Towards High-Efficiency Modular Hydrogen Liquefaction: System Design and Experimental Validation

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Abstract. Conventional hydrogen liquefaction capacity is typically increased by scaling up a single process train, resulting in large, capital-intensive plants with limited flexibility. This study investigates an alternative number-up strategy based on high-efficiency modular Modified Collins Cycle units using electronically controlled floating-piston expanders. Combined production capacity could therefore be increased by replicating standardized modules rather than enlarging individual machinery. A transient two-stage model predicted a specific energy consumption of 10.1 $kWh.kg_{H_2}^{-1}$ using helium as the working fluid, decreasing to 7.5 of $kWh.kg_{H_2}^{-1}$ with a 20:80 He–H_2 mixture. An FPGA-controlled room-temperature prototype was developed and evaluated through 16 experimental tests using air, nitrogen, argon, and helium. Stable operation was demonstrated with all gases, including a continuous 12.2 h test comprising 25,170 cycles without stalling or system failure. These results establish the feasibility of the expander architecture and support its development toward modular cryogenic liquefaction systems that can be numbered up for larger capacities or deployed in distributed, portable and zero-boil-off applications.

1. Introduction

Liquid hydrogen (LH_2) has historically served primarily as a rocket propellant. Hydrogen's high gravimetric energy density, combined with the greater volumetric density obtained through liquefaction, is now driving interest in aviation, maritime transport, heavy-duty mobility, long-distance hydrogen transport, and energy export and storage, where low-emissions hydrogen could support decarbonization [1]. Many emerging applications, however, require flexible production from sub-tonne-per-day to several tonnes per day rather than the fixed output of a large, centralized, stick-built plant.

This capacity range motivates a shift from direct scale-up towards numbering-up, where multiple identical, high-efficiency modules are deployed in parallel rather than enlarging a single bespoke liquefaction train [2]. Standardized highly efficient identical modules can be factory-fabricated, installed progressively, operated near their design point, and isolated for maintenance, enabling staged investment, redundancy, and incremental expansion. Comparable strategies are evident in SpaceX's 33-engine Super Heavy booster [3], multi-module small modular reactor plants [4], and IBM Quantum System Two, which integrates expandable cryogenic infrastructure with modular control electronics [5].

Realizing this strategy requires compact liquefaction modules without sacrificing thermodynamic performance. Stirling, Gifford–McMahon (GM) and pulse-tube cryocoolers are compact and reliable; however, current pulse-tube packages operating near 20 K require approximately 460–850 W of compressor input per watt of cooling, corresponding to only 1.6–3.0% of Carnot efficiency [6]. A high-capacity GM cryocooler requires approximately 127 W/W and achieves around 11% of Carnot [7]. By comparison, a tested reverse-Brayton cryocooler requires approximately 80–85 W/W and reaches about 16% of Carnot, although turbomachinery efficiency generally deteriorates when scaled to smaller capacities [8]. The Modified Collins Cycle (MCC) employs electronically controlled floating-piston expanders within a recuperative, continuous (DC)-flow architecture and is projected to achieve 20–25% of Carnot, equivalent to approximately 56–70 W/W [8,9]. If realized, this would reduce input power by approximately 45–56% relative to GM and 85–93% relative to current pulse-tube systems, while retaining compactness, modular construction and automated control.

This study presents the design and development of an MCC-based modular liquefaction system. A predictive transient model is developed to define operating and component-design parameters, while a room-temperature floating-piston-expander prototype is constructed to validate piston dynamics, valve timing, and the proposed control architecture.

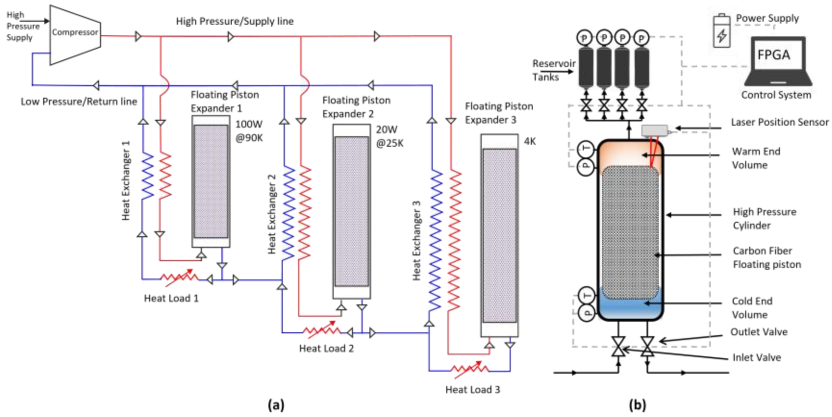

Figure 1. a) Multi-stage Modified Collins cycle architecture [13]. b) Schematic of the floating-piston expander.

2. System architecture: the Modified Collins Cycle

The Modified Collins Cycle is a continuous-flow (DC) recuperative cycle designed for high efficiency and multi-stage cooling. As shown in Fig. 1(a), high-pressure gas from the compressor flows through staged recuperative heat exchangers, where it is pre-cooled before entering the floating-piston expanders. Each expander drops the temperature through a near-isentropic expansion, and the cold, low-pressure return stream is then used to precool the incoming gas and absorb a heat load. The process repeats over several stages depending on the target temperature, e.g. 100 K, 20 K, and 4 K.

Fig. 1(b) shows details of a single floating-piston expander (FPE), the key component enabling efficient expansion. A lightweight piston moves freely within the cylinder without a mechanical linkage, driven solely by the pressure differential across its two ends. Expansion is controlled by timed inlet and outlet solenoid valves at the cold end, together with warm-end valves connected to pressurised gas reservoirs. These reservoirs act as warm-end heat-rejection volumes, receiving the energy transferred from the cold end during each cycle and dissipating it to the surroundings at approximately room temperature, while also moderating pressure fluctuations. Real-time piston-position feedback from a laser sensor, along with pressure measurements, feeds an FPGA-based control system that actuates the valves and sets the piston stroke precisely, and thus the expansion process. A high-pressure ratio (15:1) and low operating frequency (1–3 Hz) enable a near-isentropic, high-efficiency expansion (20–25% of Carnot). The MCC is well suited to a modular number-up architecture because each stage integrates a lightweight composite cylinder and free-piston expander with a finned-tube heat exchanger wrapped directly around the cylinder, forming a compact, process-intensified refrigeration module. Its operation is governed by advanced LabVIEW FPGA control of valve timing rather than by fixed crank speed or turbomachinery scale, helping to mitigate the efficiency penalties associated with miniaturization. Identical modules can therefore be manufactured and operated at their optimal design point, with the required plant capacity achieved by connecting multiple units in parallel rather than enlarging a single process train.

3. Theoretical performance model

The thermodynamic performance of a floating-piston expander is governed by a complex interplay between geometric parameters (bore diameter, stroke length, and clearance gap), valve timing, piston frequency, flow rate, pressure ratios, and working-fluid thermophysical properties. Therefore, a transient model was developed in Python to guide the design of the modular MCC and to establish performance benchmarks for half a metric tonne per day (0.5 TPD) liquefaction plant. The model solves the coupled mass and energy conservation equations together with the piston dynamics at each time step. It builds on the ideal FPE dynamics formulated by Chaudhry [10], while the dominant loss mechanisms, including shuttle heat transfer, appendix-gap conduction, axial conduction, and enthalpy transfer, were incorporated following Segado's model [9].

Fig. 2(a) shows a graphical user interface of the custom software developed to predict the system performance of a given design and enable interactive parameter tuning and real-time visualisation of system behaviour. Using the optimized geometry and pure helium as the working fluid, the transient simulation (Fig. 2(b)) shows the two-stage cooling from ambient temperature to approximately 90 K (first stage) and 22 K (second stage) within 5.5 hours.

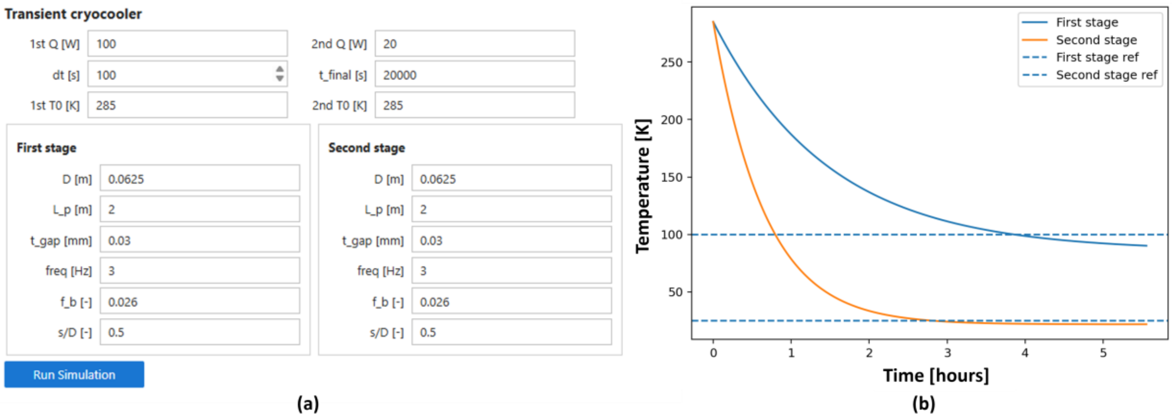

Figure 2.a) Software interface developed for the transient MCC model. b) modelled two-stage cooldown of a single module to 90 K and 22 K for 0.5 TPD liquefaction capacity.

The corresponding performance metrics are summarized in Table 1. Specifically, the model predicts a specific energy consumption (SEC) of 10.1 $kWh.kg_{H_2}^{-1}$ at a second-law efficiency of 31%, already comparable to state-of-the-art industrial liquefiers [1], with 129 modules required to meet the 0.5 TPD duty (module specifications are given in Fig. 2(a)). Further improvement was achieved with a 20:80 He–H2 mixture, which reduced the SEC to 7.5 $kWh.kg_{H_2}^{-1}$ at a second-law efficiency of 68% through improved temperature-glide matching in the recuperators, while more than halving the number of modules required for the 0.5 TPD duty to approximately 58.

Table 1. Modelled performance of a single two-stage module operating with helium for a concept 0.5 TPD liquefaction capacity.

T_{1st} (K)	T_{2nd} (K)	$\dot{Q}_{cool,1st}$ (W)	$\dot{Q}_{cool,2nd}$ (W)	$\dot{W}_{comp}$ (W)	$\eta_{isen,1st}$ (%)	$\eta_{isen,2nd}$ (%)	η_{II} (%)	SEC (kWh $kg_{H_2}^{-1}$)	$N_{modules}$
90	22	104	20	1630	91	94	31	10	129

4. Experimental validation: Room-temperature prototype

Achieving reliable and continuous control of the MCC floating-piston expander proved challenging in previous systems [11,12]. Therefore, a room-temperature demonstration prototype was designed and built to tune and validate the previously developed design parameters and control algorithms before progressing to cryogenic operation. Fig. 3 shows the assembled expander prototype. The expander comprises a carbon-fibre floating-piston, selected for its high strength-to-weight ratio and low coefficient of thermal expansion, moving within a transparent polycarbonate cylinder that allows direct visual observation of the piston motion. The system also includes four pressurized warm-end gas reservoirs, solenoid valves at both the cold and warm ends, a laser triangulation sensor for real-time piston-position measurement, and pressure and temperature sensors. Control is implemented using an FPGA-based platform (NI cRIO-9045) interfaced with LabVIEW FPGA, enabling precise valve timing based on real-time feedback from piston-position and pressure sensors.

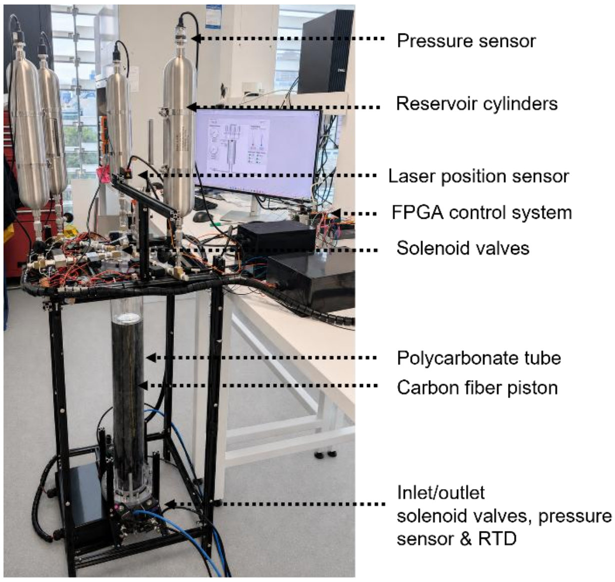

Figure 3. Floating-piston expander room temperature prototype.

The prototype was evaluated through 16 experimental tests covering a total of 51.6 h of operation using air, nitrogen, argon, and helium. The experiments included two piston lengths (500 and 600 mm, with strokes of 300 and 200 mm, respectively), two- and four-reservoir configurations, piston-cylinder clearance gaps of 0.2 and 0.3 mm, and solenoid valves with Cv values of 0.1 and 0.3, across supply pressures of up to 7 barg. Despite the substantial differences in the thermophysical properties of the four gases, the feedback controller established repeatable and continuous piston motion and maintained the reservoir pressure distribution required for continuous cyclic operation. Under the tested operating conditions, the prototype achieved operating frequencies of 0.30–0.93 Hz and pressure ratios of 1.13–2.86. Continuous operation was demonstrated in a 12.2 h endurance test using air with the 600 mm piston, four reservoirs and valves with 0.3 Cv during which the expander completed 25,170 consecutive cycles at 0.60 Hz with no stall or failure events. Reducing the piston length from 600 to 500 mm nearly tripled the stroke and swept volume, from 27 to 80 mm and from 201 to 590 cm^3, respectively, although both configurations remained well below full stroke.

Fig. 4 shows the four reservoir mean pressures over the full run. They remained at distinct and stable levels throughout the test, demonstrating that the implemented algorithm and feedback controller were capable of maintaining a consistent pressure distribution and rejecting any potential disturbances required to sustain cyclic piston motion. It's worth noting that the room-temperature prototype was operated in an open-loop, uninsulated configuration without heat exchangers or a vacuum enclosure. It was designed to validate the mechanical design parameters and, more importantly, the FPGA-based valve-control algorithm.

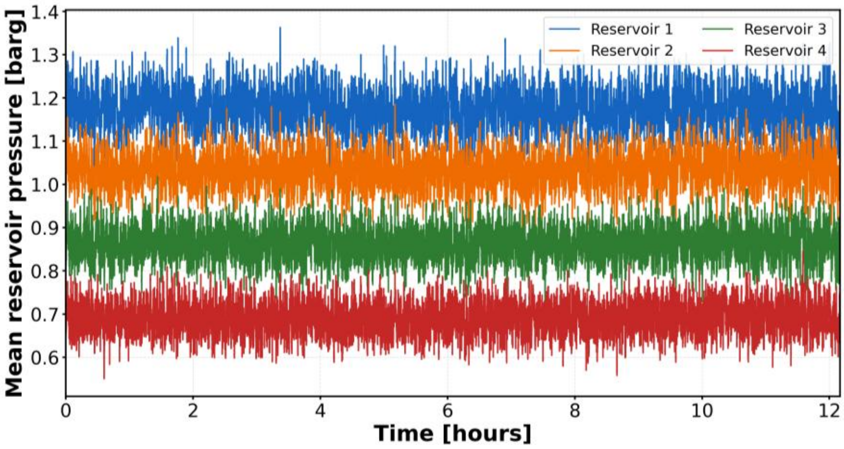

Figure 4. Reservoir mean pressures over the 12.2 h test using air.

The measured P–V diagram for the best-performing test is compared with the corresponding ideal cycle and shown in Fig. 5. Although the experimental loops followed the expected expansion and recompression sequence and enclosed a positive work area, their smaller areas relative to the ideal cycles indicate incomplete cycles and losses within the prototype. The helium test produced the highest operating frequency of 0.93 Hz and a pressure ratio of 2.25.

Measurable cooling was observed in all 16 tests, with an average peak cold-to-warm temperature difference of 1.3 °C. Helium produced the largest response, reaching 4.1 °C, while the 12.2 h air test sustained a steady temperature difference of 2.2 °C and reached a peak of 3.0 °C. These modest temperature drops, rather than the tens of degrees predicted by the ideal model, are primarily a consequence of the room-temperature test configuration, in which no insulation or heat exchangers are present and the system operates in an open loop. Several additional factors, including partial piston stroke and low flow rate, also limit cooling and overall performance. The restricted intake flow is the most significant: in the comparable 600 mm four-reservoir configuration, increasing the valve C_v from 0.1 to 0.3 raised the measured pressure ratio from 1.1–1.6 to 1.9–2.3 and the average stroke from 17–28% to 24–35%. Even so, these values remain well below the available inlet pressure of 7 barg. Gas blow-by through the 0.2–0.3 mm piston-cylinder clearance gap further degraded performance by permitting a slow net mass transfer between the reservoirs.

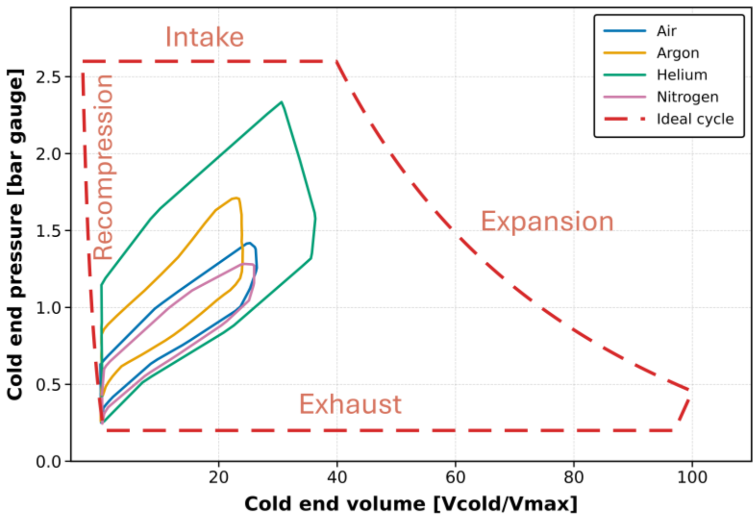

Figure 5. Experimental cold-end P–V diagrams for air, argon, helium and nitrogen compared with the ideal MCC cycle.

5. Conclusions

A modular hydrogen liquefier based on the Modified Collins Cycle was modelled, and the control and operation of its floating-piston expander was experimentally demonstrated and validated at room temperature with the absence of an insulation/vacuum chamber and a heat exchanger. The model predicted a specific energy consumption of 10.1 kWh.$\mathbf{kg_{H_2}^{-1}}$ using pure helium as the working fluid, decreasing to 7.5 kWh.$\mathbf{kg_{H_2}^{-1}}$ with a 20:80 He–H_2 mixture, while the prototype demonstrated stable, continuous operation with air, nitrogen, argon, and helium. Restricted intake flow and blow-by through the piston–cylinder clearance gap were identified as the factors not allowing the prototype reaching its full expansion cycles and further cooling. The next-generation prototype will address these limitations by reducing the clearance gap, incorporating higher-Cv valves, and systematically investigating the effect of piston length. The system will then be transitioned to a fully cryogenic configuration, incorporating vacuum insulation, a recuperative heat exchanger, and a closed-loop helium compression circuit in preparation for low-temperature testing.

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