

Validation of Two Phase Flow Modeling Techniques in Loci-Stream using Axial Jet Pressure Control and Autogenous Pressurization Experiments

Chintan S. Patel *

Qualis Corp (Jacobs Space Exploration Group), NASA Marshall Space Flight Center, Huntsville, AL

Two of the cryogenic propellant management (CFM) technologies used for controlling tank pressure are autogenous pressurization of the tank and depressurization using an axial jet. Reliable modeling of these phenomena is essential for supporting long term space flight goals of NASA. An accurate capture of the gas-liquid interface dynamics is the primary modeling bottleneck and includes integrating the dynamics of phase change and associated heat transfer as well as turbulence characteristics at and near the interface. Further, the flow regimes of low speed natural convection, and high speed jet are encountered in the same domain and make turbulence modeling challenging. Both of these major modeling challenges are addressed using computational fluid dynamics (CFD) analyses, which are indispensable for accurately simulating and understanding these problem with high fidelity. Improvements to our modeling techniques are presented here and includes utilizing IDDES (improved delayed detaches eddy simulation) to more effectively model turbulence than Reynolds-averaged Navier Stokes (RANS) modeling, adding interface turbulence damping to volume of fluid (VOF) method, modeling real gas behavior for near saturation gaseous hydrogen, and modeling temperature dependent tank solid properties to capture large variation in thermodynamic properties at cryogenic conditions. Validations with five different K-site axial jet pressure control experiments and an EDU (Engineering Design Unit) tank autogenous pressurization experiment are presented to demonstrate the effectiveness of the improved modeling techniques.

I. Introduction

NASA's near-future long-term space missions necessitate advancements in cryogenic fluid management (CFM), which includes safe, low cost, and reliable long-term propellant storage. Consequently, NASA STMD (Space Technology Mission Directorate) has established the CFM Portfolio Project to improve CFM technologies for upcoming missions.

In addition to advanced hardware development and demonstration, there is a need for accurate modeling of cryogenic system behavior to support rapid, reliable, and low-cost spacecraft design and operation. As part of the CFM modeling portfolio, the Propulsion Engineering Fluid Dynamics branch at NASA Marshall Space Flight Center (MSFC) has been tasked with assessing and improving computational tools used to support future and current flight projects such as Human Lander System and Commercial Lunar Payload Services. We are engaged in improving and validating multiphase models in the Loci-STREAM computational fluid dynamics (CFD) code to predict various CFM phenomena.

A brief introduction for both axial jet pressure control and autogenous pressurization can be found in our previous works [1][2].

In this paper, validation of NASA MSFC's Loci-STREAM CFD solver for predicting axial jet mixing and autogenous pressurization using experimental ground tests is presented to highlight the improved modeling techniques.

II. Computational Model

A. Governing Equations

The governing equations for mass, momentum, and energy conservation; phase change model; two phase sharp interface model are discussed in our previous work [2] and not presented here in interest of brevity.

*Computational Fluid Dynamicist, Fluid Dynamics Branch (ER42), NASA MSFC

For VOF simulations, interface turbulence damping (ITD) is implemented based on Fan et al [3] and will be included in the final manuscript.

III. Test Description

A. K-site Tank - Axial Jet Pressure Control

1. Testing Facilities

Several axial jet pressure control tests were conducted at the NASA Glenn Research Center's K-site testing facility [4]. These tests provide reliable time-transience of pressure and temperature data throughout the depressurization with the axial jet. An ellipsoidal tank with 2.2 m major diameter and 1.83 m minor diameter was insulated with multi-layer insulation (MLI) and stationed inside a vacuum chamber. By measuring the steady liquid hydrogen (LH2) boil-off rate, the heat leak of 4.2 W/m² was calculated.

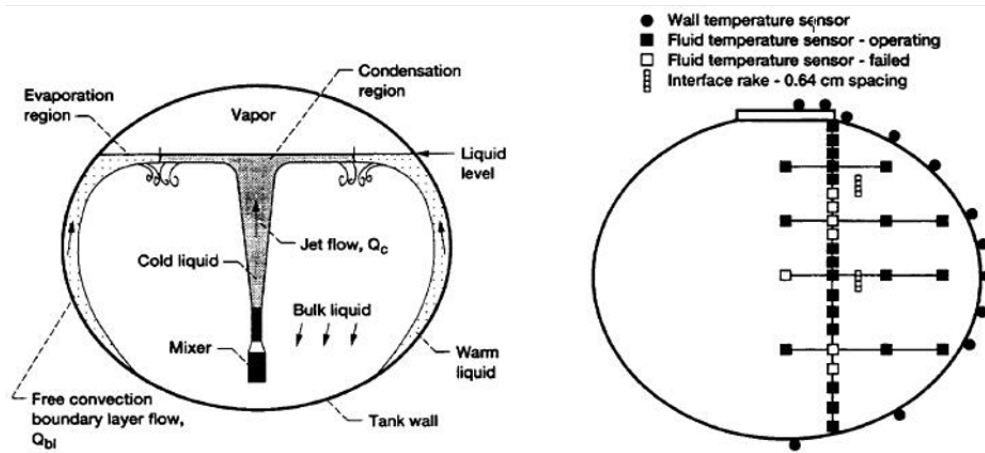


Fig. 1 Cross sections illustrating test configuration

The Shuttle Centaur LH2 tank mixer was installed along the minor axis of the tank, which also aligns with the direction of the gravity. The nozzle diameter was 22.1 mm and was located 0.51 m from the tank bottom. Figure 1 illustrates the configuration. A battery of sensors are places inside and on the tank for temperature, pressure and fill level measurements. Figure 2 demonstrates the locations of the temperature probes.

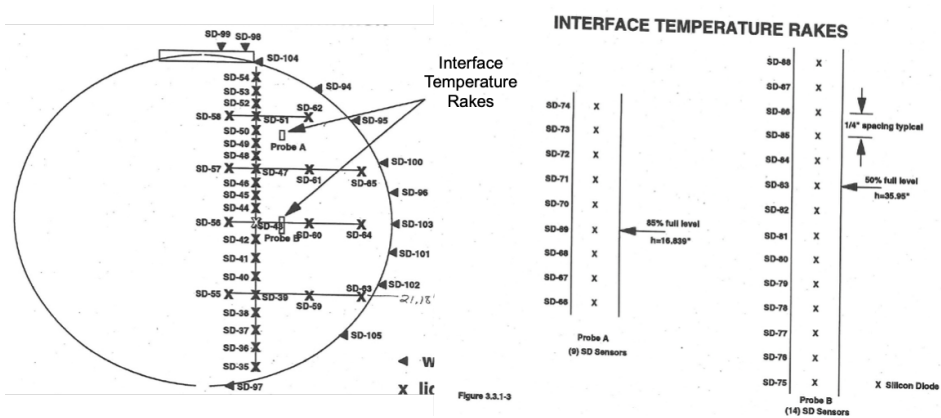


Fig. 2 Temperature sensor locations inside the K-site tank

2. Experimental Procedure

The tank was filled with liquid hydrogen, the hardware was brought to saturated liquid condition, and the tank was vented until a target pressure is achieved before the tank was locked up. Due to the heat leak to the tank, natural convection sets up a warm layer on liquid at the gas-liquid interface. Since, the process is slow, the interface is at the saturation condition, i.e., the ullage pressure is same as the saturation pressure at the interface. Initial temperature distribution from the temperature probes is shown in the Figure 5.

The test begins when the mixer at a set mass flow rate is turned on. The mixer circulates the cooler LH2 from the tank bottom to the interface continuously cooling the interface. The ullage pressure drops as the saturation temperature and pressure drops at the interface until a steady pressure is reached, and the test ends.

Five test cases are employed for the validation. Test conditions including fill level, mixer flow rate and initial tank pressures are shown in Table 1.

Case	Fill Volume [%]	Fill Height [m]	Mixer Flow Rate [m^3/hr]	Initial Tank Pressure [bar]
434	86.3	1.45 (79.1%)	3.47	1.861
436	85.3	1.43 (78.0%)	1.82	1.870
449	49.1	0.93 (50.7%)	3.41	1.861
457	49.1	0.93 (50.7%)	1.82	1.865
459	49.1	0.93 (50.7%)	0.92	1.847

Table 1 K-site axial jet pressure control tests used for validation

B. EDU Tank - Autogenous Pressurization

A brief background behind the EDU tank tests is provide in Patel et al [2].

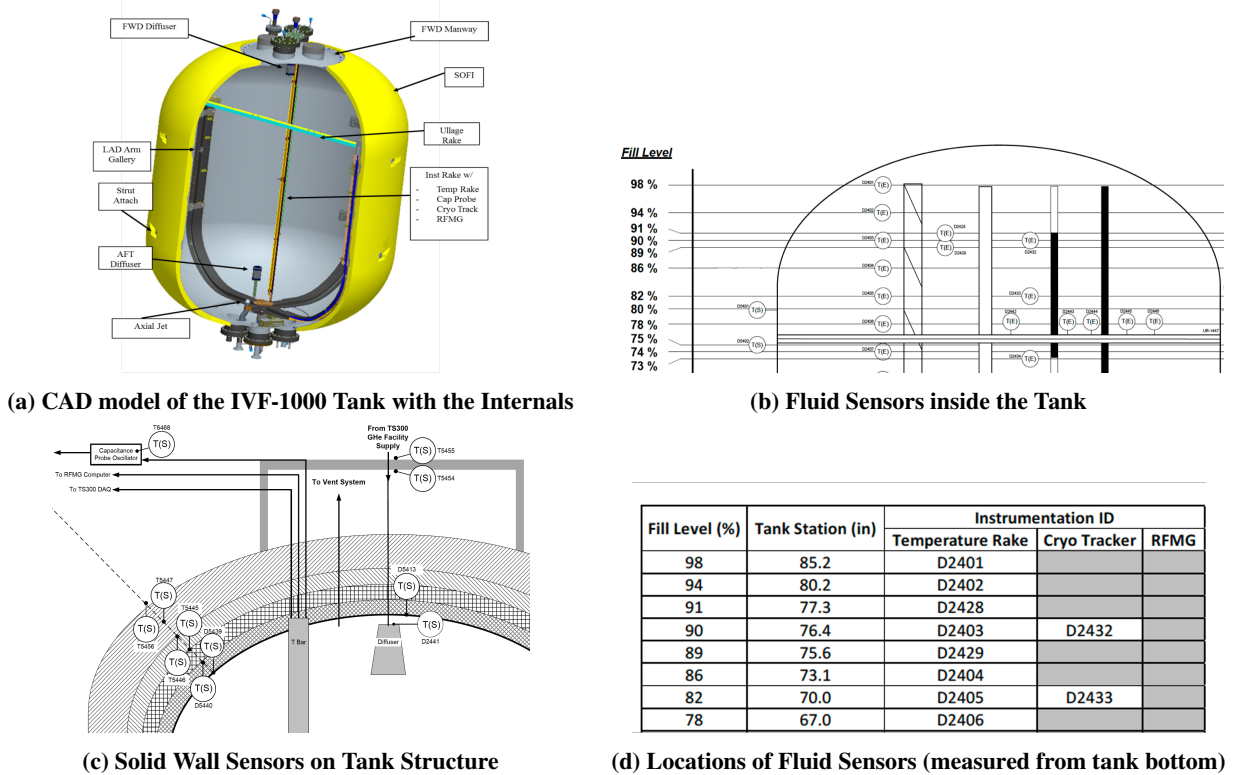


Fig. 3 Temperature and Pressure Sensors

1. Testing Facilities

The cylindrical double-domed tank is constructed with 2 mm thick 2219 aluminum and has an internal volume of 4.367 m³. The tank cylinder has a radius of 0.852 m and height of 2.334 m, and the half oblate spheroids have a minor radius of 0.615 m. The tank is insulated with Spray-On Foam Insulation (SOFI) as well as Multi-layer Insulation (MLI) for insulation in vacuum conditions. The tank was mounted on a stand with struts and housed in vacuum chamber at NASA MSFC test stand 300. A series of pressure and temperature sensors (see Figure 3) were located on horizontal and vertical rakes inside the tank, on the diffuser inlet, the tank wall, and the MLI layers. [2]

2. Experimental Procedure

For this autogenous pressurization modeling test, the tank was filled with liquid hydrogen (LH2), vented at atmospheric pressure to achieve a saturated gas-liquid interface and then locked-up at 90% fill volume. Temperature probe data from ullage, diffuser and suggest uniform initial temperature was achieved. All drain valves remained closed throughout this test. Room temperature gaseous hydrogen (GH2) pressurant at 297 K was introduced into the ullage through the diffuser for approximately 100 seconds. The measured diffuser mass flow history is shown in Figure 4 along with the corresponding ullage pressure. [?]

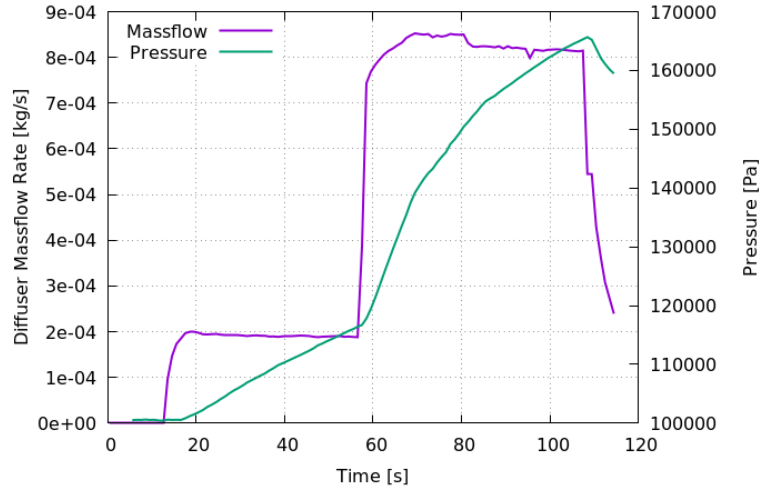


Fig. 4 Diffuser Massflow and Corresponding Tank Pressure

IV. Simulation Set-up

A. K-site Tank - Axial Jet Pressure Control

1. Geometry

The tank was modeled as an axisymmetric tank because there are no significant circumferential velocity components expected. A 1-degree tank sector was modeled. All the circumferential velocities are zeroed out in the governing equations. Tank solids were not modeled because they are not expected to impact the dynamics of the problem. Indeed, when the problem was simulated without the heat leakage to the tank walls, no significant differences were observed in the depressurization process – indicating that small amount of heat transfer between the tank fluid and the walls does not impact the problem.

Jet inlet and exit locations to the recirculating pump/mixer are known along with the jet nozzle diameter but the precise geometry of the mixer is unknown and is approximated. The mixer itself is not modeled. These approximations are expected to be inconsequential because no important thermal or flow dynamics is taking place in this region.

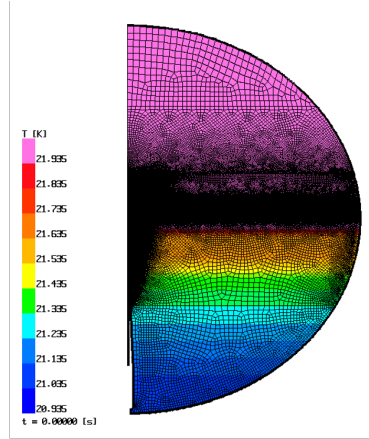


Fig. 5 Modeled CAD and refined mesh for K-site axial jet pressure control test

2. Mesh

Several mesh refinements are tested to obtain mesh insensitive predictions. A refined mesh (with temperature distribution) is shown here. Note that a heavy mesh refinement is employed in the regions of high shear and near the interface to accurately resolve the flow-field turbulence and the thermal gradients at the interface.

3. Initial and Boundary Conditions

The problem was simulated from the onset of jet turning on. Temperature point probes data from the test at the jet onset was used to initialize the temperature. This approach provides the most accurate initial condition possible. Alternate initialization was generated by simulating self-pressurization prior to the activation of the jet, but it showed no significant changes from the aforementioned initialization. Initial interface is saturated, i.e., the saturation temperature corresponding to the ullage pressure is supplied for the interface.

Tank walls were supplied with constant heat flux of 4.2 W/m² to simulate the heat leak. The mixer internals are irrelevant for the problem and not modeled. The jet mass flow was imposed using constant velocity and fixed temperature boundary condition at the mixer outlet (i.e., inlet in the computational domain), and corresponding volumetric flow rate or mass flow rate at the mixer inlet (i.e., outlet to the computational domain).

As the test progresses the tank bottom LH2 temperature warms but slightly and the inlet to the tank is warmer. So, alternatively, an inlet with time-varying temperature was considered to more accurately model the temperature of the jet entering the computational domain, but the small variation in the inlet temperature showed solution showed no impact on the depressurization rate.

B. EDU Tank - Autogenous Pressurization

1. Geometry

For modeling, salient features and retained and unimportant details are not modeled. Tank internals are removed with the exception of the forward diffuser, which is simplified. Axisymmetric flow is assumed. The diffuser is simplified to annular cylindrical slot.

The tank walls are also modeled using conjugate heat transfer analysis because heat transfer between the tank fluid and walls have significant impact on the ullage pressure. The measurements across the 40 MLI temperature probes show that the tank is well-insulated in the vacuum chamber and tank external wall is considered adiabatic.

2. Mesh

The computational mesh must be able to capture the important features of the high-speed diffuser flow, low velocity ullage flow with convection and heat transfer between ullage gas at cryogenic temperatures and pressurant gas at room temperature, and the sharp thermal gradient at the gas-liquid interface. Unstructured hexahedral meshes are used for this validation for their computational efficiency. Several meshes with varying resolution in the diffuser, ullage and interface

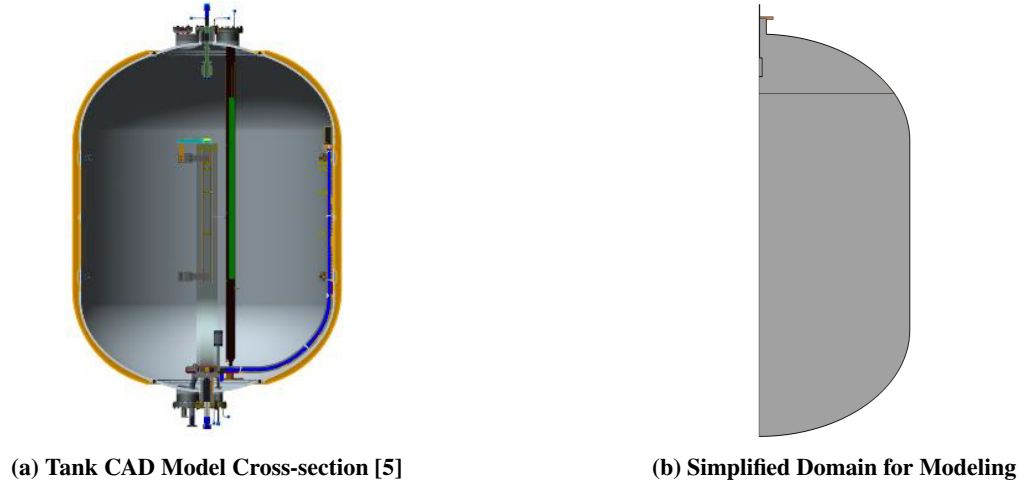


Fig. 6 Tank Model with and without Tank Internals

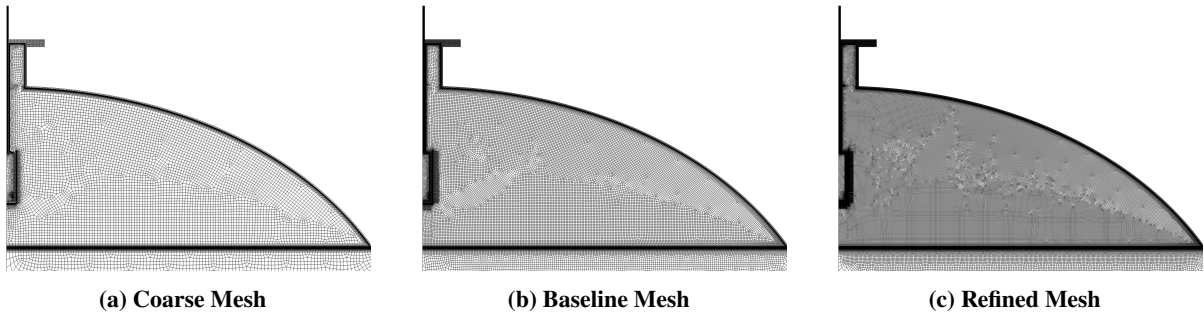


Fig. 7 Mesh Resolutions Tested for Grid-independent Solution

regions were constructed and tested. A representative sample is presented in the Figure 7. To accurately calculate the heat transfer between the wall and the fluid, full integration to the wall approach was used. The boundary first cell height was varied between 1×10^{-4} m and 5×10^{-5} m. The nominal cell edge length in the ullage varies between 2.5 mm and 7 mm. The interface first cell height was between 1×10^{-4} m and 1×10^{-6} m.

The coarsest mesh (Figure 7a) has 25,480 cells on the axisymmetric plane, which is an equivalent cell count of 9.2 million cells for a full 3-D model. Successive refinement yields equivalent cell counts of 18.8 million (Figure 7b), and 40.6 million (Figure 7c). The baseline mesh was further refined at the interface to achieve a refined-interface mesh with 22.3 million cells. Solution sensitivity to the mesh is presented in the *Mesh Resolution Sensitivity* section.

3. Initial and Boundary Conditions

Temperature probe data show nearly uniform temperature across the liquid and the ullage at the onset of the experiment. Therefore, a uniform initial temperature initial condition was applied. This temperature corresponds to the saturation condition for hydrogen at the atmosphere pressure. Time-varying massflow and temperatures were measured during the test and were prescribed at the diffuser inlet. A time-varying mass-flow boundary condition was implemented at the diffuser inlet by supplying an input file that specifies GH2 mass-flow rate and temperature.

An adiabatic wall condition was prescribed at the external tank wall surface as discussed in the *Set-up* section. As the inlet is expected to have "medium" turbulence, turbulence intensity of 5% with eddy viscosity ratio of 1000 were prescribed. Several simulation tests were carried out to ascertain impact of initial turbulence and diffuser inlet turbulence. The pressurization predictions were insensitive to a wide range of the turbulence conditions. A symmetry boundary condition was applied on the sector planes.

C. Fluid Properties

Fluid and material properties were evaluated from the current National Institute of Standards and Technology (NIST) databases [6]. The liquid hydrogen is assumed to be a Boussinesq fluid to model buoyancy, while the ullage is modeled as a compressible ideal gas. Since gaseous hydrogen deviates about 12% from the ideal gas behavior [5], real gas is modeled using tabular data from NIST which in addition to the equation of state also provided thermal properties and transport properties.

The solids are modeled with aluminum 2219 thermal properties [7]. The capability to model temperature dependent variation in the solid properties is implemented and improved the time-accurate pressure prediction in the tank ullage.

The liquid (Table 2), gas (Table 3), and tank (Table 4) properties are listed in the tables below.

Parameters	Description	Value
ρ_l	Liquid density	70.78 kg/m ³
c_{pl}	Liquid specific heat at constant pressure	9758.00 J/Kg/K
k_l	Liquid thermal conductivity	0.1008 W/m/K
μ_l	Liquid dynamic viscosity	1.3446×10^{-5} kg/m/s
α_l	Liquid volume expansion coefficient	0.016 58 K
L	Latent heat	436 480.0 J/kg
σ	Surface tension	0.001 776 N/m

Table 2 Liquid hydrogen thermodynamic properties at 20.3450 K and 133000.0 Pa

Parameters	Description	Value
ρ	Density	Real gas
c_{pg}	Specific heat at constant pressure	Function of temperature
k_g	Thermal conductivity	Function of temperature
μ_g	Liquid dynamic viscosity	Function of temperature

Table 3 Gaseous Hydrogen Thermodynamic Properties

Parameters	Description	Value
ρ	Density	2840.0 kg/m ³
c_p	Specific heat at constant pressure	Function of temperature
k	Thermal conductivity	Function of temperature

Table 4 Aluminum 2219 Thermodynamic Properties

D. Simulation Details

Time integration is computed implicitly with either first- or second-order backward Euler method. A first or second order upwind scheme is used for spatial discretization. The second order schemes are employed whenever possible. Solution insensitive to mesh, timestep size and accommodation factor will be presented in the final manuscript.

The Antoine equation [2] is used for the calculation of the saturation condition. The kinetics - balance iterative method was used for phase change, which provides more modeling flexibility than the balance method because it does not assume equilibrium condition at the gas-liquid interface. The accommodation factor coefficient in the phase change model is set to 1.0. Sensitivity to the accommodation factor used in the phase change model (Equation ??) is also presented in the forthcoming sections.

When conjugate heat transfer analysis is used, a tight fit between mating solids was assumed and a small thermal contact resistance of $1e - 16$ was supplied for their contact surface.

V. Results (Tentative)

A. K-site Tank - Axial Jet Pressure Control

Tank pressure, and probe temperatures from the test data are compared with various CFD simulations for validation showing good agreement with the test data. While the two phase sharp interface method approach is the primary focus of the study, the improvements made to the VOF method with implementation of the ITD module will also be briefly discussed.

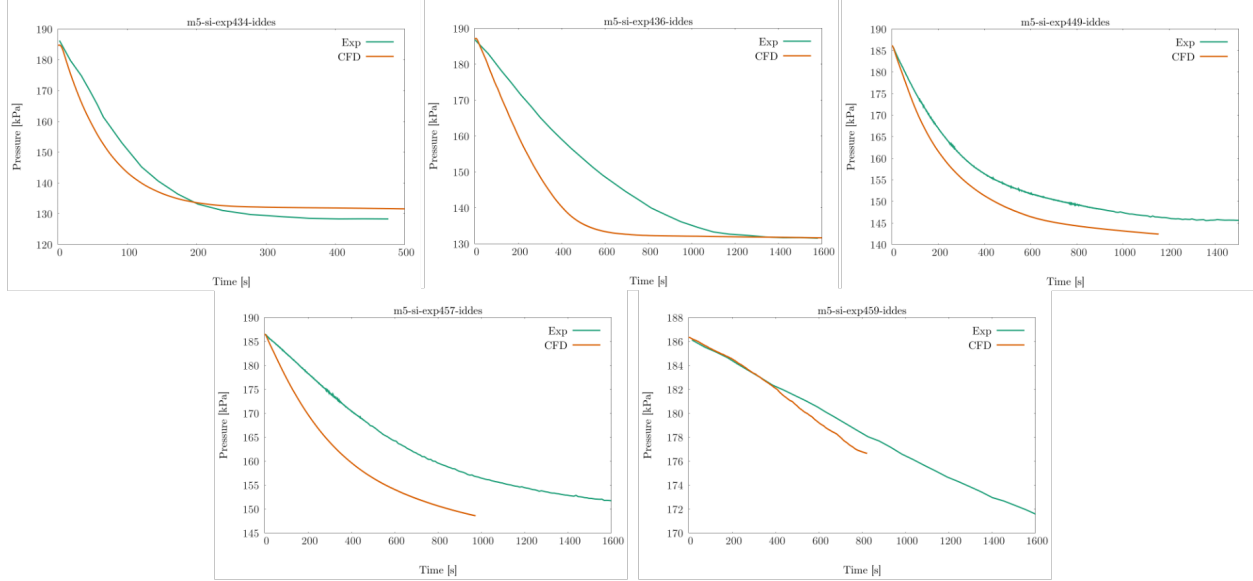


Fig. 8 CFD pressure predictions compared to the test data for Test 434, 436, 449, 457 and 459

The pressure transience is well predicted with slightly early prediction of pressure drop suggesting mildly excessive turbulent mixing at the interface. Temperature probe comparisons are also shown in Figure 10 and 11 and show satisfactorily capture the experimental trends. Note, that some experimental temperature sensor data is missing, but CFD data for those sensors is shown nevertheless in the plots. While the VOF with ITD is able to accurately predict the pressure, the VOF method is limited by exorbitantly small timestep required due to the refined interface because geometric VOF requires explicit time-integrating with strict Courant number limiter for stability. Moreover, geometric VOF also suffers from spurious velocities at the gas-liquid interface. While more work needs to be done to improve the stability, the VOF method shows drastic improvement in pressure predictions when ITD is enforced.

Final manuscript will include comparison to the RANS model temperature predictions to demonstrate superior prediction capability for this class of problem. RANS vastly over-predicts the viscosity ratio in low-speed velocity

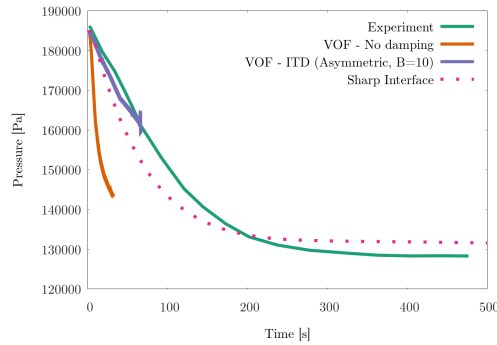


Fig. 9 Pressure predicted with VOF-ITD method for Test 434

regions in the tank and this will be demonstrated by IDDES's ability to better predict the temperature in the tank bottom showing superior turbulence modeling.

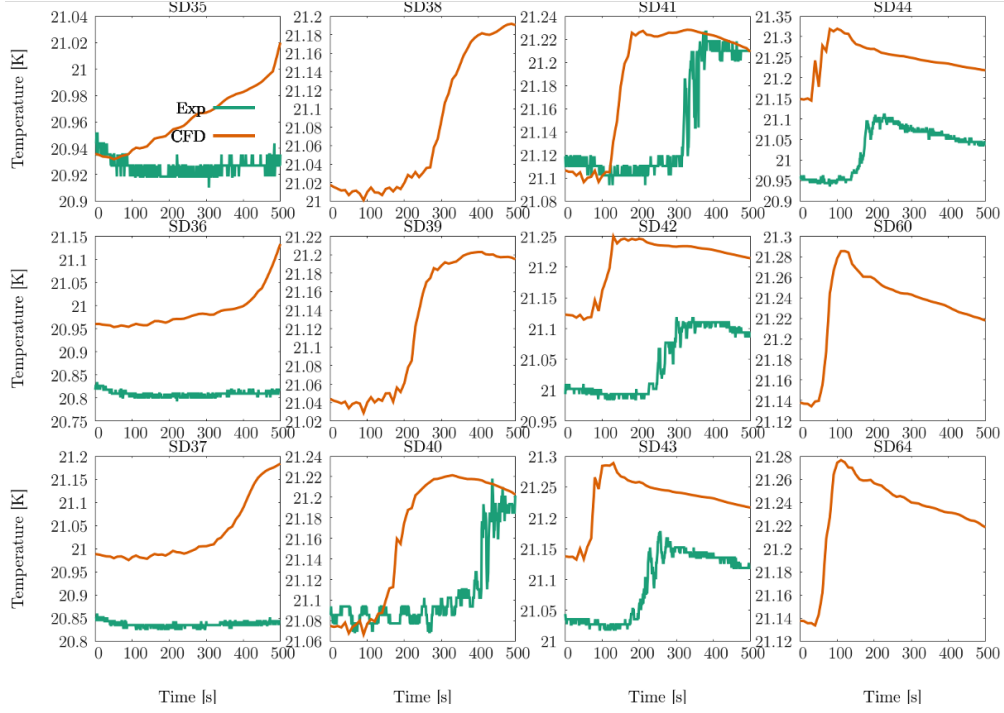


Fig. 10 Temperature Comparison with Axial Temperature Rake Probes

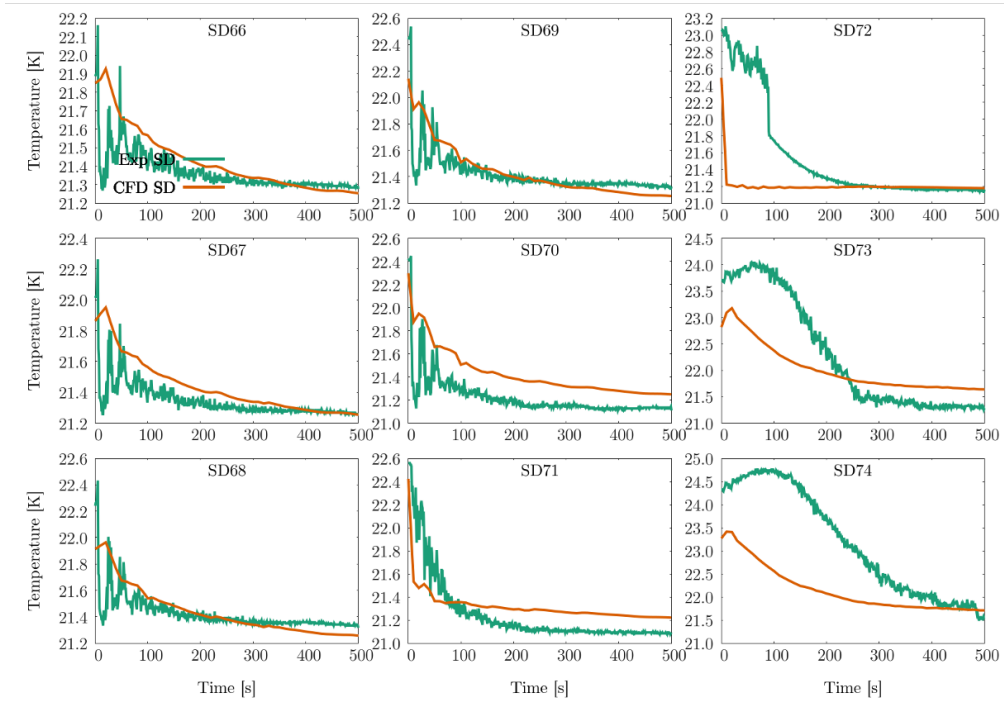


Fig. 11 Temperature Comparison with Interfacial Temperature Rake Probes

The first cell height of 0.1 mm was required to accurately capture the thermal gradient arising from the latent heat

exchange at the gas-liquid interface. More details will be added in the final manuscript.

Turbulence modeling for two phase interface is discussed in Patel et al [2]. Here we focus on the modeling on the turbulence in the rest of the tank and specifically for liquid where the turbulent jet interacts with rest of the flow in the tank liquid where away from the tank walls low turbulence is expected.

Overall unsteady URANS perform satisfactorily in predicting pressure transience of all experiments, but underpredicts final pressure due to excessive turbulent mixing. Tank temperatures are also no well predicted by the URANS modeling. Overprediction of turbulence viscosity in the low Reynolds number region of the flow-field also results in excess turbulent mixing in the tank, as indicated by higher temperatures.

Improved Delayed Detached Eddy Simulation (IDDES) modeling performs better than the URANS. Pressure transience are predicted more accurately than URANS. URANS's underprediction of final pressure is largely ameliorated because IDDES does not overpredict turbulence viscosity in the low Reynolds number region. Overt turbulent mixing is avoided and the temperature prediction across the tank are improved.

In summary, the wall bounded flow regions, the jet free shear regions, and the interface regions seem to be turbulent, while the rest of the flow region appear to have very low turbulence. The totality of the flow regime is best modeled using IDDES. The DHRL (dynamic hybrid RANS LES) approach [8] will also be tested before the final manuscript is released.

B. EDU Tank - Autogenous Pressurization

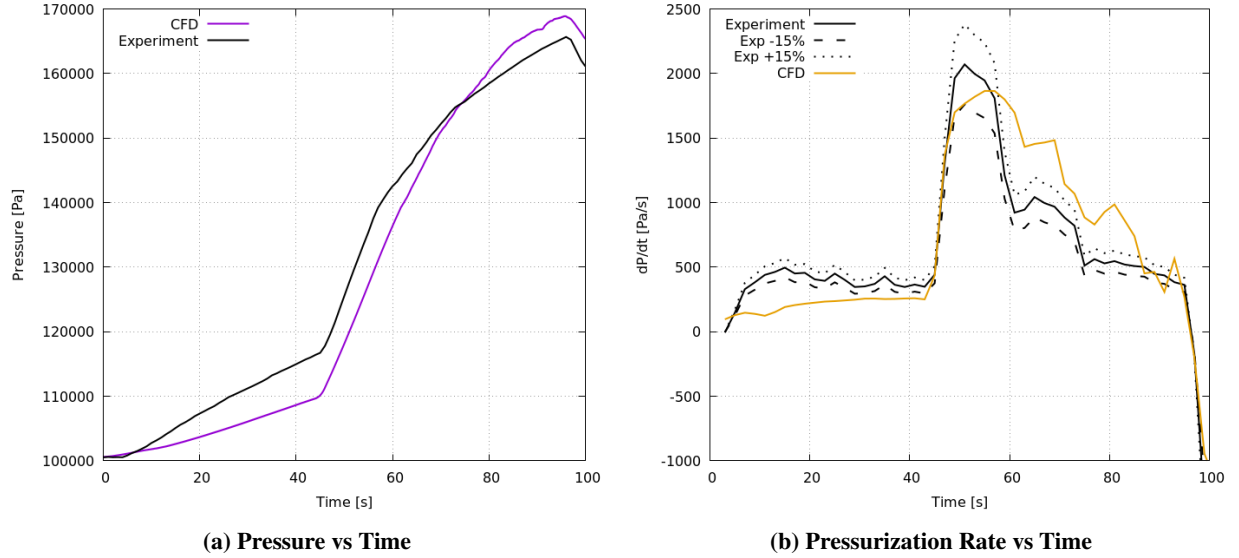


Fig. 12 Pressure and pressurization rate compared with the experiment

Pressure is predicted with high accuracy (Figure 8). This is a significant reduction in error compared to the previous predictions [2], primarily owing to implementing temperature dependent thermal properties for the tank solids as well as improving the modeling of phase change energy source term. The observation indicates the important role tank modeling plays in accurate prediction for autogenous pressurization.

VI. Conclusion (to be expanded in the final manuscript)

The IDDES approach predicts the turbulence characteristics of the high velocity jet in near-quietest tank better than RANS model. Mesh requirement for accurate interface capturing are identified. Interface turbulence damping is implemented with the geometric VOF and significantly improves the time-accurate pressure prediction. Temperature dependent solid thermal properties significantly improves the pressure prediction of the autogenous pressurization. The validated modeling approach presents a decisive step forward in reliable high-fidelity modeling techniques for supporting design of long-term space flight missions.

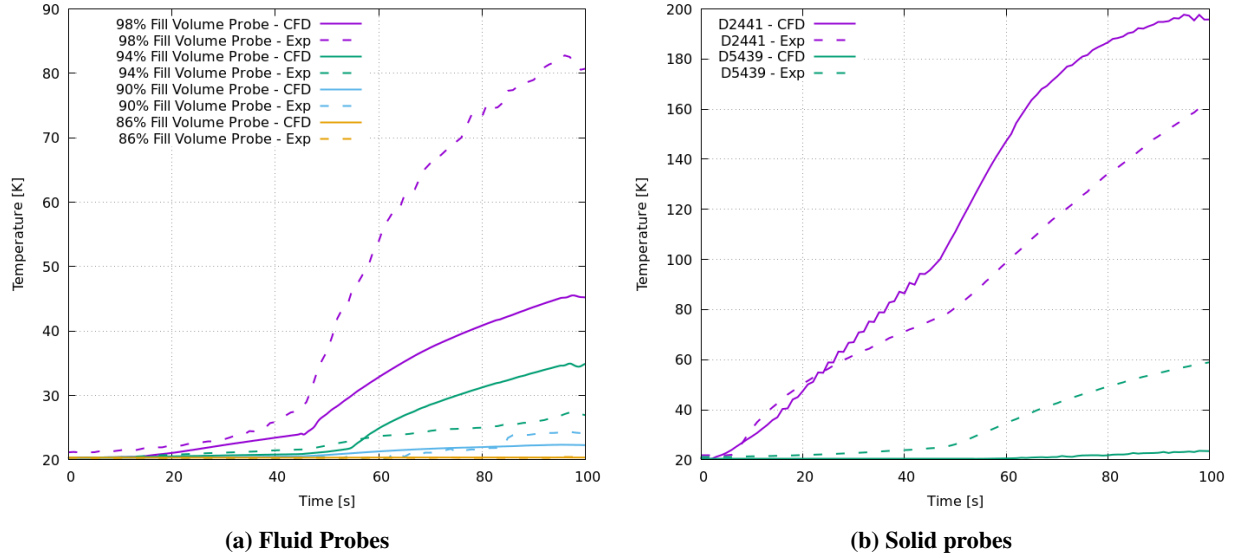


Fig. 13 Probe temperatures comparison with the experiment

Acknowledgments

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