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Development of a two-phase, variable nozzle jet pump for pressure control in liquid hydrogen systems

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Abstract. Liquid hydrogen systems require pressure control methods when flowing through complicated plumbing networks. Methods vary from autogenous pressurization, introducing a secondary fluid, buffer volumes, or utilization of transfer pumps. To address this need, a jet pump was designed, built, and tested with liquid hydrogen to assess variable primary nozzle positions in relation to secondary fluid mixing and inlet/outlet pressure differentials. This paper reviews the technological development, testing, and efficiency results when using a venturi jet pump for liquid hydrogen applications. This new liquid hydrogen pumping paradigm has the potential to reduce the level of cavitation concerns, frequency of maintenance required by reducing complex sealing, while maintaining a positive suction pressure and head and decreasing outlet quality for pressurized liquid hydrogen feed systems.

1. Introduction

Integrating liquid hydrogen into industries such as aviation, automotive, or material handling, there are multiple paths to explore as to what the best downstream consumer can be including fuel cells or gas turbines. These all have different flow requirements, pressure ratings, and ideal fluid conditions such as liquid or gas flow. Gas turbines generally require higher pressure feeds in the range of 50-100 bar while fuel cells require lower pressures to operate from 5 to 10 bar. The volumetric expansivity of liquid hydrogen is about 860 times at standard conditions during phase change from liquid to gas [1]. This expansivity allows for effective, autogenously pressurized systems that can provide fuel to a consumer in a vapor state. However, if there are components that require a single-phase liquid feed, designing with minimal heat ingress while maintaining constant pressure in a liquid state can be more challenging, particularly when transferring the saturated liquid. Autogenously pressurized systems also pose limits to pressure control if utilizing heaters to induce phase change. Previously, liquid hydrogen pumps have been developed in either a centrifugal or reciprocating configuration [2]. Centrifugal pumps accelerate the fluid radially outward as the impeller rotates. The impeller transfers kinetic energy from the blades to the fluid, creating a high-pressure region at the end of the blades, which drives the fluid to the outlet of the system. Reciprocating pumps utilize a crankshaft and piston cylinder to generate a high-pressure region using the positive displacement of the piston. Reciprocating pumps are typically used in higher-pressure applications where pulsating flow is acceptable, while centrifugal pumps are employed in lower-pressure systems where constant flow is preferred, especially in systems where dynamic sloshing is present and autogenous pressurization might be insufficient.

The challenges associated with utilizing piston and reciprocating pumps with liquid hydrogen include operability down to 21 K, the need for complex sealing surfaces, and the mechanical and high electrical power requirements that increase the risk of cavitation when operating in saturated liquid conditions. Impellers also produce a low pressure region upstream of the impeller which drops the saturated pressure into a two phase regime, increasing the risk of cavitation. Active cooling mechanisms could be implemented using cryocoolers. However, this increases the power and cost to maintain adequate subcooling for the pumps. Venturi jet pumps can function as primary pumps able to produce a net positive suction head and pressure (NPSH and NPSP) or act as a pre-cooling boost pump for a primary pump. Jet pumps can be

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configured in a compact design with minimal moving parts and work inputs to reduce the risk of cavitation or pressure surging to the main pump. The primary flow can be diverted from a section of the main pump's high-pressure flow to serve as the entrainment fluid that extracts the secondary flow, which originates from a subcooled or saturated liquid hydrogen tank feed, as illustrated in Figure 1.

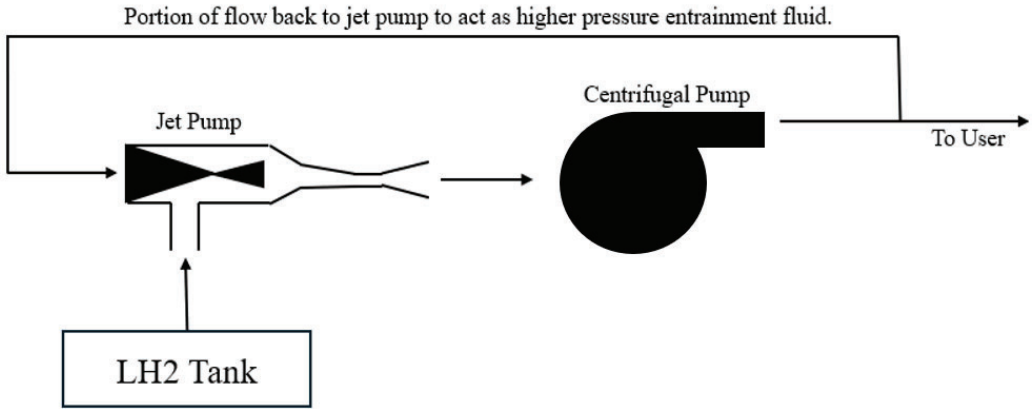

Figure 1: Venturi jet pump schematic being utilized as a boost pump by entraining the secondary fluid from an LH2 fuel tank

2. Venturi Jet Pump Development

A venturi jet pump has two critical components to effectively pump a secondary fluid and increase cooling at the outlet. Figure 2 shows the flow paths between the primary and secondary flow with critical dimensions. A primary converging-diverging nozzle constricts flow to produce sufficient pressure drop, resulting in a sudden fluid expansion at the outlet. A shockwave can be produced if the pressure drop is large enough, causing a low-pressure suction region when critical flow is achieved. This low-pressure region entrains a secondary fluid if a mixing chamber is implemented with the high velocity and fluid momentum of the primary flow. There are constant pressure and constant area mixing chambers that can be utilized downstream of the mixing region. Constant pressure mixing chambers have a diverging cross-sectional area to reduce the pressure drop as the fluids are mixing and result in a higher efficiency [3]. After full mixing is achieved at the end of the diverging mixing chamber, a diffuser allows for sudden expansion and deceleration of the fluid, resulting in a pressure increase and temperature decrease if the geometry is optimized.

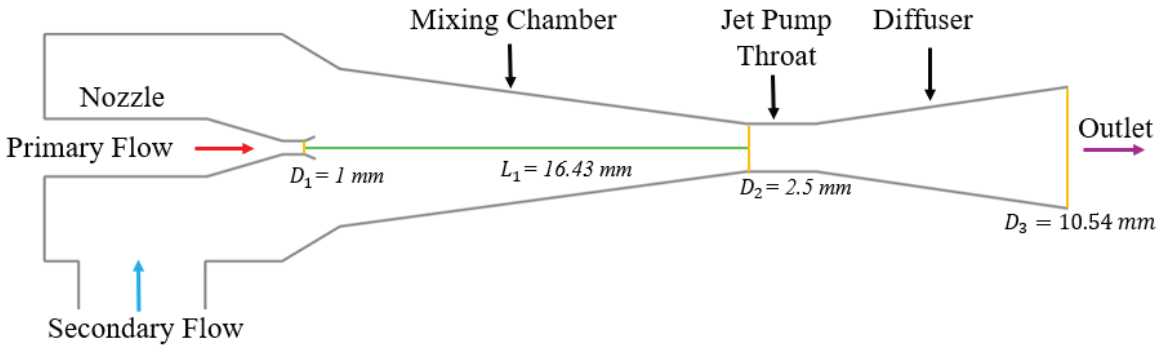

Figure 2: Venturi jet pump flow paths showing primary flow through a nozzle with secondary flow, mixing chamber, and diffuser with critical dimensions being nozzle throat diameter (D_1), outer jet pump diameter where nozzle tip is positioned (D_2), jet pump converged mixing chamber diameter (D_3), and the length from the nozzle throat to the jet pump throat at the end of the mixing chamber (L_1).

The benefit of using a venturi jet pump is the elimination of moving parts, associated with complex sealing surfaces and mechanical work inputs. The low viscosity and density of liquid hydrogen also enable a compact design. To achieve critical flow, much smaller nozzle orifices are necessary to reach a sufficient pressure drop compared to other fluids such as water. A venturi jet pump can be machined from a single piece of material, resulting in minimal sealing surfaces. However, the fluid mechanics of venturi pumps have not been well characterized in liquid hydrogen applications, and there are unknown variables in the models for designing an optimized jet pump. Geometric variables include the primary nozzle throat diameter, mixing chamber diameter, mixing chamber length, mixing chamber convergence angle, diffuser throat, and the position of the nozzle in the mixing chamber. These are shown in Figure 2. Developing a jet pump that allows interchangeable variable nozzles and adjustable nozzle positioning is beneficial for empirical validation of existing jet pump models with liquid hydrogen. A quasi-one-dimensional reduced order model (ROM) was developed in this study, assuming compressible flow with negligible viscous and heat losses, along with a thermal equilibrium model that uses real fluid properties and Leachman's equation of state for hydrogen [4]. Model input parameters include primary and secondary flow pressure, temperature, or quality. This served as the basis for designing an optimized jet pump within the experimental capabilities of the HYPER laboratory, utilizing a Mobile Hydrogen Generation Unit (MHGU), which can supply 3.5 kg of liquid hydrogen at up to 6.5 bara with 2 K subcooling [5]. The resulting critical model dimensions for the jet pump produced are detailed in Figure 2.

The prototype jet pump was produced in two pieces for model validation with varying geometries between the jet pump body and the primary nozzle as shown in Figure 3 and Figure 4. A Swagelok VCR™ fitting was utilized with the primary nozzle welded to a gland and the counterpart welded to the end of the jet pump. A VCR utilizes a copper crush seal. Changing the thickness of the copper gasket allows for varying the nozzle position in the mixing chamber. A thicker gasket will push the nozzle further out of the mixing chamber. A thinner gasket will push the nozzle further into the mixing chamber. In both cases, a seal is maintained in a liquid hydrogen cryogenic environment. The nozzle position can be measured from the outlet of the jet pump to the nozzle tip, which is an input to the reduced-order model.

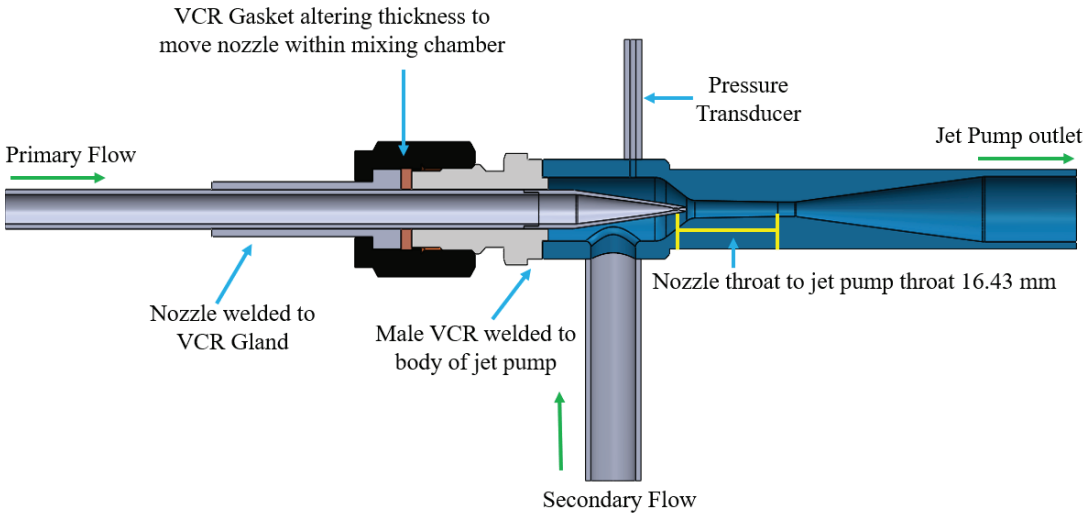

Figure 3: Internal geometry of jet pump with primary nozzle position 16.43 mm from the nozzle throat to jet pump throat and sealed in mixing chamber using a VCR gasket.

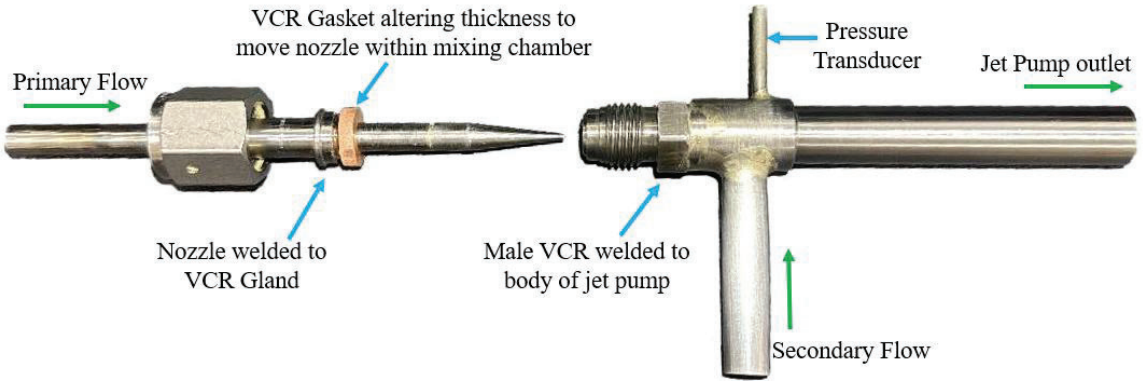

Figure 4: Machined jet pump with primary nozzle position altered and sealed in jet pump mixing chamber with a VCR gasket.

3. Experimental Setup

The system is designed for a maximum pressure of 6.5 bara and can cool down, fill a secondary fluid chamber, and test with no more than 3.5 kg of liquid hydrogen. The full assembly is designed to fit into a vacuum canister and achieve vacuum levels up to 1 Pa to mitigate heat leak effects. Essential fluid measurements to validate the model are the upstream temperature (TS) and pressure (PT) of the jet pump and secondary chamber, downstream pressure and temperature of the jet pump, the downstream temperature of the secondary chamber, pressure in the mixing chamber, and mass flow rate (FM) out of the jet pump. Figure 5 shows the plumbing and instrumentation diagram (P&ID) for the full experimental setup where liquid hydrogen flow can be diverted to either the primary flow or the secondary flow to fill the secondary chamber. Table 1 shows all measurement device uncertainties.

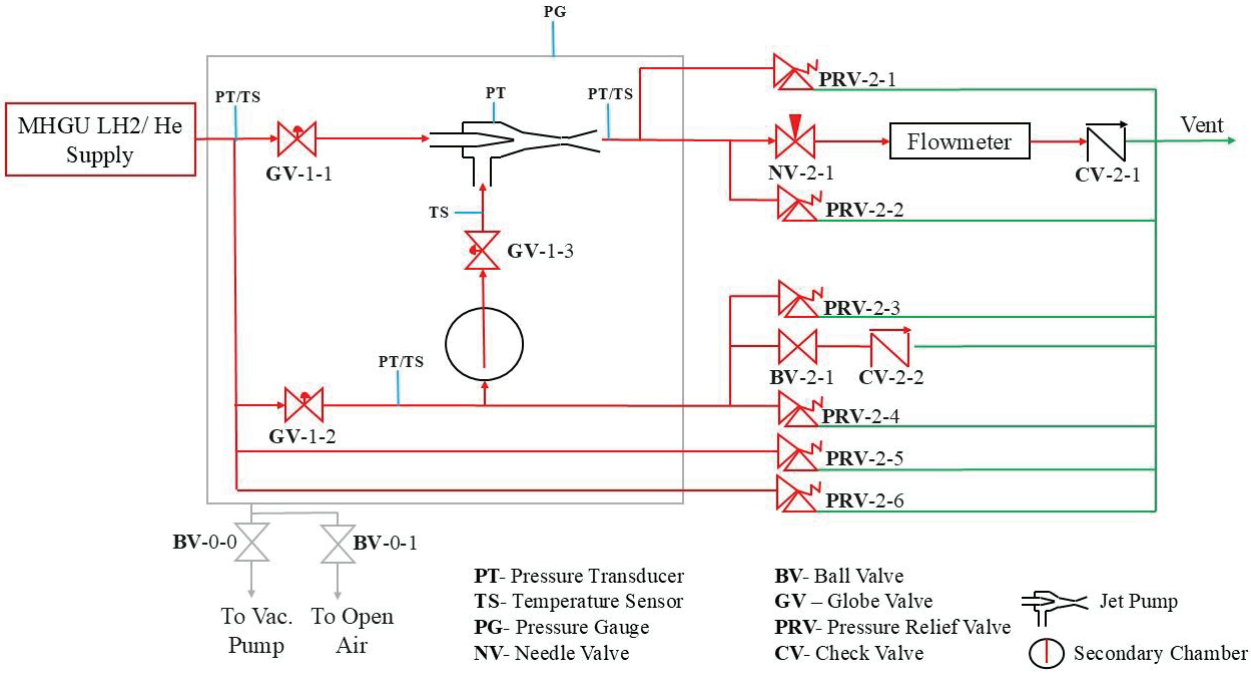

Figure 5: Jet pump test P&ID with single source LH2 from MHGU and flow diversion between primary and secondary flow where PT is pressure transducer, TS is temperature sensor, and FM is mass flow meter.

Table 1: Jet pump experimental device uncertainties.

Measurement Device	Model #	Uncertainty (+/-)	Unit
Cryotechnics Temperature Sensors	SD179-AP-CP	0.2	K
Lakeshore Temperature Reader	Model 336	0.012	K
Core Sensors Pressure Transducers	CS50-2A00150PG	0.069	Bar
Alicat Mass Flow Meter	M-2000SLPM-D/SM	$\dot{m}_{measured} * 0.008$	g/s

A secondary chamber was designed with a 0.5 kg capacity to allow approximately 100 seconds of steady-state full system operation based on the maximum secondary flow rate predicted by the model. A pipe is fed through the top of the secondary chamber to ensure the jet pump secondary flow is saturated liquid and not pulling vapor from the top of the secondary chamber to the bottom as it empties during a test. A flow annulus is built in to allow for the secondary chamber to maintain constant pressure with gaseous hydrogen. A small annulus will allow for gaseous hydrogen introduction with minimal heat load effects when being introduced into the ullage space as the secondary chamber is emptied. Figure 6 shows the structure of the secondary chamber, and the full system built primarily composed of stainless steel 316 to reduce cryogenic and hydrogen embrittlement.

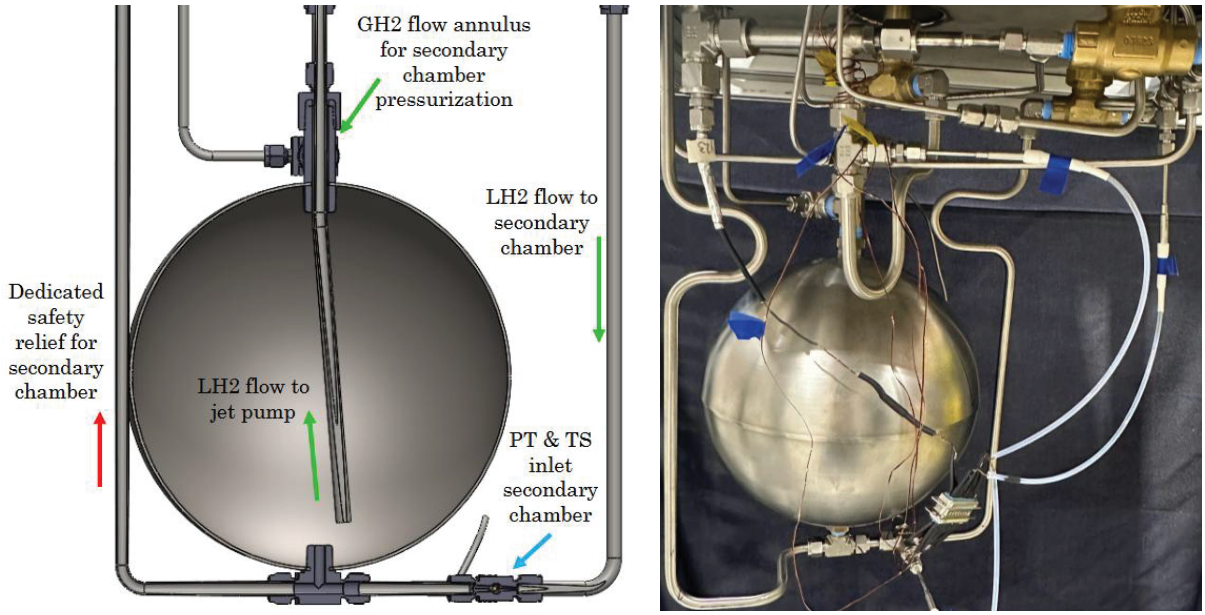

Figure 6: Conceptual and manufactured secondary chamber indicating positions of pressure transducers and temperature sensors up and downstream and the extraction tube to the bottom of the sphere.

The system is prepared for testing first with batch purging using gaseous helium. A 99.999% ultra-high purity hydrogen concentration is acheived by pressurizing the system to 5 bara then de-pressurizing to 1.1 bara 16 times. The system is then cooled down with the liquid hydrogen supply from MHGU through both the primary and secondary flow until the outlet jet pump temperature sensor reaches saturated or slightly sub-cooled liquid conditions. To minimize transient effects, steady state liquid flow is completed for an additional 10 minutes at 1 g/s. After the system is fully cooled down as verified by the downstream temperature and pressure transducer, the globe valve, GV 1-1, to the primary flow is closed to allow for just the secondary chamber to be filled with liquid hydrogen at a low pressure target of 2.5 bara. This secondary tank is utilized to simulate the main fuel tank entrainment of the low pressure fluid. Once the secondary chamber is full, GV 1-2 is closed, GV 1-1 is opened to the primary flow, pressure on the MHGU dewar is increased to 5 bara, and the steady-state jet pump test begins.

4. Results

Two cases for the jet pump test were completed to assess the effects of the secondary to primary pressure ratio on liquid hydrogen pumping. The first is at the pressure ratio (secondary/ primary) target of 0.5 to validate the required pressure differential between the primary and secondary flow to achieve pumping. The second was to test where pumping is no longer achieved with a pressure ratio of 1 where the primary and secondary flow are at the same pressure. The nozzle position tested was 68 mm from the outlet of the jet pump from the nozzle throat shown in Figure 3.

The secondary chamber was filled with liquid hydrogen at a low-pressure target of 3.5 bara and the primary pressure at 5.5 bara. Figure 7 shows the raw data for one of the jet pump tests in detail. There was a comparison to the saturated liquid conditions to analyze if the temperature sensor error was within the saturated liquid deviation. The primary flow had significant sub-cooling of about 1.9 K. However, the secondary flow temperature sensor was very close to the error margin, meaning there is a small amount of uncertainty whether saturated liquid was being pumped or if it was a low-quality 2-phase mixture. Pumping and subcooling were still achieved in the jet pump, even if not saturated liquid fluid conditions. This supports the viability of using the venturi jet pump as a form of pumping and subcooling for liquid hydrogen even if not in the optimized saturated liquid conditions. Approximately 0.45 bar net positive suction pressure with 0.3 K subcooling was achieved. The approximate level of subcooling is determined by calculating the saturated liquid temperature at the recorded pressure. The temperature sensor error is ±0.2 K meaning the error bound indicates close to saturated liquid conditions, however, is out of the bound still supporting viability of subcooling. To validate temperature sensor performance, they were submersed in liquid nitrogen and indicated a reading of 77.24 K within 0.18 K of saturated liquid nitrogen at ambient pressure.

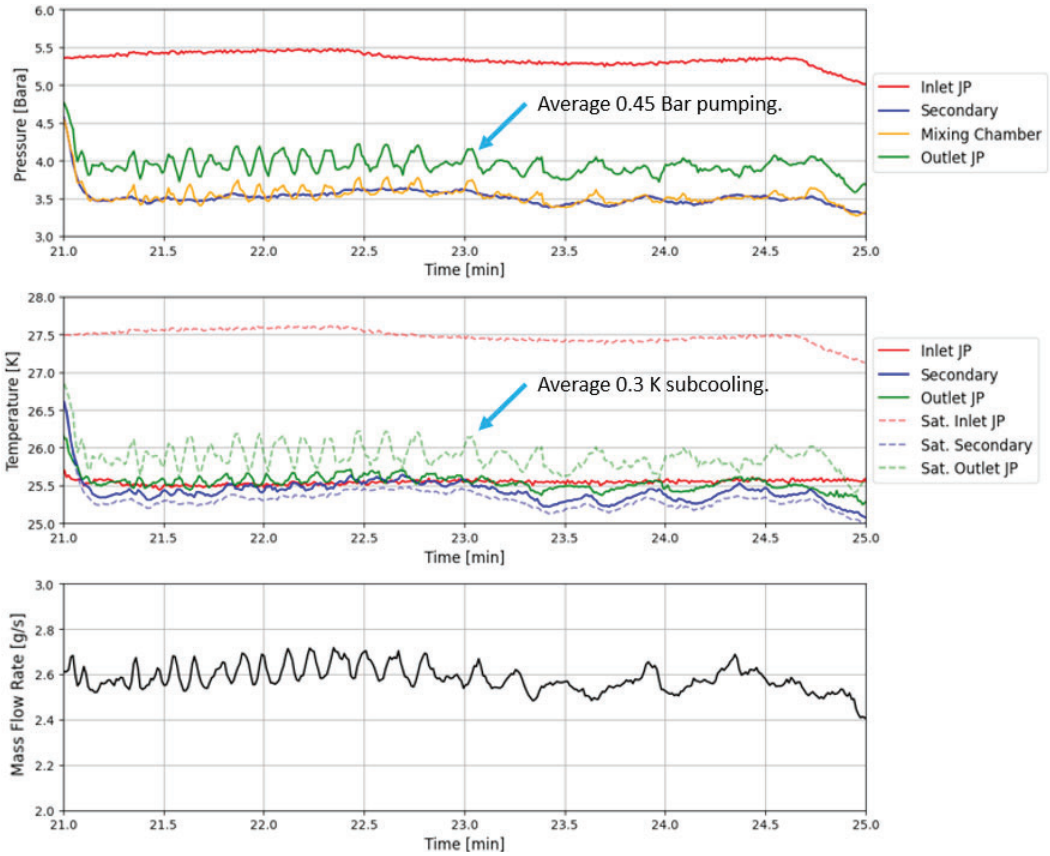

Figure 7: Venturi jet pump with liquid hydrogen steady-state test results with 5.5 bara subcooled primary fluid conditions and 3.5 bara saturated liquid secondary conditions indicating pumping and subcooling at the outlet is achieved.

Table 2: Average experimental jet pump performance achieved 0.43 bar pumping and 0.3 K subcooling at the outlet.

Location	Pressure [Bara] ± 0.069	Temperature [K] ± 0.2	Quality [-]	Mass Flow Rate [g/s] ± 0.02
Inlet JP	5.34	25.54	1.9 K subcooled	-
Secondary	3.50	25.40	~ 0	-
Mixing	3.52	-	-	2.58
Outlet JP	3.93	25.52	0.3 K subcooled	2.58

The second test assessed the effects of pressure ratio (secondary: primary) on fluid pumping. If there is insufficient pressure drop across the primary nozzle, there will be no fluid expansion and velocity increase at the outlet. The minimum pressure drop required was approximately 1.5 bar to achieve fluid pumping. Figure 8 shows the effects of minimal pressure drop for the primary flow. When the secondary pressure approaches 4 bara near minute 66, there is a drop in the outlet pressure to where it is below the secondary pressure, indicating no pumping is occurring. This shows how critical it is to have the right fluid conditions between the primary and secondary flow to achieve sufficient pumping.

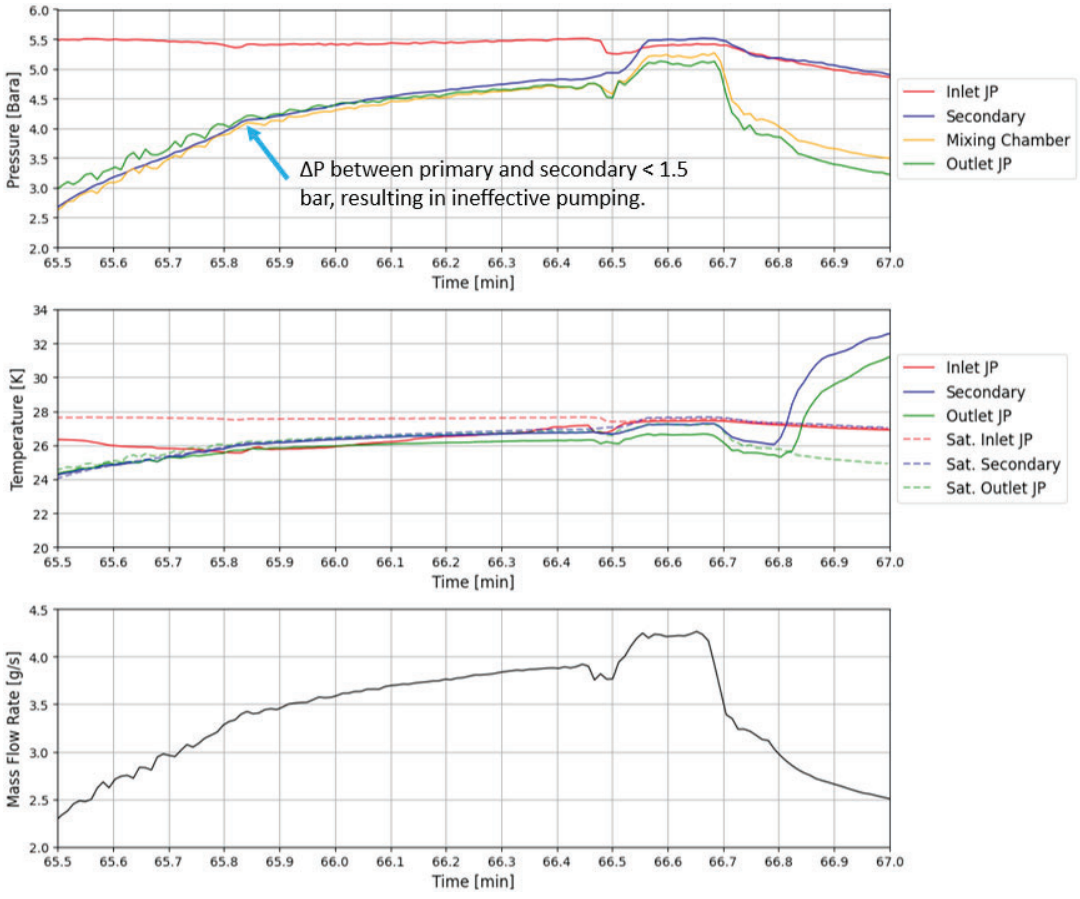

Figure 8: Venturi jet pump with liquid hydrogen transition test results with 5.5 bara subcooled primary fluid conditions and 5.5 bara secondary pressure showing no pumping as the pressure differential was below about 1.5 bar at minute 65.8.

5. Conclusions and Future Work

A liquid hydrogen venturi jet pump was developed to produce net positive suction head and subcooling while minimizing thermal mass, moving parts, and complex sealing surfaces. Three liquid hydrogen tests have been conducted with the nozzle in the theoretically determined optimized nozzle position in the mixing chamber. Preliminary results show that liquid hydrogen cooling and pumping is possible with a 12% pressure increase, even in non-optimal conditions. Further work is needed to determine test repeatability and investigate methods to achieve better subcooling in the secondary fluid to reduce the fluid quality resulting in a lower mass flow rate than theoretically predicted.

Additional work will verify the effect of nozzle position on secondary fluid entrainment and overall system pumping. Using a converging/ diverging mixing chamber, there could be significant inefficiencies in the jet pump. If the nozzle is positioned non-optimally from the mixing chamber, an excessive pressure increase can occur within the chamber, resulting in shockwave formation in the throat of the jet pump just before the diffuser. Two additional nozzle positions will be tested with this experimental setup: one test will involve positioning the nozzle farther into the mixing chamber, and the other will place the nozzle further out of the mixing chamber. This data will then be utilized to validate the reduced order model that is in development for use in liquid hydrogen jet pump performance prediction in larger scale applications. These non-optimized test cases will increase our understanding of how nozzle positioning affects liquid hydrogen flow, allowing for the design of more optimized jet pumps to improve efficiency. Preliminary test data supports the feasibility of utilizing venturi jet pumps in a compact design with minimal moving parts to generate net positive suction pressure and subcooling for downstream component usage.

Acknowledgements

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