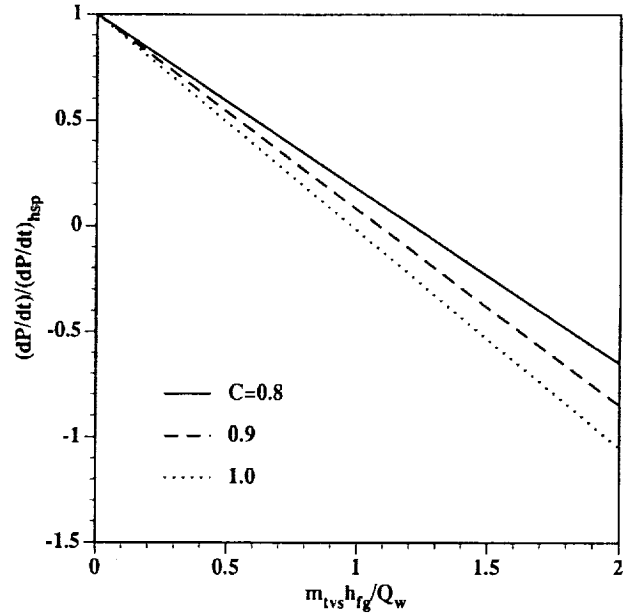


Figure 6 - Passive TVS Performance For Liquid Hydrogen at 138 kPa.

(1) If $m_{\text{TVS}} = 0$ ($r_1 = 0$) and $W_{\text{mix}} > 0$, then fluid mixing only is performed and Eq. 14 will yield Eq. 7.

(2) If $m_{\text{TVS}} > 0$ ($r_1 > 0$) and $W_{\text{mix}} = 0$, then passive TVS only is performed yielding Eq. 10.

(3) If both $m_{\text{TVS}} > 0$ ($r_1 > 0$) and $W_{\text{mix}} > 0$, then active TVS is performed.

Equation 14 is functionally equivalent to:

$$\frac{dP/dt}{\phi Q_w / V} = f \left[r_1, \frac{W_{\text{mix}}}{Q_w}, \frac{m_{\text{TVS}} h_{\text{fg}} \rho^*}{Q_w} \right] \quad (16)$$

Based on Eq. 16, the performance of an active TVS can be evaluated as a function of relevant parameters. Figure 7 shows the effect of r_1 on the pressure change rate for $W_{\text{mix}}/Q_w = 0.05$ and 0.5 and $m_{\text{TVS}} h_{\text{fg}} \rho^* / Q_w$ equal to 0.05 and 0.1 which are representative values for spacecraft tanks. If the actual heat removed by the TVS heat exchanger has a 50 percent variation from the required value for $dP/dt = 0$, (i.e., $r_1 = 0.5$ or 1.5), then $dP/dt = \pm 50$ and ± 75 percent for $W_{\text{mix}}/Q_w = 0.05$ and 0.5, respectively. The sensitivity of $(dP/dt)/(dP/dt)_{\text{hsp}}$ to r_1 increases as W_{mix}/Q_w increases. Thus, lower jet flow rate and/or improvement of mixer efficiency, resulting in lower required mixer power, gives improved mixing performance.

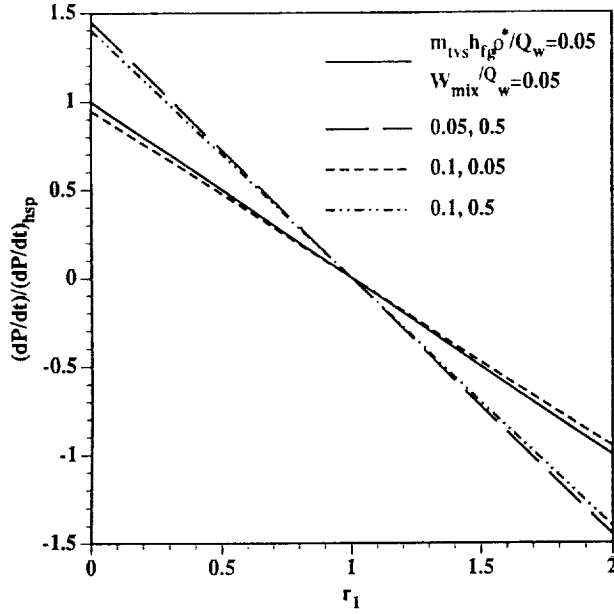


Figure 7 - Effect of r_1 on pressure change rate

The required energy removal rate specified by Eq. 13 to maintain $dP/dt = 0$ includes that required to offset heat addition from the mixer. A parameter, r_2 , representing the fraction of the total TVS mass flow rate used to remove this mixer-generated heat can be defined as:

$$r_2 = \left[\frac{W_{\text{mix}}}{Q_w + W_{\text{mix}} - m_{\text{tvs}} h_{\text{fg}} \rho^*} \right]_{dP/dt=0} \quad (17)$$

Figure 8 shows the effect of the mixer power-to-volumetric heating ratio, W_{mix}/Q_w , on the parameter r_2 . It is observed that the effect of the ratio, $m_{\text{tvs}} h_{\text{fg}} \rho^*/Q_w$, on r_2 is quite small for a wide range of mixer power input. However, for a given TVS flow rate, r_2 increases significantly with increasing W_{mix}/Q_w . For example, $W_{\text{mix}}/Q_w = 0.05$ has a value of r_2 equal to 0.04 while the value of r_2 for $W_{\text{mix}}/Q_w = 0.5$ is greater than or equal to 0.3. The design of an active TVS should have a value of W_{mix}/Q_w as low as possible.

Combining Eqs. 9 and 12 gives:

$$\begin{aligned} \frac{dP}{dt} &= \frac{\phi Q_w}{V} \left[1 + \frac{W_{\text{mix}}}{Q_w} - \frac{m_{\text{tvs}} h_{\text{fg}}}{Q_w} (C + \rho^*) \right] \\ &= \left(\frac{dP}{dt} \right)_{\text{hsp}} \left[1 + \frac{W_{\text{mix}}}{Q_w} - \frac{m_{\text{tvs}} h_{\text{fg}}}{Q_w} (C + \rho^*) \right] \end{aligned} \quad (18)$$

It is observed from Eqs. 10 and 18 that, based on the homogeneous model, more TVS mass flow is vented by an active TVS than by a passive TVS due to the additional heat input by mixer power provided C is equal for both systems. However, in the active TVS operation, the heat transfer process is due to forced convection which has a much faster time response than the conduction/natural convection processes in the passive TVS. Also, tank thermal stratification is largely

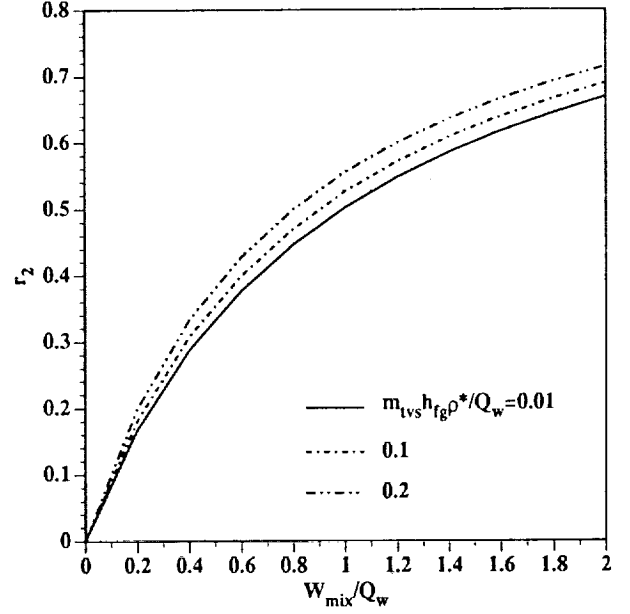


Figure 8 - Effect of mixer power on the parameter r_2

destroyed in active TVS, but not in passive TVS. Therefore, in reality, the homogeneous model is more appropriate for an active TVS than for a passive TVS. If the mixer power is not exceptionally large, the active TVS should be more efficient than a passive TVS. It is noted that in the design of an active TVS pressure control experiment, one can evaluate the order of magnitude for each term on the right hand side of Eq. 18.

Conclusions

An analysis of the self-pressurization of a cryogenic storage system and the performance of pressure control devices such as fluid mixing, passive TVS, and active TVS has been presented. Simplified equations based on homogeneous models are given for the prediction of pressure change rate for a self-pressurizing tank with or without a pressure control device. Several conclusions are drawn:

1. The pressure change rate of a tank generally increases with tank volumetric heating rate, Q_w/V .

2. Pressure rise rate can be temporarily reduced by fluid mixing, an appropriate method for moderate extension of short-term storage duration. An optimal design of the mixer device is one which has the lowest value of the product of mixer power and mixing time, $W_{\text{mix}} t_{\text{mix}}$, for a given tank pressure reduction rate. Intermittent mixing is preferred over continuous mixing. The mixing time and mixing power of a given system should be accurately predicted to obtain an efficient pressure control procedure (i.e., to minimize $W_{\text{mix}} t_{\text{mix}}$).

3. For a given pressure change rate in a passive or active TVS, the required flow rate decreases with increasing $C(=\Delta h_{\text{TVS}}/h_{\text{fg}})$. An active TVS design should have a value of W_{mix}/Q_w as low as possible such that the fraction of total TVS flow rate required to remove the heat generated by the mixer is minimized. The determination of actual TVS heat exchanger and mixing device effectiveness are needed to predict the values of $m_{\text{TVS}}\Delta h_{\text{TVS}}$ and $m_{\text{mix}}\Delta h_{\text{mix}}$, respectively, required for either maintaining constant tank pressure ($dP/dt = 0$) or reducing the tank pressure at a desired rate.

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| 16. Abstract This paper examines self-pressurization of cryogenic storage tanks due to heat leak through the thermal protection system and the performance of various pressure control technologies for application in microgravity environments. Methods of pressure control such as fluid mixing, passive thermodynamic venting, and active thermodynamic venting are analyzed using the homogeneous thermodynamic model. Simplified equations suggested in the paper may be used to characterize the performance of various pressure control systems and to design space experiments. | | | | | |
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