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Liquid pump-enabled hydrogen refueling system for heavy duty fuel cell vehicles: Pump performance and J2601-compliant fills with precooling

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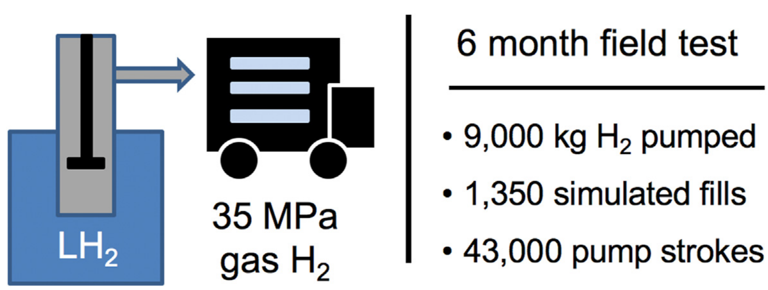
HIGHLIGHTS

- A liquid H_2 pump-enabled solution for 35 MPa hydrogen refueling has been developed.
- Results from field testing of a full-scale mobile trailer system are reported.
- Pump performance: 40 MPa outlet pressure; up to 285 kg/h; 43,000 pump strokes.
- System performance: 1350 J2601-2 compliant fill cycles, including back-to-back fills.

GRAPHICAL ABSTRACT



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ABSTRACT

We have developed a hydrogen (H_2) refueling solution capable of delivering precooled, compressed gaseous hydrogen for heavy duty vehicle (HDV) refueling applications. The system uses a submerged pump to deliver pressurized liquid H_2 from a cryogenic storage tank to a dispensing control loop that vaporizes the liquid and adjusts the pressure and temperature of the resulting gas to enable refueling at 35 MPa and temperatures as low as −40 °C. A full-scale mobile refueler was fabricated and tested over a 6-month campaign to validate its performance. We report results from tests involving a total of 9000 kg of liquid H_2 pumped and 1350 filling cycles over a range of conditions. Notably, the system was able to repeatably complete multiple, back-to-back 30 kg filling cycles in under 6 min each, in

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J2601
Liquid hydrogen
Pump
Refueling station

full compliance with the SAE J2601-2 standard, demonstrating its potential for rapid-throughput HDV refueling applications.

Introduction

Hydrogen (H_2) fuel cell vehicles (FCV) are an important complement to battery electric vehicles (BEV) in efforts to decarbonize ground transportation, particularly for use profiles which require high power, long operating range, or rapid refueling [1]. One prerequisite for the widespread adoption of FCVs is a refueling network. Two barriers must be addressed in the development of an affordable H_2 fuel supply chain. First, the cost of dispensed H_2 must be reduced to make the total cost of operation competitive against alternatives. Second, the capital costs of stations must be reduced to allow an adequate network of H_2 refueling stations (HRS) to be established. Both barriers can be addressed through the development of new concepts that can significantly reduce the capital and operating costs of the H_2 refueling step.

Current refueling practice involves dispensing H_2 as a compressed gas into onboard vehicle storage tanks. Refueling station designs can be grouped into two categories, according to whether H_2 is stored on-site as a compressed gas or a cryogenic liquid [2]. Stations with on-site gaseous H_2 storage require gas compressors to charge high pressure ground storage (*e.g.*, cascade tubes), which then deliver fuel to the vehicle. Designs using gas compressors have several limitations that contribute to high costs. First, compressors are energy-intensive and also have high maintenance costs. Second, the largest gas compressors which practically fit in the footprint of an urban HRS on the market today are only capable of approximately 40 kg/h [3–8]. This severely limits the rate at which vehicles can be refueled and requires significant investment in cascade storage to accommodate high vehicle throughput. This, in turn, increases the station footprint since high pressure storage requires roughly 10 m^2 per 130 kg stored[1]. Finally, for fast filling capability, refrigeration units precool the gas to offset heating when the gas is dispensed to the vehicle tank.

Stations with liquid H_2 (LH_2) storage have the potential to overcome these limitations [9]. The use of a liquid pump to pressurize the fluid before it is vaporized and dispensed to the vehicle can reduce the energy required for compression and enable direct filling to reduce or eliminate ground storage. Heat integration with the vaporization process can also eliminate the need for refrigeration. Together, these features can simplify the process flowsheet, decrease the station footprint, and reduce capital and operating costs. These advantages must be balanced against the added life cycle costs of producing LH_2. Generic analyses indicate savings from distribution and at the station more than offset the added costs of liquefaction, but detailed analysis is needed for each situation [10]. In practice, LH_2 stations are limited by the performance of existing pumps. Current pumps are external to the storage tank necessitating cooldown of the pump upon startup, with concomitant delays and boil-off losses; they also have limited seal life resulting in high maintenance costs [3].

To address these issues, we have developed a LH_2 pump that can be installed submerged in a cryogenic storage tank. In this paper, we describe the design, construction, and field validation of a LH_2 refueling solution using our submerged pump to deliver precooled 35 MPa, maximum pressure up to 41.3 MPa, compressed gaseous H_2 down to −40C for heavy duty vehicle (HDV) refueling applications. The discussion is divided into four parts. The first part introduces the liquid pump and other core components of the system. We also describe our mobile refueler and a simulation trailer used for validation testing. The second part reports the performance of the liquid pump, with an emphasis on flow rate and pressure control, and key performance indicators including volumetric efficiency and energy requirement. The third part covers the system performance. We summarize results from over 1350 filling cycles over a range of conditions, involving a total of over 9000 kg LH_2 pumped. Detailed results showing the ability of the system to repeatably complete multiple, back-to-back 30 kg filling cycles in under 6 min each, in full compliance with the SAE J2601-2 standard, are presented to demonstrate the potential of the system. The final section discusses the implications for future hydrogen refueling station designs.

NICE America's hydrogen refueling system

NICE America has designed, constructed, and tested a high-pressure liquid hydrogen pump and related refueling components for 35 MPa hydrogen refueling. The liquid hydrogen refueling station (LHRS) along with all the refueling components was mounted on a trailer as shown in Fig. 1.

General description of the LHRS concept

The hydrogen refueling system is a core feature of the hydrogen infrastructure. Key challenges for the deployment of a hydrogen infrastructure for road transport are the

[1] At 350 bar, the density of compressed gaseous H_2 is 23 g/L. The volume occupied by 130 kg of gas at this pressure is approximately 3.0 m^3. A typical diameter for a high pressure cascade tube is about 0.5 m i.d.; larger tubes need thicker walls and this makes them uneconomical. The footprint is estimated by assuming cascade tubes are arranged in a 3 high × 4 wide bank, and the spacing between the tubes is 0.5 m. In this arrangement, the width of the bank is 4 m and the length is 1. 27 m (3.0 m^3/12 tubes/cross-sectional area). The footprint of the storage system is approximately 10 m^2. Setback distances are not included in this estimate of footprint.

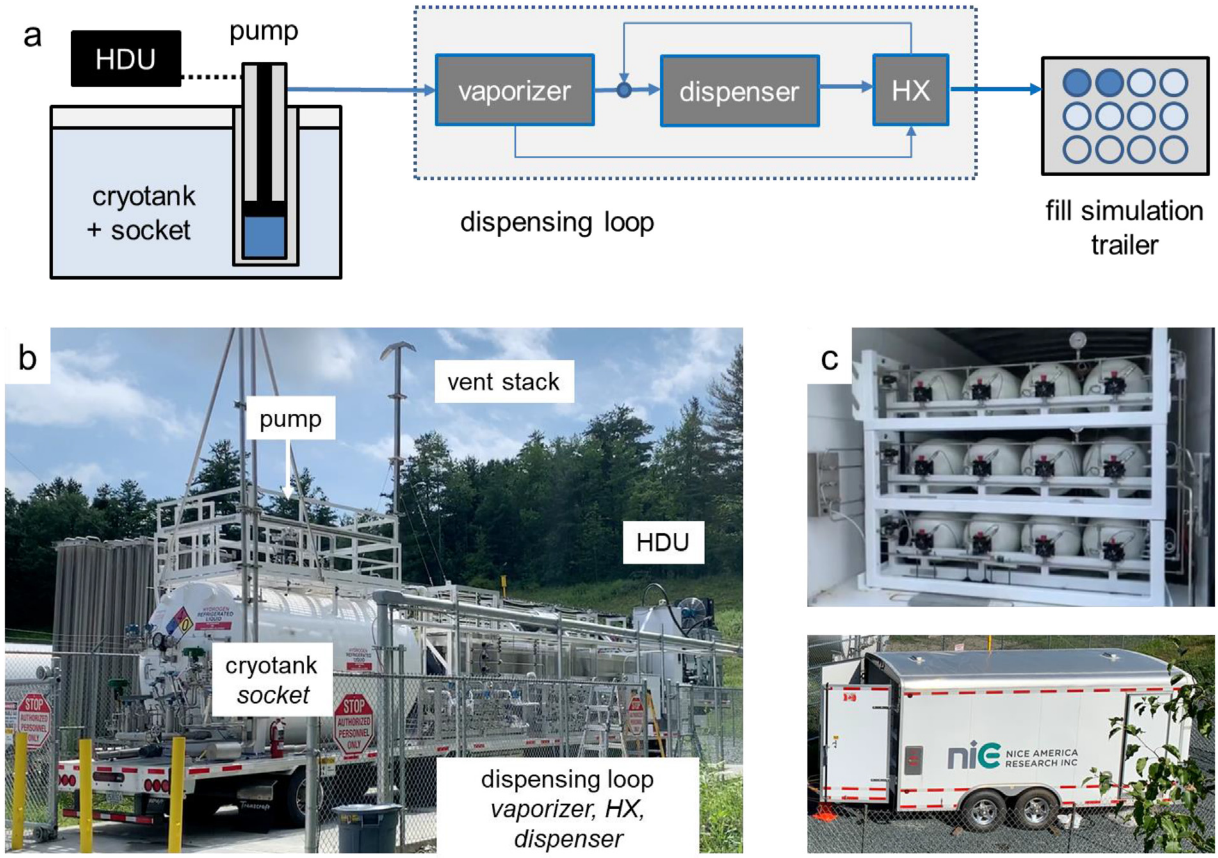

Fig. 1 – NICE LHRS system. a. Schematic layout of the system, including LH2 cryotank and socket, liquid H_2 pump and hydraulic drive unit (HDU), dispensing loop with vaporizer, heat exchanger, and dispenser, and fill simulation trailer. b. Photograph of the 53 ft mobile station trailer, set up for testing at the demonstration site. c. Photographs of the 35 MPa vehicle fill simulation trailer interior (top) and exterior (bottom).

significant capital requirement for equipment and the large physical footprint of refueling stations. Moreover, currently available refueling station designs have several technical limitations. Compressed gas stations require energy-intensive compression, the need for expensive cascade tubes for intermediate storage, and refrigeration which can limit startup times. Liquid H_2 stations, depending on design may also have some of these drawbacks, and also suffer from liquid boiloff losses during start-up. The NICE LHRS design addresses these technological issues through specific design features. These technology advancements are necessary to reduce capital and operating costs, to decrease land requirements, and to increase fueling capacity. Table 1 summarizes the novel features of the system. The benefits of these features will be discussed in the *Implications* section.

NICE mobile refueler system

The NICE LHRS test unit is fully integrated onto a 53 ft trailer, allowing it to be moved to different locations. The mobile refueler system comprises two units, the refueling station unit and a vehicle simulation unit, shown in Fig. 1. Table 2 lists the key components of the system.

Safety features

Potential hazards with the cryogenic H_2 fueling system include high pressure, low temperature, and explosion risks. Safety management was implemented at multiple levels from system design, to hardware and control system software protection, to standard operating procedures, to personnel protection equipment and operator training. On the system design and hardware protection levels, the system was designed to comply with ASME boiler and pressure vessel code, section VIII, Division 1; ASME B31.12, hydrogen piping and pipelines; NFPA 2, hydrogen technology code; and National Electrical Code. All enclosures are NEMA4 with Z-purge or better, per NFPA 496. Since the test system is intended for outdoors, all components are IP66 rated for ingress protection. Four hydrogen detectors were strategically located to detect leaks, and a combined UV/IR fire eye was positioned to have full view of the dispenser and the cryotank. Pressure relief valves are plumbed directly to CGA G-5.5 compliant vent stacks to guard against over-pressurization of the system. The system is equipped with multiple emergency stops located next to the operator panel at the fore of the trailer, at the rear next to the cryogenic tank, next to the vehicle simulation system, on the

Table 1 – Features of NICE LHRS.

Key Feature	Advantages
Liquid hydrogen	Pumping LH_2 is up to 10 times more efficient than compressing gas.
Submerged pump	Eliminates the thermal cycles that cause failures in the equipment, slow start-up times, and boil-off losses. Allows LHRS to start almost instantly.
Hydraulic drive unit (HDU)	HDU allows physical separation of the drive motor from cryotank, enabling placement of the pump directly inside the liquid tank without introducing motor heat or vibration. Hydraulic actuation also provides additional turndown control so that the pump is capable of smooth operation.
Large, slow-moving piston	Minimizes seal travel and extends pump seal lifetime. With a large pump, high flow rate is possible with a small number of piston strokes. Fewer and slower piston strokes reduces seal wear rates, leading to improved reliability.
Direct-filling of multiple vehicles consecutively	High flow pump can fill multiple fuel cell vehicles back-to-back without the need of a cascade storage system.
Precooling	Refrigeration is provided by heat integration with a slipstream of pumped LH_2 within the dispensing loop. The thermal management subsystem controls the slipstream flow rate so that the target H_2 dispensing temperature (*e.g.*, −40 °C) is achieved without the need for separate refrigeration units.

Table 2 – LHRS Key component.

Component	Description
Cryogenic H_2 storage tank	Vacuum-insulated 1500 US gallon (450 kg LH_2) tank with integrated pump socket. The cryotank includes extra structural support for dynamic load during road transport. The tank is compliant with the MC338 code*.
Submerged LH_2 pump	The single-acting piston pump delivers H_2 at a design average flow rate of 230 kg/h (maximum tested to 285 kg/h) and pressures up to 45 MPa.
Hydraulic drive unit (HDU)	The HDU is built on a self-contained integrated skid. Hydraulic oil temperature is self-regulated via a side pump, a cooling fan, and an immersion heater, allowing operation over wide range of ambient temperature (−10 to 30 °C). The hydraulic flow rate and direction is powered by a 150 kW motor.
Control and power system	Power supply contained in a purged power distribution cabinet (PDC). Main programmable logic controller (PLC) and a human-machine interface (HMI) housed in an unpurged cabinet.
Instrument air system	The instrument air system consists of 80-gallon pressure storage, a compressor, air dryer, and several regulators. It is designed to generate sufficient air pressure to operate automatic valves of the entire system and purge the power cabinet. A snorkel is used to bring air into the compressor from outside the classified zone.
Dispensing system	The dispensing system consists of all the supporting components and auxiliary systems of the refueling station to enable code compliant vehicle refueling. Key components include: a proportional pressure controller; a forced-air vaporizer; and a thermal control unit (TCU) enabling flexible fueling temperature control down to −40 °C. The vaporizer provides up to 270 kW heat duty, and is powered by a 7.5 kW forced-air fan motor.
Simulation trailer	The simulation trailer contains of twelve 312.9 L gaseous H_2 pressure vessels (7.5 kgH_2 per vessel at 35 MPa, carbon fiber wrapped Type III), in a 3-bank by 4-vessel arrangement, each isolated by a manual valve and a solenoid valve in series. The vessels are individually plumbed, allowing any number of vessels in combination for maximum flexibility during testing. For example, each bank can simulate a 30 kg H_2 vehicle.

* 49 CFR 178.338 - Specification MC-338; insulated cargo tank motor vehicle. https://www.govinfo.gov/app/details/CFR-2011-title49-vol3/CFR-2011-title49-vol3-sec178-338.

power distribution cabinet, and on the entry door to the test system enclosure fence. Electrical panels are purged with air, supplied from a snorkel more than 50 ft away from the system. On the control system software protection level, select stops, interlocks, and permissives were implemented. For example, the hydrogen detectors and the flame detector are managed by the control system in the safety circuit to take action, including immediate shutdown of the system. Standard operating procedures for operators include specification of hazards, along with required personal protective equipment.

Liquid H_2 pump

A key enabler of NICE LHRS system is the submerged high-pressure LH_2 pump. The hydraulically-driven reciprocating pump is installed inside a fitted, insulated socket. At the bottom of the socket is a foot valve which can isolate the pump from the rest of the tank during maintenance. The LH_2 pump is a single-acting, double-chamber piston pump. The direction of hydraulic flow from the HDU determines the piston pump moving up (a retract stroke) or down (an extend stroke). When the hydraulic flow pushes the piston up, the bottom piston draws LH_2 through the intake check valve. When the hydraulic flow direction switches, the bottom piston also reverses the direction and discharges the LH_2 at high pressure through the discharge port. Simultaneously during an extend stroke, back of the bottom piston draws LH_2 through a check valve, ensuring the pump intake port is in contact with cold LH_2 from the cryotank. When the piston reaches the bottom, an unloading valve in the hydraulic piston opens to relieve the high pressure, eliminating the need to switch the hydraulic flow direction immediately and preventing the hydraulic piston from impacting the cylinder head.

Dispensing loop

The second enabler for the NICE LHRS system is the dispensing loop. This subsystem enables direct filling with active temperature control, and replaces cascade storage and active refrigeration in conventional station designs. Direct filling is possible because the cryogenic piston pump is active throughout the duration of any fill; the dispensing loop contains three components that work together to vaporize the LH_2 at high pressure and adjust the dispensing temperature of the resulting compressed gaseous H_2 to meet dispensing requirements. Fig. 2 shows a process flow diagram of the dispensing loop. The pump discharge flows through an ambient temperature vaporizer. A slipstream bypasses the vaporizer and provides the cooling medium to control the temperature of the stream downstream of the vaporizer. A PID-controlled flow regulator on the bypass stream determines the amount of hydrogen flow to split away from the mainstream for dispensing temperature control. This bypass stream is recombined with the main gas stream prior to entering the heat exchanger. Using cooling from a bypass stream allows flexible dispensing temperature control for back-to-back filling, and overcomes a drawback of station designs that use active refrigeration based on cooling blocks (*e.g.*, heat exchangers, large metal blocks).

Experimental results and discussion

Table 3 summarizes the outcomes from the 6-month test campaign conducted in 2020. The test campaign was divided into three phases: commissioning; pump performance validation; and system performance validation. Commissioning activities included safety reviews, shakedown of all system components, and initial fill and cooldown of the cryotank. Active testing was performed on 91 days during the campaign. Pump validation included verification of pump operating ranges, and optimization of performance. System validation involved simulated fills under a range of conditions, including multiple back-to-back cycles.

A computerized Reliability Centered Maintenance (RCM) database system was implemented to track, down to individual component level, the root cause, duration, and outcome of each downtime, whether planned or unplanned. Data from this RCM system allows statistics on component, subsystem, and system level. In this paper, system reliability measures the impact of unplanned corrective maintenance on system runtime, while system availability reports total downtime – planned or unplanned – on system runtime. Although the submerged pump had no downtime, other components such as sensors and valves required maintenance. The single largest system downtime was caused by the main electric circuit breaker due to water ingress to the power distribution panel, which accounted for half of the lost availability.

Table 3 – Summary of test campaign.

Test campaign period	Jun–Nov 2020
Days of active testing	91 days
Total number of pump strokes	43,000
Total LH_2 pumped	9000 kg
Total simulated fill cycles	1350
System reliability	98.5%
System availability	97.3%
Safety incidents during the campaign	No incidents

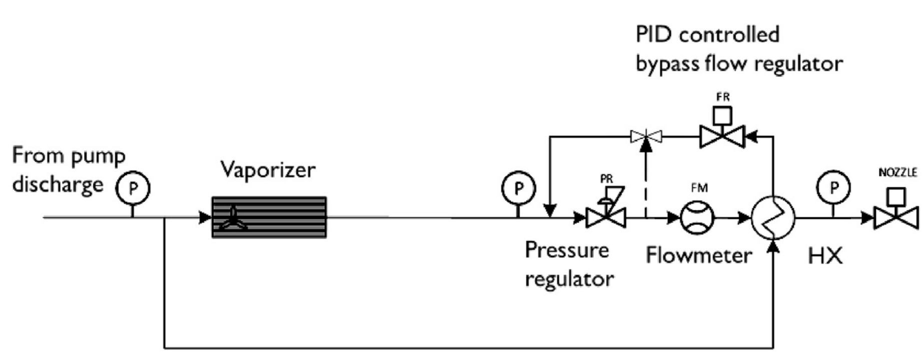

Fig. 2 – Process flow diagram for dispensing loop.

Liquid H_2 pump performance

Fig. 3 shows the operational capability of the pump to deliver high pressure H_2 over a range of discharge flow rates and outlet pressures. Data points represent stable operation of the pump for a minimum of 20 pump strokes at the specified conditions. The pump was able to meet its design performance target of 230 kg/h at 45 MPa. The maximum sustained average flow rate was 285 kg/h, but the outlet pressure could only be maintained at 44 MPa at that flow rate. The pump could also be turned down to flow rates as low as 10 kg/h, with fully controllable outlet pressure up to 45 MPa.

In addition to discharge flow rate and pressure, the volumetric efficiency (VE) and specific energy consumption are other key performance indicators. Volumetric efficiency is the ratio of the actual and theoretical mass flow through the pump, a measure of the pump piston stroke volume utilization. Actual flow is obtained from mass flow measurements out of the pump, while the theoretical mass flow is the product of the cylinder volume and the intake fluid density. Losses in volumetric efficiency occur due to leakage through seals, and vaporization of liquid due to heat leak; inlet fluid conditions (*e.g.*, temperature of the LH_2 in the tank) can also impact volumetric efficiency.

Fig. 4 shows the ranges of volumetric efficiency computed over different operating conditions. The range shows the combined effects of pump operating details (*e.g.*, stroke profile) and variability contributed from other system parameters (*e.g.*, temperature of the LH_2 in the tank). The maximum VE possible appears to be insensitive to the discharge pressure, although at the higher discharge pressure it takes a larger fraction of the stroke to compress the LH_2 and leaks across the hydrogen piston seals are more likely. Conversely, at lower discharge pressures, particularly in the 20–25 MPa range, the VE under some conditions drops due to the slower piston stroke associated with such operation. This results in a longer time to complete the stroke, allowing more time for leaks at the piston seals and intake check valve (albeit with lower pressure driving force). The VE at the design conditions of the pump (230 kg/h and 45 MPa) was

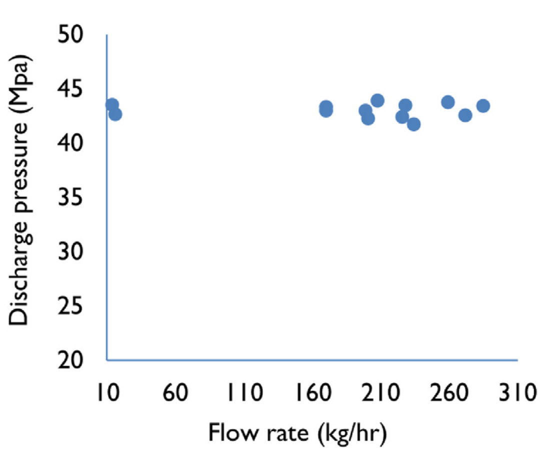

Fig. 3 – Operational capabilities of the LH_2 pump. Pump operating range showing pump capability with respect to discharge pressure and flow rate. At 285 kg/h, the HDU reaches its flow limit and its compensator limits the hydraulic pump displacement.

93.6%. The discharge flow rate, which is roughly proportional to the pump stroke rate, had minimum impact on VE since this metric is computed on a per stroke basis.

Specific energy consumption is electricity use per unit hydrogen pumped. Fig. 5 plots the energy consumption as a function of VE over the range of operating conditions during the validation campaign. Energy use is correlated with VE; a higher VE indicates more effective delivery of fluid by the pump, and a lower energy demand on a normalized flow basis. The effects of discharge pressure and flow rate on VE can also be seen in Fig. 5. Energy requirements increase with discharge pressure since more work is needed to pressurize the LH_2 from tank conditions to the discharge conditions. Practically, this manifests in a negative correlation with VE, while remaining consistent with the need for a longer stroke to maintain VE at higher discharge pressure. The minimum energy consumption at the design conditions of the pump (230 kg/h and 45 MPa) was 0.2 kWh/kg.

Simulation of precooled filling cycles for heavy duty vehicle refueling

H_2 refueling operations for HDV must comply with the SAE J2601-2 standard [11]. This standard is a performance-based protocol that establishes boundary conditions for pressure and flow rate throughout a refueling cycle to ensure safe filling for 35 MPa HDV. We performed simulated filling cycles ranging from 7.5 to 60 kg, using our simulation trailer. For fill cycles corresponding to transit agency bus refueling, three (22.5 kg) or four vessels (30 kg) were filled in parallel for each simulated bus.

In addition to compliance with the J2601-2 standard, additional desirable performance features for HDV filling systems include:

- Precooling, to allow short fill times. A key metric for refueling performance is the filling time. The filling process compresses the gas in the tank, leading to heating as the tank is pressurized. Precooling of the dispensed gas partially offsets this temperature rise, allowing higher flow rates. A benchmark target for filling time is less than 10 min for a 30 kg fill.
- Full charge. The final state of charge (SOC) in the vehicle tank needs to be above 95% but less than 100% to allow maximum vehicle range without storage system overpressurization. Precooling is an important measure to achieve a full charge without increasing the vehicle storage system pressure to unsafe levels.
- Fast start. The time needed to commence filling also contributes to the filling time. Conventional systems relying on mechanical refrigeration must expend energy to keep cooling blocks at operating temperature, or delay refueling until the cooling blocks reach operating temperature. The J2601-2 protocol requires the dispensing temperature to reach target range within 30 s, and any delay increases the total vehicle fill time.
- Back-to-back fill capability. This capability impacts station throughput. Ideally, the number of vehicles a system can service should be limited by the fuel storage supply at the station, and the vehicle throughput should be as fast as possible. Conventional designs that use cascade tubes or active refrigeration experience limitations in the number

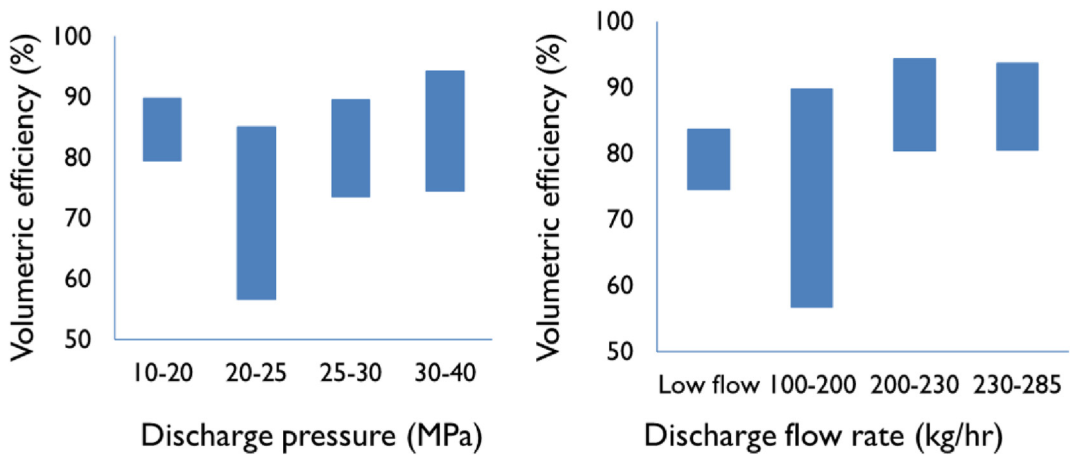

Fig. 4 – Effects of pump discharge pressure (left) and hydrogen flow rate (right) on VE.

of vehicles that can be refueled without delays. Throughput, capacity, and back-to-back fill capability can be improved through additional equipment, but this increases the capital costs of the station.

- High storage tank utilization. Currently available liquid pumps require a minimum net positive suction head (NPSH). Practically, this means cryotanks must remain at least 30% full to ensure adequate pressure head at the pump inlet to prevent cavitation. Higher storage tank utilization could increase the capacity and throughput of the station, extend the interval between refilling deliveries, and improve the capital efficiency of the cryotank.

In this section, we present technical data showing the ability of the NICE LHRS system to deliver each of these features. Boil-off of LH_2, both during operation and due to static heat leak into the tank, will be discussed in a separate section.

Fast fill and fast start

Fig. 6 shows H_2 flow, pressure at the dispenser, dispensing temperature, and SOC data for a single non-communication 30 kg fill, performed with precooling consistent with a J2601-2 compliant T20 fill with a range of −17.5 to −40 °C. The precooled fill was completed in 7.2 min. The pressure spikes at the beginning correspond to pulses called for by the J2601-2 protocol to determine the volume and SOC of the vehicle tank; the short pause occurring near the midpoint of the fill cycle corresponds to a leak check, also mandated by the protocol. The oscillations in the flow rate and pressure at the dispenser correspond to individual pump strokes; these are detectable given the temporal high resolution of the controls and monitoring system. The smooth increase in both tank pressure and SOC confirm that the fill was compliant with J2601-2 bounds during the entire cycle.

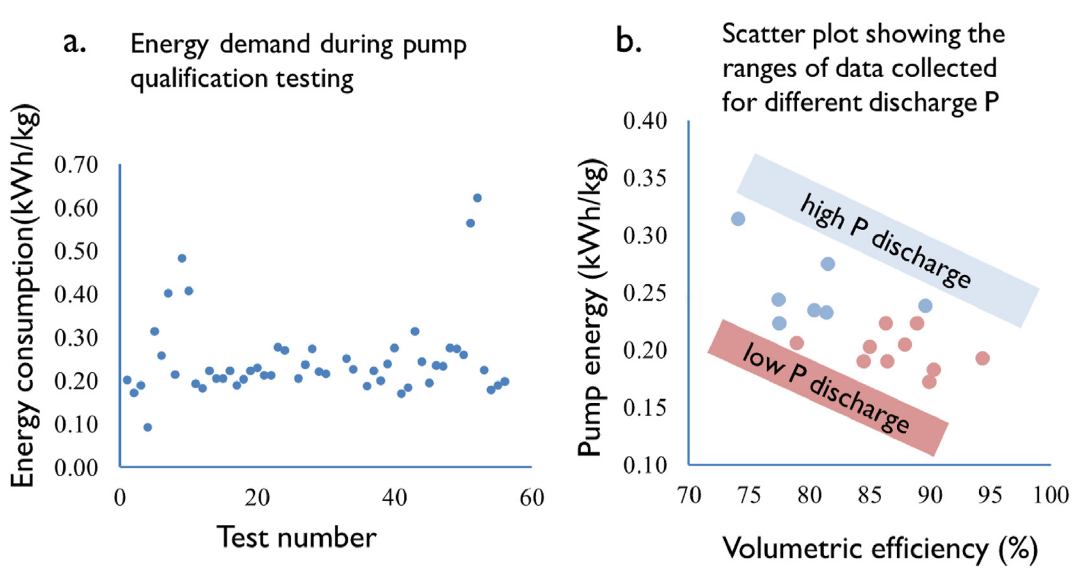

Fig. 5 – Pump energy consumption. a. Plot of pump energy consumption for each of the pump qualification tests. b. Plot of pump energy consumption versus volumetric efficiency. The different colors for the data points correspond to the specified ranges of discharge pressure during each corresponding test. The high and low P discharge ranges were 36.8 and 20.8 MPa, respectively. (For interpretation of the references to colour in this figure legend, the reader is referred to the Web version of this article.)

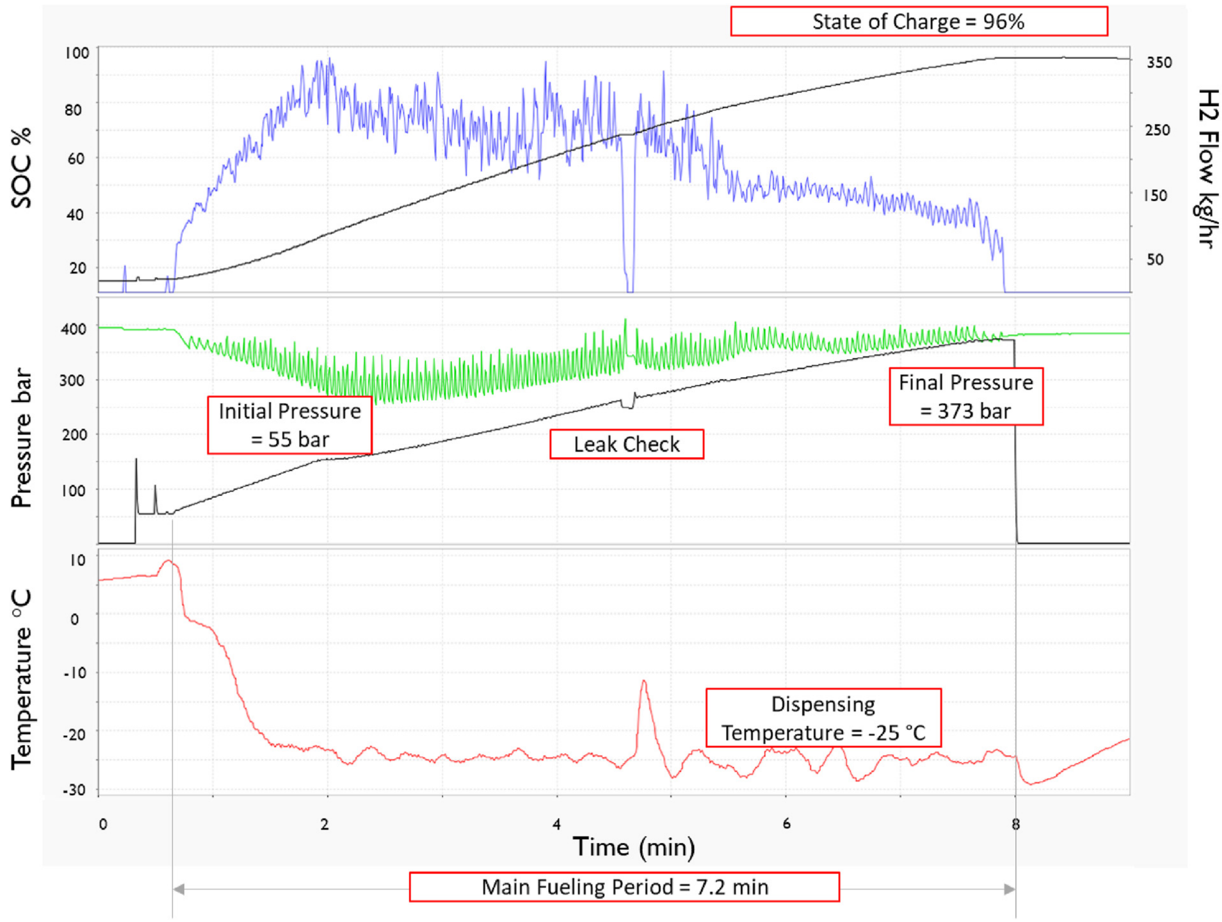

Fig. 6 – Single 30 kg fill cycle with precooling at -33°C. The top panel shows the instantaneous flow rate (blue, right axis) measured at the dispenser over the duration of the fill cycle, and state of charge in the vehicle tank (gray, left axis). The middle panel shows traces for the pump discharge pressure (green) and pressure measured at the dispenser (black). The bottom plot shows the dispensing temperature (red, dotted). (For interpretation of the references to colour in this figure legend, the reader is referred to the Web version of this article.)

In the precooled fill, flow commences after the protocol mandated initialization and leak check sequence. The initial tank pressure was 55 bar, corresponding to an SOC of 15%. The dispensing temperature reached −25 °C. The J2601-2 mandated leak check correctly executed at around 262 bar, a little after 4 min into the fill. The system delivered a total of 25 kg H_2 within 7.2 min and brought up the SOC to 96%.

Back-to-back fills

Multiple simulations of back-to-back fills were performed to assess the performance of the system under conditions that might be encountered in transit bus refueling applications. In real-world applications, buses arrive at the station with different SOCs. To test this scenario using our simulation trailer, we configured the system to sequentially fill three groups of four vessels (30 kg total capacity). Each set of vessels was interconnected and set to a different starting pressure to simulate different starting SOCs for each fill cycle. The total number of consecutive simulated fills is limited to three because there is a limited number of vessels to fill in the bus simulation trailer, and rapid defueling of previously filled vessels causes their temperature to go below limit. Each fill cycle was performed with the goal of reaching SOC >95%, in full compliance with J2601-2 protocol requirements, with a refueling time less than 10 min for a full fill [12]. In addition, the second and third fills were scheduled to begin 2 min after completion of the previous fill. This interval mimics the time needed to drive buses through the station consistent with service depot operating practice.

Fig. 7 shows data for three back-to-back fills targeting a final capacity of 15 kg, 15 kg and 30 kg. The initial pressure of the vessel sets used for the first, second, and third fills was 34, 24 and 38 bar, respectively. The dispensing temperature setpoint was set to −33 °C to simulate T40 fill cycles. The plots show the flow rate and pressure at the dispenser. During the first fill, the SOC was raised from 4% to 98% by moving 12.5 kg over 5 min. In the second fill, the SOC was raised from 3% to 98% by moving 13.1 kg H_2 over 5 min. In the final fill, the SOC was raised from 5% to 97% by delivering 25 kg H_2 over 9 min.

For all three fills, the final state of charge (SOC) exceeded 95%, and the lengths of main fueling time stayed within the satisfactory range. There were no qualitative differences in the ability of the system to deliver appropriate flow rates during any of the fills. In addition, the system was able to

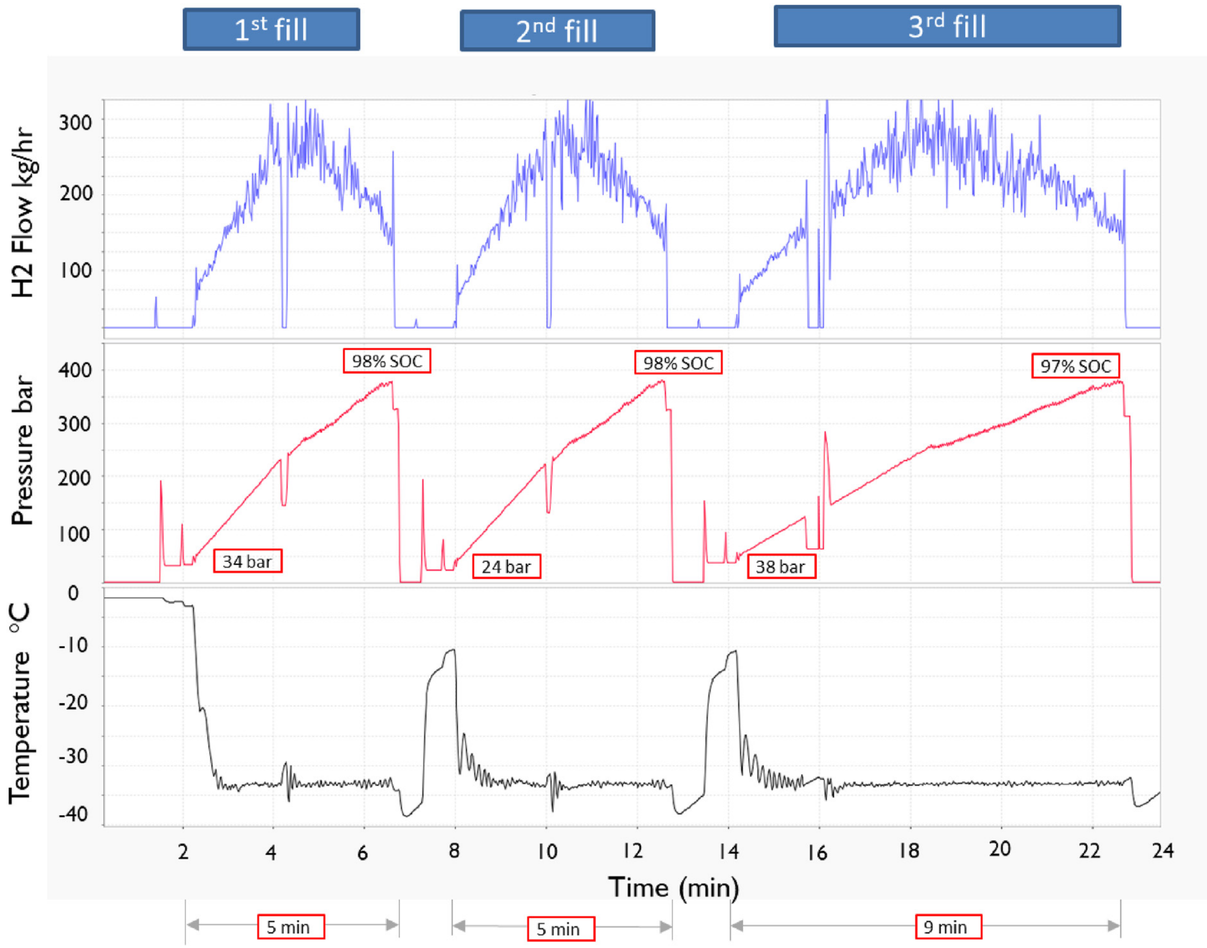

Fig. 7 – Back-to-back filling demonstration. Three back-to-back fills (15 kg, 15 kg, and 30 kg) were performed with different starting pressures in the simulated vehicle tanks.

actively maintain the dispensing temperature at the −33 °C setpoint during the entire set of three fill cycles.

The data shown in Figs. 6 and 7 are representative of the performance of the system over a larger set of simulated fill cycles. In total, the system was used to simulate 90 single fills (ranging from 7.5 kg to 60 kg) and 97 back-to-back fills (ranging from 7.5 kg to 75 kg) for multiple vehicles in a row. Fig. 8 shows the number of back to back fills conducted for 2 to 10 vehicles in a row over the testing period. X-axis in Fig. 9 represents the number of consecutive vehicles filled during back-to-back fill. For example, B2B6 stands for back-to-back fill conducted for 6 vehicles in a row.

Tank utilization

The working capacity of our cryotank was about 400 kg. During extended testing periods, the tank was refilled on a regular basis to provide sufficient LH_2 for operations. Fig. 9 shows the tank level throughout the test campaign. Tank levels in June were initially maintained at high levels between refilling due to the high boil-off losses associated with initial cooldown of the tank, and lower level of system activity (*viz.*, limited pump operation for safety checks and other commissioning activities). Pump validation activities were performed in July, resulting in higher LH_2 utilization and lower levels of refueling. System validation testing in August through October continued this trend; activities were limited by the LH_2 delivery schedule.

In August, a series of tests were performed to evaluate the ability of the pump to operate under different tank conditions, including lower tank levels. During these tests, the pump was able to operate over the entire range of tank fill levels. Operation was also possible while the tank was being refilled from a LH_2 tanker. These tests culminated in a series of fill cycles where the tank was run continuously until “dry”. This corresponded to a final tank liquid level of 4%, mainly limited by the physical distance between the pump intake port and the bottom of the tank. Because the cryotank is essentially a horizontal cylinder, 4% liquid level corresponds to 1.3% full capacity. The immediate delivery after this test required measures to refill a tank with no liquid – a “warm fill” in industry lingo, though the tank was not actually warm or allowed to warm internally; higher tank levels were maintained in the aftermath to provide margin for the additional cooldown of the cryotank.

Boil-off

Static and dynamic boil-off are the main sources of H_2 loss in existing LH_2-based refueling stations. Static boil-off occurs

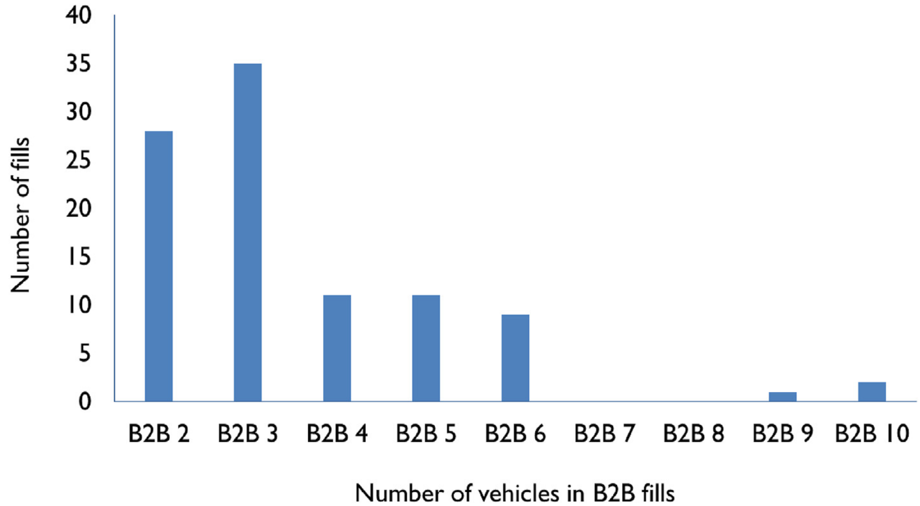

Fig. 8 – Back-to-back fills for conducted for 2 to 10 vehicles in a row.

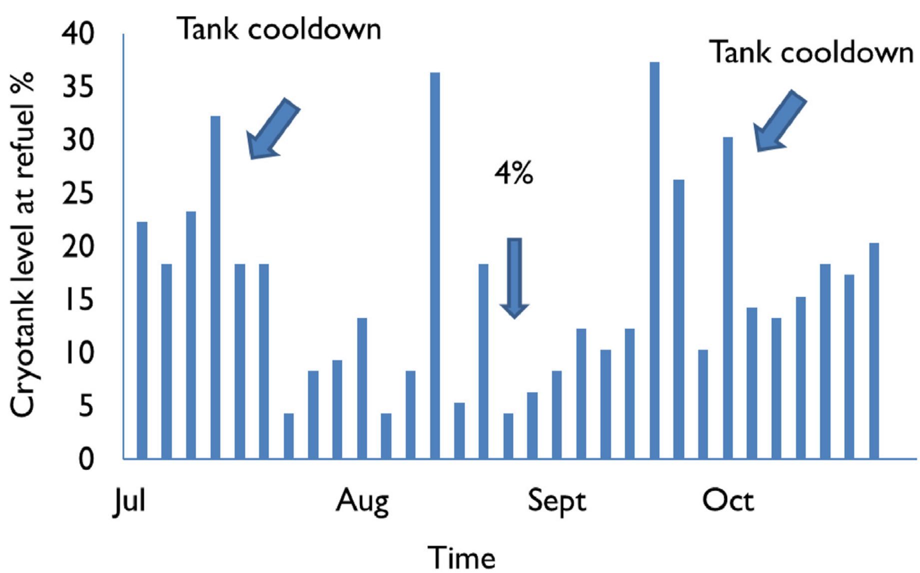

Fig. 9 – Cryotank liquid level at delivery during the test campaign. Higher levels were required between deliveries at the beginning of testing to provide margin during tank cooldown. A test to determine the minimum operating level for the pump (and maximum tank utilization) also required higher tank levels immediately afterwards to accommodate tank cooldown.

through heat leak into the cryotank via conduction and radiation. In our cryotank design, the socket and the pump offer additional paths for static heat leak that is not present in cryotanks with external liquid pumps. Dynamic boil-off occurs due to heat leak into the tank during operation of the pump. It includes mechanical energy used to overcome friction instead of discharge pressure.

Boil-off was computed from the headspace pressure, the liquid level, and temperature. Static boil-off was estimated from data during idle periods, and dynamic boil-off was estimated during active pump operation. Based on direct measurement during idle periods, the normal evaporation rate (NER) of the combined cyrotank, socket, and pump system was approximately 7.5% (6%–9%) per day or approximately 150 W for the entire assembly. Using heat transfer analysis for the physical design, we estimate the contributions from the cryotank, socket, and pump in our prototype system to be 103, 19, and 28W, respectively. Based on measurements during pump operation, we found the contribution of dynamic boil-off to be negligible relative to the static boil-off. More specifically, we inferred the magnitude of the dynamic heat leak from the headspace pressure in the cryotank during pumping. Under static conditions, heat leak into the tank results in boil-off which causes a steady rise in the headspace pressure. During pump operation, enthalpy is removed as liquid H_2 is extracted from the tank. If the magnitude of this enthalpy loss balances the sum of the static heat leak and dynamic heat leak from pump operation, the headspace pressure in the tank will remain steady. During testing, we observed a decrease in headspace pressure allowing us to estimate an upper bound on dynamic heat leak. This upper bound was significantly lower than our estimate of static heat leak.

In a full-scale system, we expect the boil-off to be significantly reduced from the 7.5% per day level observed in our prototype system. Improvements are expected from three changes. First, the 1500-gal cryotank in the prototype system is designed for road transport according to Federal Code MC338 which required extra structural support to accommodate the dynamic loads expected during transport. The system was also designed to hold liquid nitrogen (LN_2), which was used for initial system validation. LN_2 is more than 11 times heavier than LH_2 requiring further reinforcement of the tank supports. Together, the tank supports account for close to half of the total heat leak, and design of dedicated, stationary LH_2 cryotanks for future systems will result in significant reductions in heat leak and boiloff. Second, the stations based on the NICE LHRS concept will use cryotanks with larger capacity. An industry standard (18,000 gal) tank would have a lower surface to volume ratio, further reducing the relative contributions to boil-off. Finally, the larger cryotank would require a longer socket and pump. The increased lengths will further reduce heat conduction. In addition, the heat leak contributions from the socket and pump will decrease relative to cryotank in larger systems leading to additional improvement. The exact level of boil-off in full-scale systems will depend on the specific design, but is expected to approach the industry standard of 0.5–1.0% per day in large cryotanks [13].

Implications for future hydrogen refueling station design

Our results suggest that the submerged LH_2 pump concept has the potential to enable four desirable features in future HRS design:

- Low energy use.
  - Pumping LH_2 is up to 10 times more efficient than compressing gaseous H_2. Under representative fueling conditions, our LH_2 pump requires 0.3 kWh/kg to complete a 35 MPa fill cycle of up to 30 kg following at J2601-2-compliant profile. This compares favorably to the energy requirements for existing liquid pumps (1.0 kWh/kg) and gas compressors (3 kWh/kg) to achieve similar performance [2].
  - Our system concept can deliver precooled H_2 at temperatures as low as −40 °C, without additional energy requirements for refrigeration. The pump energy consumption accounts for more than 90% of the total energy use of our system during filling.
- Fast start-up and unlimited back-to-back filling without a cascade system.
  - The use of a submerged pump eliminates the need to cool the pump at start-up. During testing, our system was able to consistently begin dispensing H_2 nearly instantly, without cooldown losses. Refrigeration is not needed in our system to adjust temperature of the dispensed fuel. This eliminates both the capital and operating costs of refrigeration units and possible delays associated with cool-down of refrigeration blocks.
  - Our system allows direct filling of vehicles. The ability of the pump to continuously deliver compressed gas at the desired temperature, pressure and flow rate eliminates the throughput limitation imposed by the need to recharge ground storage and allows unlimited back-to-back filling.
- Low pump and system boil-off.
  - Pump operation does not contribute to boil-off as shown by the pressure decrease in the tank during operation – enthalpy outflow overwhelms any frictional loss from the pump.
  - Additional boil-off is avoided due to the submerged nature of the pump.
  - Boiloff from the cryotank in the prototype system was estimated to be 7.5% NER. This was found to be driven by the small size of the tank; boil-off in full-scale systems should approach the industry standard of 0.5–1.0% for large cryotanks. Heat leak through the pump socket was estimated to be less than 19 W for full-scale station designs.
- Small station footprint.
  - The liquid pump demonstrated in this study is capable of delivering more than 230 kg/h of H_2. When combined with the integrated dispensing loop, the system has the potential to eliminate gas compressors, cascade tubes, and refrigeration resulting in reduced station footprint, in addition to potential savings in capital and operating costs. The economic potential and performance of the system will be discussed separately in a future report.
  - Our LH_2 pump system was able to operate at low tank liquid levels down to 4%. This means the pump has the potential to utilize virtually all of the LH_2 in the cryotank. Moreover, the socket design makes our system compatible with buried storage cryotank concepts. Underground LH_2 storage requires further development, including the adoption of appropriate codes and standards, but has the potential to further reduce station footprint.

Together, these innovative features are expected to enable a more compact HRS with daily dispensing capacity of at least 1000 kg/d. Large capacity HRS is the key to profitability and scale. For reference, the liquid hydrogen pump under development in this project is capable of 285 kg/h. Fewer machines and no cascade tubes directly lead to capital and operating cost reduction. Furthermore, elimination of a refrigeration system is another source of capital and operating cost reduction. Finally, storing LH_2 in an underground tank allows a large amount of on-site storage with a small footprint, further reducing the cost of land. Detailed station designs are being developed, and the economics of operating such stations will be described in the future.

Conclusions

We successfully tested a hydrogen refueling station concept capable of delivering precooled, compressed gaseous hydrogen for HDV refueling applications. Over a 6-month period, we pumped a total of 9000 kg of LH_2 and performed 1350 simulated filling cycles. The submerged LH_2 pump can deliver pressurized LH_2 at 45 MPa and flow rates up to 285 kg/h; the full system is capable of multiple J2601-2 compliant precooled fills demonstrating its potential for transit bus refueling applications. The system concept offers a number of desirable features for cost-competitive refueling stations and efforts are underway to fully commercialize the technology.

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Declaration of competing interest

The authors declare the following financial interests/personal relationships which may be considered as potential competing interests: NICE America is working to commercialize the pump technology described in the paper.

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