Extended Abstract for AIAA SciTech 2025 Conference

Nodal Modeling of Tank Pressurization and Draining using a Multi-Node-Ullage Approach

Andre LeClair and Alok Majumdar

NASA/Marshall Space Flight Center

1. Introduction

The purpose of the pressurization system in liquid rocket propulsion is to control the pressure in the gas space of the propellant tank (known as the ullage space) and the propellant mass flowrate to the engine. A mathematical model is required to predict the amount of pressurant necessary to ensure that pressure and temperature levels inside the tank remain within acceptable limits and that the propellant pressure leaving the tank satisfies the net positive suction pressure (NPSP) requirement of the pump feeding the engine.

Nodal codes typically model tank pressurization and draining using a single node to represent the ullage and a single node to represent the propellant. As the tank drains, the ullage node grows and the propellant node shrinks . The heat transfer between ullage to wall and ullage to propellant is governed by natural convection [1]. Designers of liquid propulsion systems often use empirical correlations [2] to estimate the "Collapse Factor" which represents the ratio of pressurant required with heat transfer and the amount of pressurant required without heat transfer. A single node ullage model [3] of tank pressurization was developed using GFSSP to compute the collapse factor reasonably well and later was used to model tank pressurization during test firing of the FASTRAC rocket engine. The predicted tank pressure compared well [4] with the test data.

In the early 1970's, pressurization and drain tests with liquid methane [5] were performed at NASA Lewis Research Center in a vacuum chamber. A 5 ft diameter spherical aluminum tank (Figure 1) was tested to drain from 95% to 5% full using gaseous helium, hydrogen, nitrogen and methane as pressurant. Tests were conducted with different pressurant inlet temperatures and drain times. Measured data include pressurant requirement, amount of pressurant condensed, and ullage and wall temperatures at various heights in the ullage space at the end of draining.

A single node GFSSP [6] model was developed [7] to simulate helium pressurization of the methane tank. Predicted helium consumption was 8-23% less than measured. The average error of the six test cases was 16%. A single ullage node with multiple solid node model was developed [8] using Thermal Desktop. Predicted helium consumption compares with the test data within 2%.   This paper describes the development of a GFSSP multi-node ullage model of the test configuration and compares the predicted pressurant consumption for both helium and autogenous pressurization using gaseous methane with experimental as well as TD predictions.

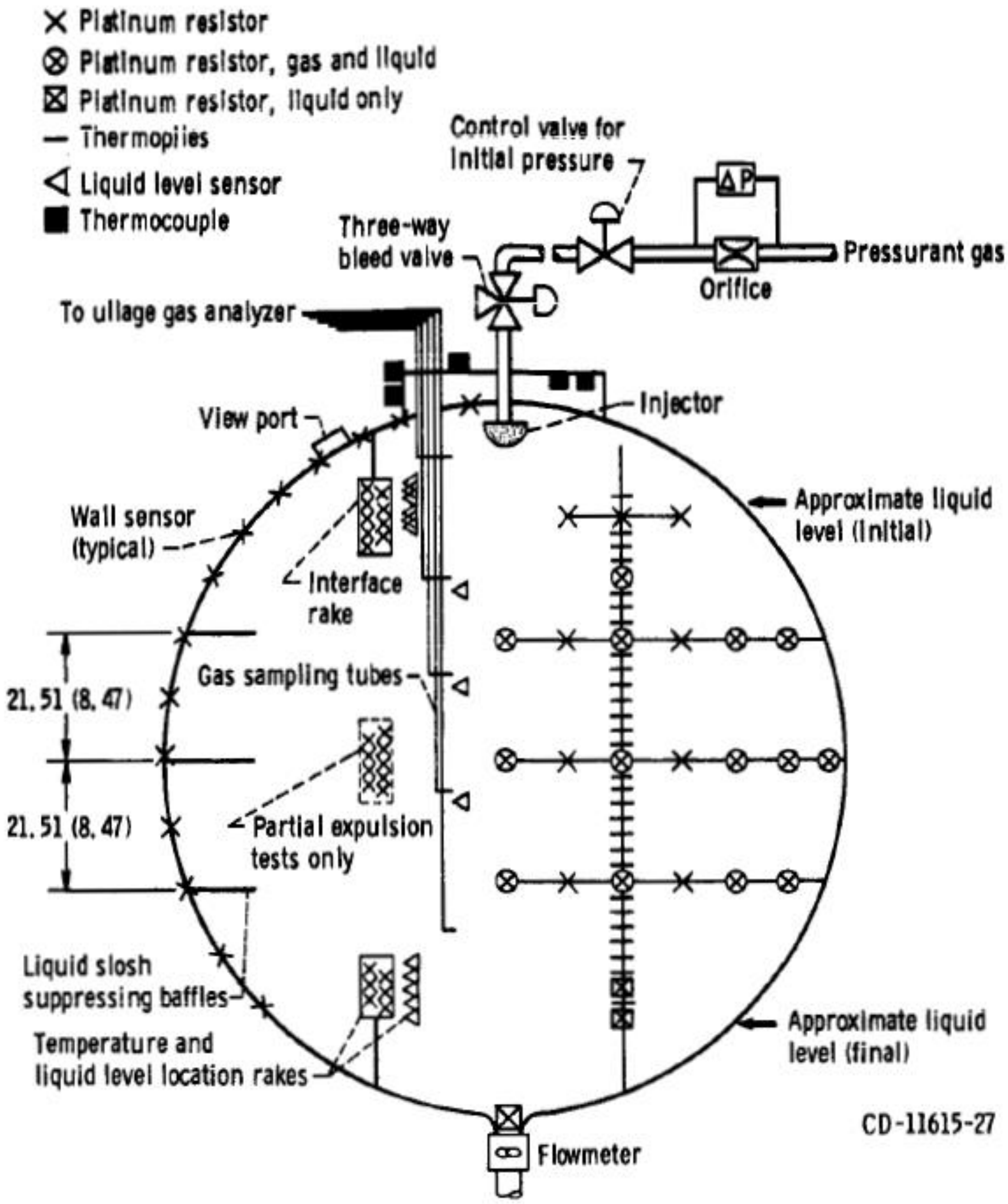

Figure 1. K-Site Test tank and its instrumentation

2. GFSSP Model

The ullage is discretized into six layers of equal height and represented by six nodes (Node 2, 3, 4, 5, 6 and7). The nodes are connected by 5 branches (Branch 23, 34, 45, 56 and 67) of nearly zero resistance. However, the momentum equation solved in these branches includes gravitational force.

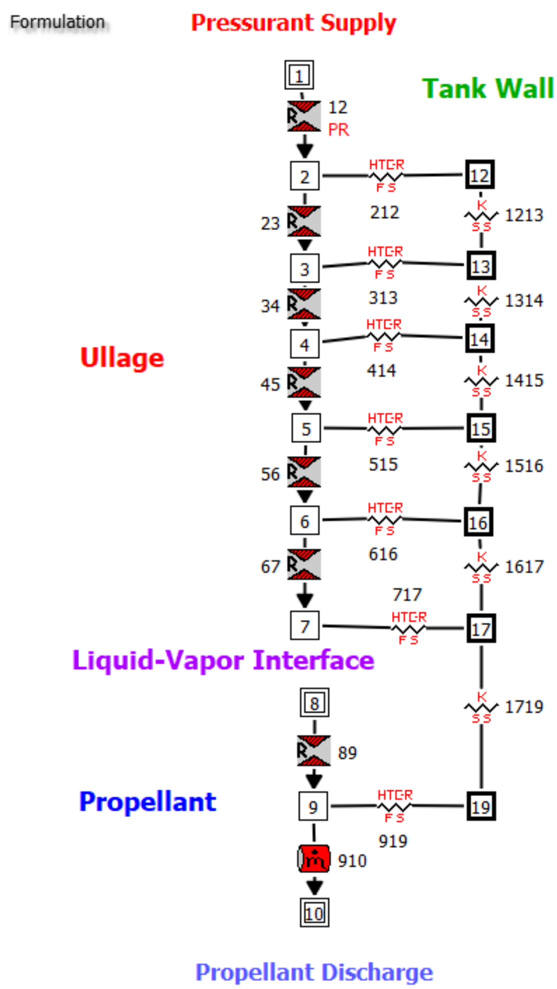

Figure 2. GFSSP Model of K-Site Test Tank [5]

The propellant is represented by a single fluid node (Node 9). The tank wall is discretized into 7 solid nodes (Node 12, 13, 14, 15, 16, 17 & 19) with the same height as the ullage nodes and propellant node. As the tank drains, the ullage volume grows and the propellant volume shrinks. In each time step, the ullage is re-discretized into layers of equal height. At the start of the run, when ullage is 5% of total volume, each layer is 0.11 feet high. At the end of the run, when ullage occupies 95% of the total volume, each layer is 0.64 feet high. In each time step, the tank wall solid nodes are re-discretized, and an energy balance is performed to calculate the wall temperature.

<u>Heat Transfer from ullage to wall and propellant</u>

This paper investigates the use of natural and mixed convection correlation in calculating the heat transfer between ullage to wall and ullage to propellant.

For natural convection, heat transfer between ullage and wall is given by:

$$Nu = 0.54Ra^{0.25} \quad (1)$$

and heat transfer between ullage and propellant is given by:

$$Nu = 0.27Ra^{0.25} \quad (2)$$

For mixed-convection the Woodfield correlation [9] is applied:

$$Nu_H = 0.56Re_D^{0.67} + 0.104Ra_H^{0.352} \quad (3)$$

Thermal Desktop offers this correlation as an option for simulating Tank Pressurization.

Condensation at the wall

It is necessary to model condensation for autogenous pressurization when gaseous methane is used to pressurize liquid methane. Condensation takes place at the wall when the wall temperature is below saturation temperature.

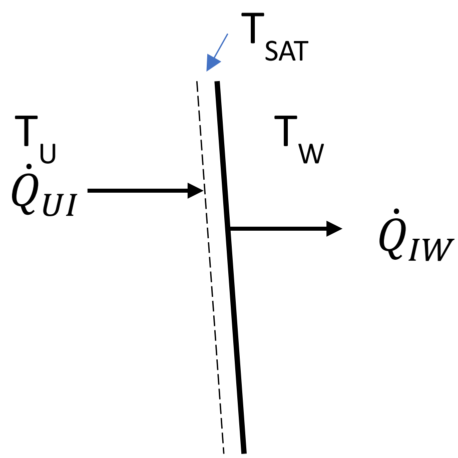

Figure 3. Condensation model at the wall

A schematic of the condensation model at the wall is shown in Figure 3. It is assumed that condensation occurs in a layer adjacent to the wall when the wall temperature is below the saturation temperature, and the layer is also assumed to be at saturation temperature.

The heat transfer from the ullage to the saturated layer is given by:

$$\dot{Q}_{UI} = [h_c A]_{U-I}\left(T_{ullage} - T_{SAT}\right) \quad (4)$$

Where the heat transfer coefficient, h_c is computed from Equation (1)

The heat transfer from the saturated layer to wall is given by:

$$\dot{Q}_{IW} = [h_c A]_{I-W}(T_{SAT} - T_w) \quad (5)$$

h_c of Equation 5 is evaluated from the following equation which is known as the Dhir-Lienhard [10] correlation of condensation heat transfer:

$$Nu_D = C\left[g\frac{\rho_f(\rho_f-\rho_g)h'_{fg}D^3}{\mu k \Delta T}\right]^{\frac{1}{4}} \tag{6}$$

<u>Condensation at ullage-propellant interface</u>

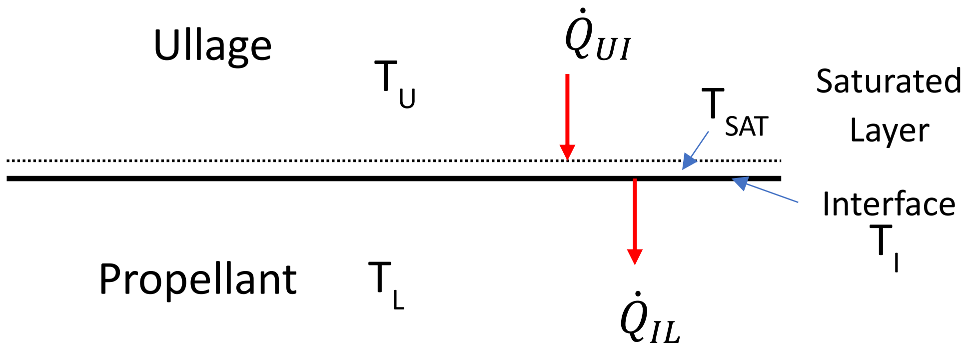

Figure 4. Condensation model at the ullage propellant interface

Condensation occurs at the ullage to propellant interface because $T_{SAT} > T_I$. The condensation rate is given by:

$$\dot{m}_{cond} = \frac{\dot{Q}_{UI} - \dot{Q}_{IL}}{h_{fg}} \tag{7}$$

$$\dot{Q}_{UI} = h_{c,UI}A(T_U - T_{SAT}) \tag{8}$$

$$\dot{Q}_{IL} = h_{c,IL}A(T_{SAT} - T_I) \tag{9}$$

$h_{c,UI}$ and $h_{c,IL}$ are computed from Equation 2 using vapor and liquid properties respectively.

The interface temperature, T_I is calculated from:

$$T_I = \frac{h_{c,UI}T_U + h_{c,IL}T_L}{h_{c,UI} + h_{c,IL}} \tag{10}$$

3. Results and Discussion

Helium Pressurization

DeWitt and McIntire [5] conducted six tests with helium as pressurant. They tested with three different drain times at two different inlet pressurant temperatures. For each run, simulations were performed using both natural convection correlation (Ring [1]) and mixed convection correlation (Woodfield [9]).

Run	He Inlet T (K)	Drain Time (s)	Measured He Mass (kg)	Ring Predicted He Mass (kg)	Ring Error (%)	Woodfield Predicted He Mass (kg)	Woodfield Error (%)
37	229	223	1.990	1.845	-7%	2.080	+5%
36	226	389	2.096	1.936	-8%	2.082	-1%
33	231	622	2.145	1.974	-8%	2.059	-4%
42	324	224	1.895	1.674	-12%	1.997	+5%
41	331	390	2.000	1.791	-11%	1.994	-0.3%
40	311	599	2.091	1.868	-11%	1.988	-5%

Table 1. The comparison of measured and predicted helium consumption using natural and mixed convection correlation.

The comparison of measured and predicted helium consumption using natural and mixed convection correlations is shown in Table 1. The model underpredicts helium consumption using the natural convection correlation. In general model predictions are more accurate with the mixed convection correlation. The observed discrepancies can be attributed to the absence of condensation and evaporation of methane which is present in the ullage. It may be noted that presence of methane during the start of pressurization has not been considered in this computation.

The predicted and measured temperature of the wall adjacent to ullage using natural and mixed convection correlation are shown in Figure 5. Contrary to helium consumption prediction, wall temperature prediction using natural convection correlation is more accurate than the predicted temperature with mixed convection correlation. The predicted and measured temperature of the ullage using natural and mixed convection correlation are shown in Figure 6. It may be noted that ullage temperature predicted by mixed convection correlation is more accurate than that predicted by the natural convection correlation in the slow drain cases.

Autogenous Pressurization

Six test cases of autogenous pressurization, where gaseous methane was used as pressurant, were reported. All six cases were simulated using the natural convection correlation. Table 2 provides comparison of the measured and predicted methane consumption and measured and predicted condensation for each test case. The model under-predicts methane consumption by an average of 10%. This is primarily because of underprediction of condensation by an average of 5%. In the proposed paper the mixed convection correlation of Woodfield and other condensation correlations will be attempted to improve the comparison. The proposed paper will also include the pressurization with gaseous hydrogen and nitrogen and will be compared with the measured data for all 22 cases reported by DeWitt and McIntire [5]. The effect of discretization of the ullage will also be investigated.

Run	Inlet Methane Temp (K)	Drain Time (Sec)	Measured Methane Mass (kg)	Predicted Methane Mass (kg)	Percent Discrepancy	Measured Condensate (kg)	Predicted Condensate (kg)	Percent Discrepancy
8	226	231	9.28	7.93	-15	2.85	2.16	-24
7	227	405	9.94	9.18	-8	3.29	3.14	-5
6	226	633	10.91	10.7	-2	3.63	4.40	21
11	338	234	7.87	6.52	-17	2.62	1.89	-28
10	344	410	8.71	7.66	-12	2.95	2.72	-7.9
9	339	638	9.64	9.14	-5	3.41	3.95	16

Table 2. The comparison of measured and predicted methane consumption and condensation using natural convection correlation.

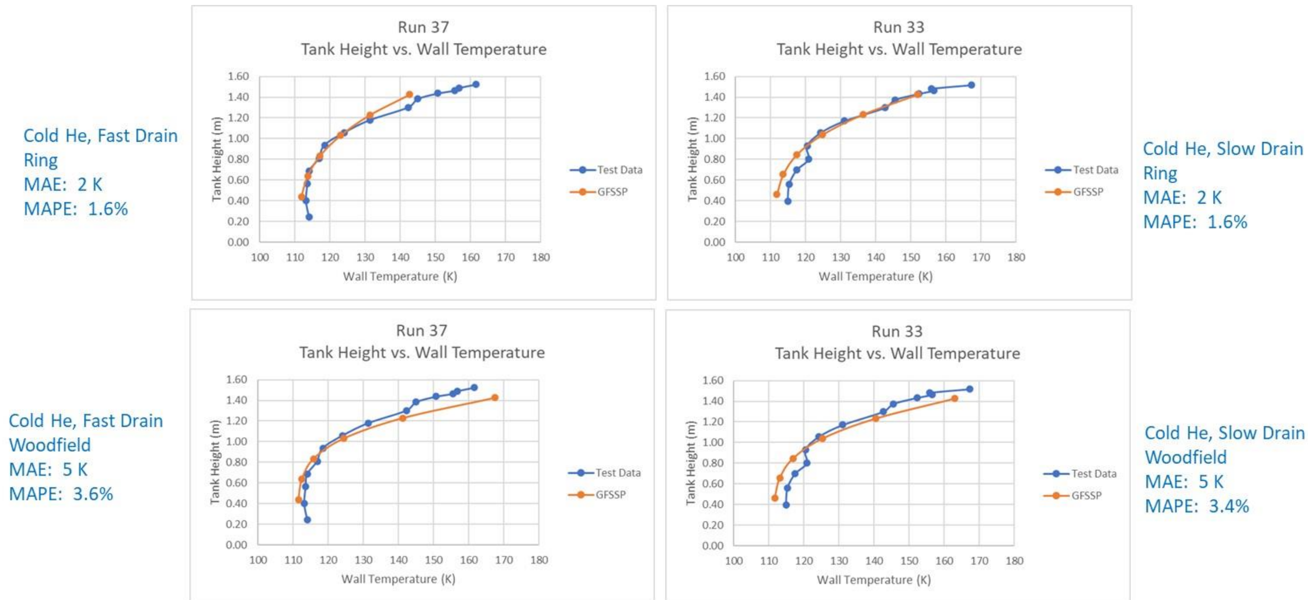

Figure 5. The predicted and measured temperature of the wall adjacent to ullage using natural and mixed convection correlation.

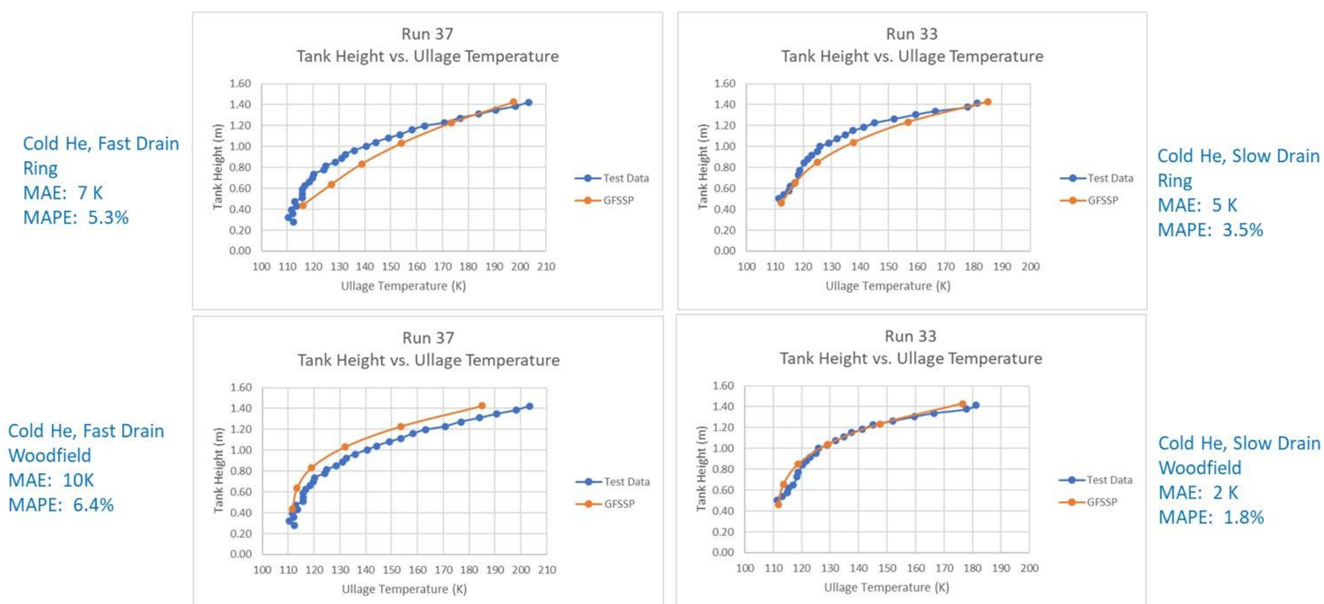

Figure 6. The predicted and measured temperature of the ullage using natural and mixed convection correlation.

4. References

1. Ring, E. Rocket Propellant and Pressurization Systems, Prentice Hall, Upper Saddle River, NJ, 1964, pp 173-245.
2. Epstein, M., and Anderson, R.E., "An Equation for the Prediction of Cryogenic Pressurant Requirements for Axisymmetric Propellant Tanks," *Advances in Cryogenic Engineering*, Vol. 13, Plenum, New York, 1968, pp. 207-214.
3. Majumdar, A. K. and Steadman, T., "Numerical Modeling of Pressurization of a Propellant Tank", Paper No. AIAA 99-0879, 37th AIAA Aerospace Sciences Meeting Conference and Exhibit, January 11-14, 1999, Reno, Nevada.
4. Steadman, T., Majumdar, A. K. and Holt K., "Numerical Modeling of Helium Pressurization System of Propulsion Test Article (PTA)", 10th Thermal Fluid Analysis Workshop to be held from September 13 – 17, 1999 at Huntsville, Alabama.
5. Dewitt, R. L. and McIntire, T.O., "Pressurant Requirements for Discharge of Liquid Methane from a 1.52 meter (5 ft.) diameter spherical tank under both static and slosh conditions." NASA TN D-7638, May 1974.
6. Generalized Fluid System Simulation Program, Version 6.0, User Manual, NASA/TP-2016-218218, March, 2016.
7. Flores, Noe, "Modeling Tank Pressurization with Generalized Fluid System Simulation Program", NASA Internship Final Report, Summer 2018.
8. Wendt, Bruce, "Thermal Desktop Pressurization Model Validation Report", Document No. CFM-Model -RPT-0016, June, 2022.
9. Woodfield, P. L., Monde, M., Mitsutake, Y., "Measurement of Averaged Heat Transfer Coefficients in High-Pressure Vessel during Charging with Hydrogen, Nitrogen or Argon Gas", Journal of Thermal Science and Technology, Vol. 2, No. 2, 2007.
10. Dhir, V. and Lienhard, J., "Laminar Film Condensation on Plane and Axisymmetric Bodies in Nonuniform Gravity", Journal of Heat Transfer, ASME, February, 1971.