

# LH<sub>2</sub> tank pressure control by thermodynamic vent system (TVS) at zero gravity

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**Abstract.** Thermodynamic vent system (TVS) is employed for pressure control of propellant tanks at zero gravity. An analytical lumped parameter model is developed to predict pressure variation in an 18.09 m<sup>3</sup> liquid hydrogen tank equipped with TVS. Mathematical simulations are carried out assuming tank is filled up to 75% volume (liquid mass equals to 945 kg) and is subjected to heat flux of 0.76 W/m<sup>2</sup>. Tank pressure controls at 165.5-172.4, 165.5-179.3 and 165.5-182.2 kPa are compared with reference to number of vent cycles, vent duration per cycle and loss of hydrogen. Analysis results indicate that the number of vent cycles significantly decreases from 62 to 21 when tank pressure control increases from 6.9 to 20.4 kPa. Also, duration of vent cycle increases from 63 to 152 and cycle duration decreases from 3920 to 3200 s. Further, the analysis result suggests that LH<sub>2</sub> evaporation loss per day decreases from 0.17 to 0.14%. Based on the results of analysis, TVS is found effective in controlling the propellant tank pressure in zero gravity.

## 1. Introduction

Cryogenic propellants are applied gradually in space missions due to their high fuel performance and nontoxicity. Deep space exploration calls for long term storage technologies of cryogenic propellants on orbit. The heat leak into the tanks from the ambient causes the cryogenics boiling off which will definitely lead to increase of the internal pressure. Pressure control is one of the enabling technologies for storing two-phase cryogenics at zero gravity. A traditional solution for propellant venting is to employ auxiliary systems to separate the liquid and vapor by accelerating or rotating the tank. However, it must incur weight penalties and increase mission complexity. Thermodynamic vent system (TVS) was proposed by NASA as an effective technology for pressure control and venting of propellant without resettling [1]. However, the first generation TVS played a limited role in eliminating the fluid thermal stratification due to its structural constraint. The second generation TVS [2] was proposed by Marshall Space Flight Center (MSFC) and Rockwell® in 1993. It replaced the compact heat exchanger by a spray bar heat exchanger and also introduced a mixing pump to accomplish a better destratification effect. Heat was extracted from the tank contents without significant liquid losses. Lin *et al* [3] simulated a 52 m<sup>3</sup> LH<sub>2</sub> tank which only equipped with a mixer by homogeneous thermodynamic model. He obtained the behavior curves of the self-pressurization and destratification processes inside the tank. The performance of the spray bar TVS for liquid hydrogen (LH<sub>2</sub>) and liquid nitrogen (LN<sub>2</sub>) were experimentally investigated at MSFC in 1996 and 1998 respectively [4-6]. The experiments validated that the TVS could control tank pressure amplitude within a prescribed value of 6.9 kPa. Experimental study on TVS in liquid oxygen (LO<sub>2</sub>) tank was conducted at Glenn Research Center (GRC) in 2008[7]. Their results showed that the tank pressure



**Nomenclature**

TVS	thermodynamic vent system	$(\dot{m}h)_s$	heat flow of the spray fluids (W)
$m_u$	ullage mass (kg/s)	$h_g$	enthalpy of saturated vapor (J/kg)
$\dot{m}_{du}$	droplet evaporation rate (kg/s)	$w_u$	work done on the ullage (W)
$\dot{m}_{lu}$	bulk liquid boiling rate (kg/s)	$q_e$	energy added to the ullage (W)
$\dot{m}_{bw}$	boiling rate of wall liquid (kg/s)	$p_u$	Ullage (vapor) pressure (kPa)
$h_{fg}$	latent heat of liquid nitrogen (J/kg)	$V_l$	volume of the liquid (m <sup>3</sup> )
$\dot{m}_c$	condensation rate on liquid-vapor equilibrium surface (kg/s)	$V_u$	volume of the vapor (m <sup>3</sup> )
$\dot{m}_{ul}$	condensation rate within the ullage region (kg/s)	$V_{wl}$	volume of the wall liquid (m <sup>3</sup> )
$q_{ud}$	heat transfer rate between the ullage and the droplet (W)	$N_t$	transient rotation speed of the pump (r/min)
$\dot{m}_{su}$	mass flow rate of the spray fluids (kg/s)	$n_f$	rated speed of the pump (r/min)
$c_{vu}$	specific heat at constant volume of the vapor (J/(kg K))	$H_l$	head of the pump (m)
$t_{sat}$	saturation temperature of the vapor (K)	$p_1, p_2$	inlet and outlet pressure of the pump (kPa)
$t_s$	temperature of spray fluids (K)	$g$	gravity acceleration (m/s <sup>2</sup> )
$h_f$	latent heat of vaporization (J/kg)	$M_u$	average molar mass of the ullage (g/mol)
$q_l$	energy added to the liquid (W)	$R$	ideal gas constant (J/(mol K))
$q_{uwl}$	heat transfer rate between the ullage and the wall liquid (W)	<b>subscript</b>	
$\dot{m}_{wl}$	mass flow rate of wall liquid (kg/s)	$u, l$	Ullage (vapor), liquid
$t_{wl}$	temperature of wall liquid (K)	$w$	wall
$q_{ul}$	heat transfer rate between the ullage and liquid (W)	$iv$	initial condition of vapor
$q_{wu}$	heat transfer rate between the tank wall and ullage (W)	$s$	spray
$q_{us}$	heat transfer rate between the ullage and spray bars (W)	$f$	vaporization of fluids

amplitude could be controlled within 3.45 kPa. The question is, how about applying higher limit range of the control pressure if the tank allows higher operation pressure? Therefore, the pressure control range (set value) is an important parameter of interest for the purpose of further reducing propellant venting loss from the tank operated with TVS.

This paper proposes a lumped parameter model for predicting pressure variation in a TVS equipped liquid hydrogen tank with a volume of 18.09 m<sup>3</sup> and a surface area of 35.74 m<sup>2</sup>, and to verify the pressure control performance of TVS. The effects of pressure control band on cycle durations, running numbers and vapor mass loss etc. will be discussed.

**2. Analytical model**

Spray bar TVS system consisting of spray manifold, cryogenic pump, throttling valve, cut-off valves and interconnection pipelines is shown in Fig. 1. Thermal behavior of all the system elements is modeled for system simulations. A lumped parameter model is used to simulate the thermal behavior of cryogenic fluid contained in the tank. The ullage mass  $m_u$  is given by

$$m_u = (m_u)_{iv} + \int \left( \frac{dm_u}{dt} \right) dt \quad (1)$$

The mass changing rate is

$$\frac{dm_u}{dt} = \dot{m}_{du} + \dot{m}_{lu} + \dot{m}_{bw} - \dot{m}_c - \dot{m}_{ul} \quad (2)$$

where  $\dot{m}_{du}$  is droplet evaporation rate,  $\dot{m}_{lu}$  is bulk liquid boiling rate,  $\dot{m}_{bw}$  is boiling rate of wall liquid,  $\dot{m}_c$  is condensation rate on liquid-vapor equilibrium surface, and  $\dot{m}_{ul}$  is condensation rate within the ullage region.

The droplet evaporation rate is

$$\dot{m}_{du} = (q_{ud} - \dot{m}_{su} \cdot c_{vu} \cdot (t_{sat} - t_s)) / h_f \quad (3)$$

where  $q_{ud}$  is the heat transfer rate between the ullage and the droplet,  $\dot{m}_{su}$  is the mass flow rate of the spray fluids,  $c_{vu}$  is the specific heat at constant volume of the vapor,  $t_{sat}$  is the saturation temperature of the vapor,  $t_s$  is the temperature of spray fluids,  $h_f$  is the latent heat of vaporization.

The bulk liquid boiling rate is

$$\dot{m}_{lu} = q_l / h_f \quad (4)$$

where  $q_l$  is the energy added to the liquid.

The boiling rate of liquid layer on the tank wall is

$$\dot{m}_{bw} = (q_{uwl} - \dot{m}_{wl} \cdot c_{vu} \cdot (t_{wl} - t_{sat})) / h_f \quad (5)$$

where  $q_{uwl}$  is the heat transfer rate between the ullage and the wall liquid,  $\dot{m}_{wl}$  is the mass flow rate of wall liquid,  $t_{wl}$  is the temperature of wall liquid.

The condensation mass rate on liquid-vapor equilibrium surface is

$$\dot{m}_c = q_{ul} / h_f \quad (6)$$

where  $q_{ul}$  is the heat transfer rate between the ullage and liquid.

The condensation rate within the ullage region is

$$\dot{m}_{ul} = (q_{ud} + q_{uwl}) / h_f \quad (7)$$

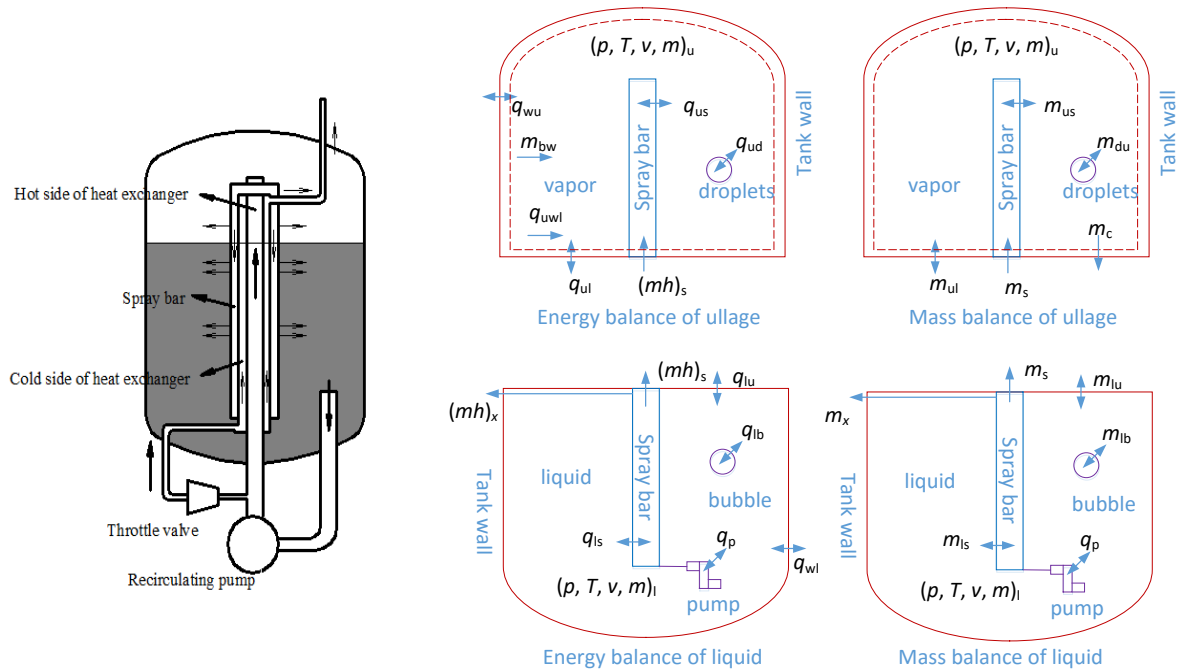


Fig. 1 Spray bar TVS schematic

The change rate of ullage temperature ( $T_u$ ) is calculated by

$$\frac{dT_u}{dt} = \frac{\Delta e_u - q_e - w_u - c_{vu} \cdot T_u \cdot \dot{m}_u}{m_u c_{vu}} \quad (8)$$

where  $\Delta e_u$  is the heat transfer to the ullage

$$\Delta e_u = q_{wu} - q_{ul} - q_{ud} - q_{us} - q_{uwl} - (\dot{m}h)_s \quad (9)$$

where  $q_{wu}$  is the heat transfer rate between the tank wall and ullage,  $q_{us}$  is the heat transfer rate between the ullage and spray bars,  $(\dot{m}h)_s$  is the heat flow of the spray fluids.

$q_e$  in Eq. (8) is the energy added to the ullage by mass transfer by boiling or condensation,

$$q_e = \left( \frac{dm_u}{dt} \right) \cdot h_g \quad (10)$$

where  $h_g$  is the enthalpy of saturated vapor (ullage).

$w_u$  in Eq.(8) is the work done on the ullage,

$$w_u = p_u \frac{dV_u}{dt} \quad (11)$$

where  $p_u$  is the ullage pressure,

$$p_u = \frac{m_u \cdot R \cdot T_u}{V_u \cdot M_u} \quad (12)$$

in which  $R$  is the ideal gas constant = 8.314J/(mol K),  $M_u$  is the average molar mass of the ullage, and the change of the ullage volume  $\frac{dV_u}{dt}$  could be calculated by

$$\frac{dV_u}{dt} = -\frac{dV_l}{dt} - \frac{dV_{wl}}{dt} \quad (13)$$

where  $V_l$  is the volume of the liquid,  $V_{wl}$  is the volume of the wall liquid.

The thermal modeling of the liquid is similar with the vapor model. These equations will be not brought out here because of the length limitations of the paper. For the recirculation pump, its transient rotation speed of the pump ( $N_t$ ) is given by

$$N_t = n_f (1 - \exp(-t/T_{na})) \quad (14)$$

where  $n_f$  is rated speed of the pump, and  $T_{na}$  is the time for 63% of acceleration from static to rated speed.

The head of the pump is

$$H_l = \frac{p_2 - p_1}{\rho g} \quad (15)$$

where  $p_2$  is the outlet pressure of the pump,  $p_1$  is the inlet pressure of the pump,  $\rho$  is the density of the fluid,  $g$  is gravity acceleration which is assigned to  $10^{-6} \text{ m/s}^2$ .

A fitting empirical correlation between head ( $H_r$ ) and flow rate ( $q_r$ ) for the pump is used according to a data from the manufacture

$$H_r = Aq_r^2 + Bq_r + C \quad (16)$$

where  $A, B, C$  are three constants coefficients.

The heat exchanger is simplified by giving an exact value of temperature drop of 2 K. The flow in and out of the throttle valve is treated as isenthalpic for a pressure difference 200 kPa.

### 3. Calculation procedure

The key of the TVS control is the start-up and shut-down events of the throttle valve and recirculation pump. The pressure control range is a specified target value which is higher than the initial pressure of the tank. When the ullage pressure is greater than the lower limit of the tank pressure control band, the recirculation pump starts up. And the ullage pressure will firstly decrease only by spray-mixing of vapor and liquid. However, once the vapor pressure of the saturated liquid reaches the higher limit of the pressure control band, mixing is not enough to control the pressure from not going too high. Then a small amount of fluid flows through the throttling valve where it is expanded to a thermodynamic state with lower pressure and temperature. This “low” temperature stream extracts heat from the recirculation flow in the heat exchanger, and after that turns into vapor and is vented to the environment. The program has been written in FORTRAN language based on the modeling equations in the previous section and the flow chart as shown in Fig. 2.

### 4. Results and discussion

For the sake of evaluating the effect by pressure control range only, the heat leak, initial state of fluid and fill rate etc. are all fixed. The tank size and working condition parameters are shown in Tab. 1. Three pressure bands, namely 165.5~172.4, 165.5~179.3, and 165.5~182.2 kPa are chosen for the evaluation. That is, the amplitudes of the pressure control ranges are 6.9, 13.8 and 20.4 kPa, respectively.

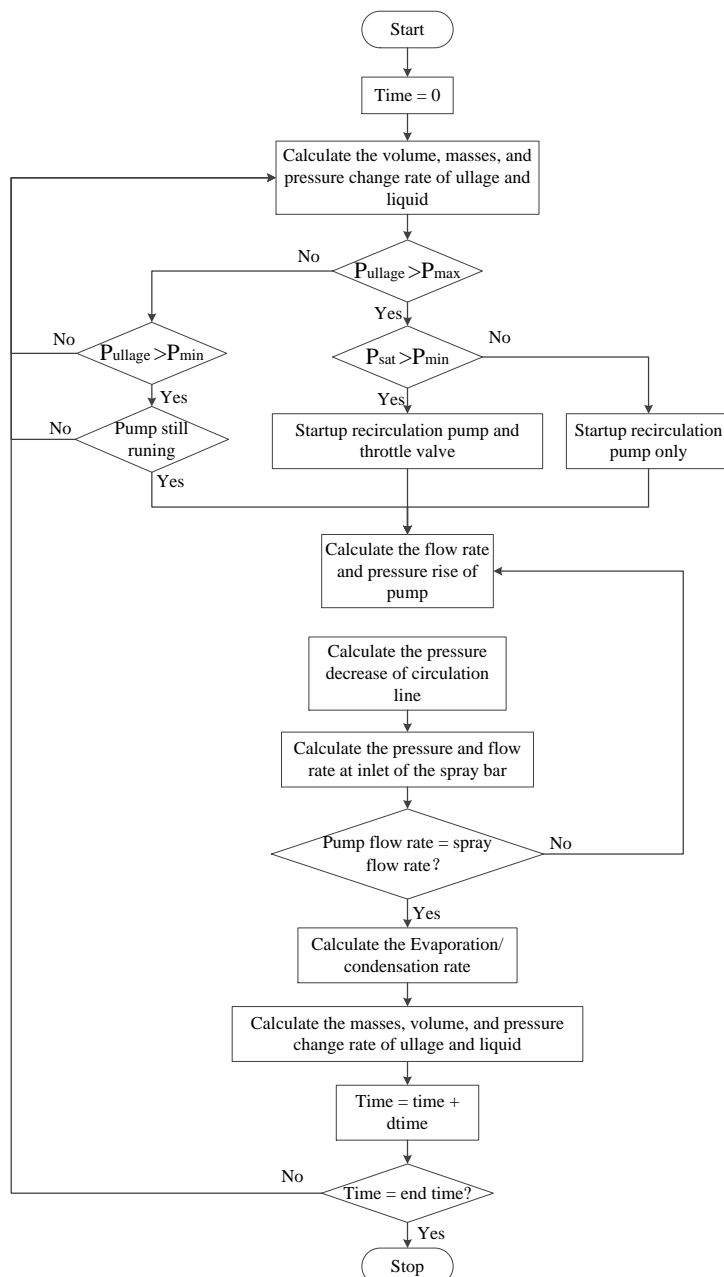


Fig. 2 Flow chart of the program for pressure control increases with the lifting of the higher limit of the control range. The period time of each cycle also increases approximately proportionally to the pressure band.

Table 1 Tank size and working conditions parameters

parameter	Tank volume (m <sup>3</sup> )	Heat leak (W/m <sup>2</sup> )	Fill level (%)	Flow rate of pump (L/min)	Initial temperature (K)	Initial pressure (kPa)	Pressure control range (kPa)
value	18.09	0.76	75	1070	21.5	137.9	165.5-172.4 165.5-179.3 165.5-182.2

For a 140-hours storage task of hydrogen as an example, the total cycle number (one cycle starts at the moment the TVS turning on and ends at the moment just before the TVS turning on again) and

Figure 3 shows the present calculation results compared with the experimental data of Hastings [8] for self-pressurization and venting with a 75% fill level and a 0.76 W/m<sup>2</sup> heat leak flux. First of all, the result indicates that the TVS model successfully simulates the self-pressurization process and confirmed the feasibility and effectiveness of TVS system in maintaining the tank pressure for long-time run. The predicted ullage pressure rises faster than the experimental values before the first start-up of the mixing mode, which takes about 32,000 s and 61,000 s, respectively. Our analytical model did not take into account the complex energy exchange that actually occurred at the liquid-vapor interface because for a zero-g case the total surface area of these L-V interfaces is hard to count. Besides, the model assumed thermal energy addition to the ullage at a constant rate, whereas it is possible that condensation mass transfer across the liquid-vapor interface began to suppress the ullage pressure rise rate. However, once the mixing and venting cycles begin, both the predicted pressure rise and reduction rates matched very well with the measured data. The pressure variation profiles corresponding to the three concerning pressure control ranges are shown together in Fig. 4. As expected, the time before the first mixing cycle

cycle interval time (after the current TVS operation stops and before the next start-up) are plotted in Figures 5 and 6 respectively for the three control ranges. The cycle number decreased from 62 to 21 while the cycle interval time increased from 1.9 h to 5.8 h when the control range increased from 6.9 kPa to 20.4 kPa. It means that the extension of the cycle time is almost proportional to the pressure gap and the main contribution comes from the delayed completion of the self-pressurization under the same heat leak due to the rising of the setting values of the higher pressure limit. In addition, the throttling valve will keep in open state for relatively longer time in every cycle because of higher starting pressure, which leads to a longer ullage cooling duration.

The running time is defined as the period time per cycle multiplied by the total cycle number during the storage task. The TVS running time and fluid mass venting loss under these pressure control ranges are also calculated and shown in Figures 7 and 8. It should be noted that these results are all with respect to the same  $18.09 \text{ m}^3$  tank with 75% filling rate ( $= 945 \text{ kg}$ ). It could be seen that the running time reduces from 3920 to 3200 s and the fluid mass loss decreased from 473 kg to 385 kg, when the pressure control ranges changed from 6.9 kPa to 20.4 kPa. Although the net period time per cycle gets longer, the running time still shortens significantly due to large reduction of the cycle number. The results indicate that the greater control range is, the less vapor mass loses, because less TVS running time means less heat leak generated by the pump operation. However, the mass of the tank wall increases because more material will be used to support the increment of the higher limit design pressure. A reasonable pressure control range for the TVS control strategy is required by trading-off the propellant saving and the cost of additional weight load of the tank.

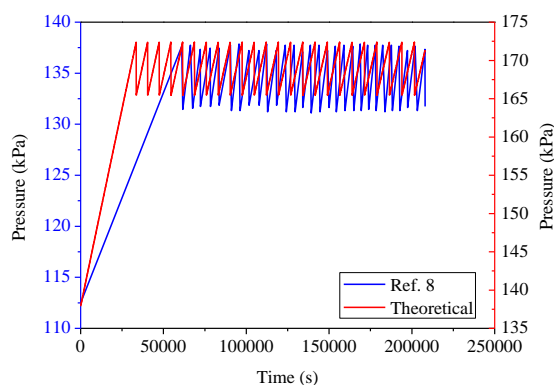


Fig. 3 Ullage pressure variation compared with Ref [8]

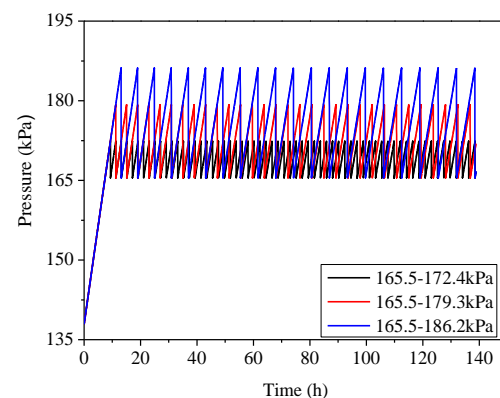


Fig. 4 Pressure variation profile under three pressure bands

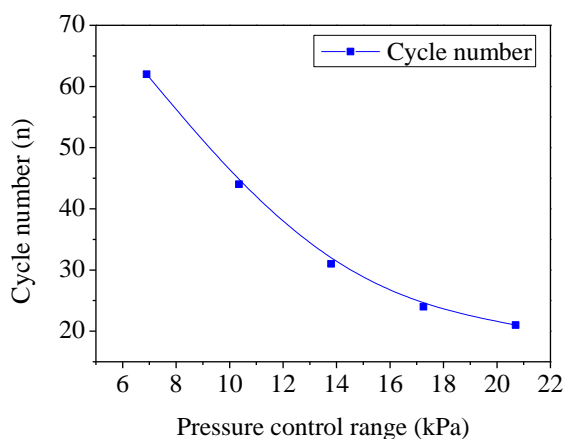


Fig. 5 TVS cycle number vs. pressure range

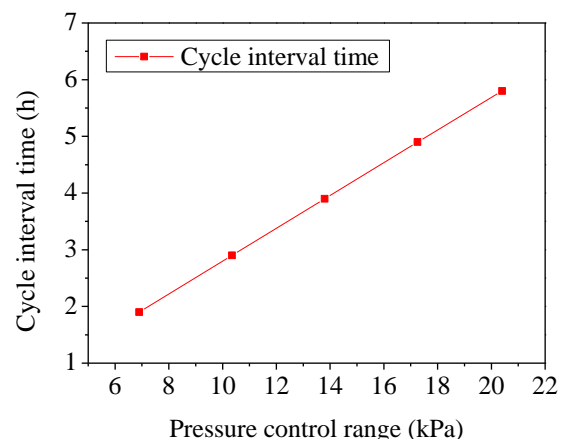


Fig. 6 TVS cycle interval time vs. pressure range

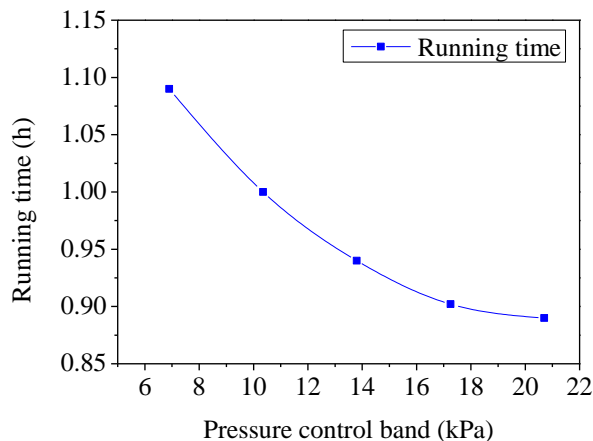


Fig. 7 TVS running time vs. pressure band

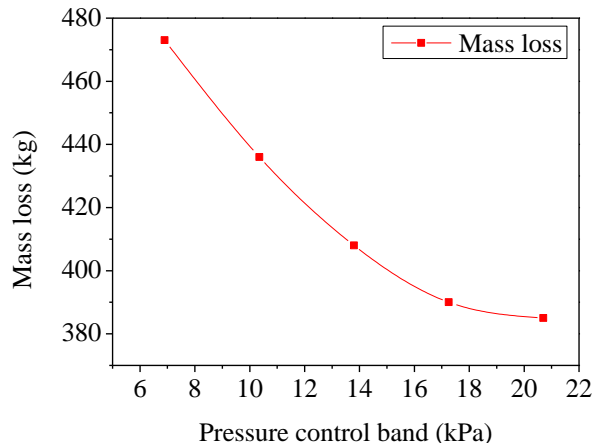


Fig. 8 Fluid mass loss vs. pressure band

## 5. Conclusion

A transient analytical model was created to predict the behavior of pressure variation inside a liquid hydrogen tank with TVS installed. The model successfully simulates the self-pressurization process and confirmed the feasibility and effectiveness of TVS system in maintaining the tank pressure for long-time run. The pressure control band has a great influence on the TVS operating characteristics and the fluid mass loss. For a 140-hours storage task of hydrogen as an example, when the pressure band increases from 6.9 kPa to 20.4 kPa, the cycle number decreases from 62 to 21, while the single running time increases from 63.3 to 152.2 seconds, and the cycle duration hour reduces from 1.09 to 0.89 hour. The rate of LH<sub>2</sub> evaporation loss in one day decreases from 0.17% to 0.14%. If higher weight load and pressure proof of the tank are allowed, larger control-range is preferred.

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