PROCEEDINGS OF ECOS 2026 - THE 39TH INTERNATIONAL CONFERENCE ON
EFFICIENCY, COST, OPTIMIZATION, SIMULATION AND ENVIRONMENTAL IMPACT OF ENERGY SYSTEMS
28 JUNE – 03 JULY 2026, CONSTANTA, ROMANIA

Modelling and Simulation of a Hydrogen Refueling Station for Heavy-duty Vehicles

José Luís Grilo[a], Ana Cristina Ferreira[b,c], Carla Martins[d], Senhorinha Teixeira[e] and José Carlos Teixeira[f]

[a]CMEMS, Department of Mechanical Engineering, University of Minho, 4800-058 Guimarães, Portugal; id8983@alunos.uminho.pt
[b]COMEGI - Centro de Investigação em Organizações, Mercados e Gestão Industrial, Universidade Lusíada, 4760-108 VNF, Portugal
[c]MEtRICs, Department of Mechanical Engineering, University of Minho, 4800-058 Guimarães, Portugal; acferreira@dps.uminho.pt
[d]IPC - Institute for Polymers and Composites, University of Minho, 4804-533 Guimarães, Portugal; cmartins@dep.uminho.pt
[e] ALGORITMI Centre, Department of Production Systems, University of Minho 4800-058 Guimarães, Portugal; st@dps.uminho.pt, CA
[f]MEtRICs, Department of Mechanical Engineering, University of Minho, 4800-058 Guimarães, Portugal; jt@dem.uminho.pt

Abstract:

Hydrogen refueling stations (HRS) for heavy-duty vehicles face rigorous requirements in terms of refueling time, thermal safety and energy efficiency. Modelling and simulation refueling transient thermodynamic behavior allows supporting station design and component sizing. This study develops and evaluates a progressively refined numerical model to simulate the refueling of heavy-duty vehicles, with particular emphasis on transient thermal effects and heat-exchanger integration. Therefore, the main objective was to quantify the impact of non-ideal hydrogen behavior, active precooling on pressure, temperature and heat-removal requirements during filling.

A zero-dimensional, time-dependent control-volume model was implemented in MATLAB software, solving mass and energy conservation equations using real-gas thermophysical properties obtained from hydrogen property databases. Several modelling levels were considered during the research, culminating in a transient non-ideal heat-exchange model in which hydrogen is precooled to -40°C, before inflowing to the vehicle tank at the designated pressure ramp rate, while both storage and vehicle tanks are treated as adiabatic.

Simulation results demonstrated that neglecting thermal effects leads to unrealistic temperature predictions under heavy-duty refueling conditions. Without active cooling, tank temperatures rise due to compression heating and Joule-Thomson effects. The inclusion of a heat exchanger can mitigate such effect, by maintaining tank temperatures below the safety threshold of 85°C, according to SAE J2601 guidelines. Nonetheless, even with precooling, tank reaches temperatures of approximately 72°C, highlighting the persistent thermal challenge of high-pressure hydrogen dispensing. Also, the simulations revealed that the skid storage temperatures decrease during depletion, reducing the cooling required for subsequent refueling processes.

The cumulative heat removal demand depends on both ambient temperature and the thermodynamic state of the storage cascade system. In conclusion, the results underlined the need for integrated thermodynamic and control-oriented modelling when designing heavy-duty HRS. The proposed model provides a cost-effective tool for analyzing and supporting future optimization developments for hydrogen refueling infrastructure.

Keywords:

Hydrogen refueling stations; Transient thermal modelling; Energy transition technologies.

1. Introduction

The decarbonization of the transportation sector is an important vector of the global energy transition [1]. Heavy-Duty Vehicles (HDVs), including buses and long-haul trucks, contribute significantly to greenhouse gas emissions due to their high energy demand and widespread operational cycles [2]. Hydrogen fuel cell technology has emerged as a promising solution for long-range and high-utilization transport applications, offering fast refueling, high energy density and zero tailpipe emissions [3, 4].

Hydrogen fuel cell vehicles offer extended driving ranges and faster refueling times compared to alternative electric battery cars, being especially relevant for high-utilization vehicles with predictable routes [5].

By the end of 2024, over 1,160 hydrogen refueling stations (HRS) were operating worldwide, with 45 countries having infrastructure in operation or under construction. In Europe, there were 294 hydrogen refueling stations, 113 of which were in Germany [6].

Despite this milestone, the lack of hydrogen refueling infrastructure remains a barrier to fuel cell technology adoption. However, the deployment of hydrogen-powered HDVs requires reliable and thermally safe hydrogen refueling stations capable of delivering large quantities of hydrogen within short time frames. The design and operation of hydrogen refueling infrastructure remain technically and economically challenging.

Hydrogen dispensing at high pressures requires significant energy for gas compression and pre-cooling, while ensuring strict compliance with safety limits related to temperature and pressure inside the vehicle tank. In addition, the management of cascade storage systems and the variability of refueling demand can significantly affect station performance and operating costs [7,8]. Most existing hydrogen refueling stations operate with gaseous hydrogen storage at pressures of 350 bar or 700 bar, typically using cascade storage configurations combined with multi-stage compression and pre-cooling systems to enable fast and standardized refueling procedures [9]. While cascade storage can reduce compressor energy consumption, the number of storage banks, their volumes, and the pressure switching strategy directly influence energy use, refueling time, and overall system efficiency [10].

The SAE J2601 standard establishes the fueling protocol requirements for light-duty hydrogen vehicles, defining the pressure ramp rates, temperature limits, and density targets that ensure safe and complete filling of the compressed hydrogen storage system. The protocol specifies that the internal gas temperature must remain within the design range of -40 °C $\leq T_{gas} \leq$ 85 °C throughout the refueling process. For heavy-duty vehicles, the SAE J2601/2 standard extends these principles [11]. To prevent the gas temperature from exceeding the 85 °C limit and to maximize the state of charge, hydrogen must be precooled before entering the vehicle tank. In the specific case of heavy-duty refueling, the precooling challenge is more demandable because the higher mass of hydrogen needed to be dispensed per fill (30–100 kg).

For infrastructure planning and system design, it is therefore essential to link the nominal capacity of a refueling station with its operational strategy. Oversizing storage or compression capacity can lead to unnecessary capital and operational costs, whereas undersizing these components may prevent the station from meeting peak demand. In this context, the modelling and simulation of hydrogen refueling stations have become essential tools for supporting infrastructure design and operational planning. Zero-dimensional models solve mass and energy conservation equations, whereas the tank is treated as a uniform control volume, offering computational speed suitable for system-level simulation and protocol development. In [12], a thermodynamic model was applied to heavy-duty bus refueling at 350 bar, validated against real operational data. The model predicted pressure, temperature and mass flow over time, allowing to find out that refueling from half to full capacity at an Average Pressure Ramp Rate (APRR) of 0.03 MPa/s required approximately 10 minutes. Also, the study concluded that the initial vehicle tank pressure had a more considerable effect on refueling process than ambient temperature. Another work has further developed MATLAB-Simulink-based dynamic models specifically for heavy-duty applications. In [13], a customizable model was developed for systems with up to 100 kg of capacity, achieving the fastest refueling (~10 min) with low ambient temperatures (20 °C), and dispenser temperatures of -40 °C, while maximizing the APRR up to 9 MPa per minute, without exceeding SAE J2601 limits. Other models, in addition to the thermodynamic filling modelling, some studies integrate a techno-economic analysis [5].

Accurate simulation models allow the evaluation of thermodynamic behavior during the refueling process, including the prediction of pressure and temperature evolution in both vehicle tanks and station storage systems. Such models also enable the assessment of trade-offs between energy consumption, operational cost, and refueling performance. Furthermore, modelling frameworks provide a basis for analyzing different operating scenarios, from light-duty vehicles to heavy-duty buses, and for developing control strategies that can adapt to variable operating conditions.

It is clear on the literature that, while research on modeling hydrogen refueling for HDVs has increased, there are several research opportunities. Larger tanks, greater hydrogen demand, and newer refueling protocols all introduce added complexity. In particular, it is important to understand how factors like non-ideal hydrogen behavior and active precooling influence pressure, temperature, and heat removal during heavy-duty refueling. Addressing this gap is what drives the present study. In this context, the present work develops a numerical methodology for modelling and simulating the operation of a hydrogen refueling station designed for HDVs.

The proposed model integrates the main station subsystems, including hydrogen supply, cascade storage, pre-cooling and dispensing processes. Using realistic technical parameters and standardized fueling protocols, the model provides a flexible framework for analyzing the thermodynamic behavior of the refueling process and evaluating the performance of hydrogen refueling infrastructure during heavy-duty vehicle operations.

2. System description

The objective of this research is to develop a mathematical model that characterizes the thermodynamic behavior of a hydrogen refueling station for heavy vehicles. A case study was defined considering the refueling requirements of heavy vehicles at the rated H35 (35 bar) and the required operation parameters as well as a skid with fixed volume for each study case and model. The whole refueling is simulated according to SAE TIR J2601/2 which applies to gaseous hydrogen powered heavy-duty vehicles. The model is composed of the following components: heavy vehicle tank of a fuel cell electric vehicle, the station storage tanks and skid levels, cooling system between the dispenser and the storage and the group of valves connecting each pair of components. The station storage is split into 3 tanks (skid level) assembled in a cascade system, each with a different pressure capacity. During refueling, each level will supply its hydrogen until the vehicle's pressure surpasses that of the current skid level pressure.

The hydrogen refueling station configuration concerning the skid level volume is selected from a previous optimization study [6]. The skid is composed of three containers, functioning as a hydrogen storage, each at different pressure levels. The skid storage banks have an initial pressure, pressure limits, and initial temperatures, which values are presented in **Erro! A origem da referência não foi encontrada.**. The ambient temperature is 20 ºC and the hydrogen precooling target temperature to −40 ºC. The APRR to fuel the buses was assumed to be 1.95 MPa/sec, in order to comply with a 3-minute refueling time for heavy vehicles. The time between the end of each filling and the start of next is assumed to be instant ($\Delta t = 0$). Each bus has an initial pressure of 5 bar and will be fueled to the specified 350 bar.

Table 1. *Vehicle and skid parameters considered for the system configuration*

Parameter	Vehicle Tank	Skid HP	Skid MP	Skid LP
Volume (m^3)	1.56	11.1	9.4	12.4
Temperature (ºC)	25	25	25	25
Initial Pressure (bar)	5	500	350	207
Maximum Pressure (bar)	350	500	350	207
Minimum Pressure (bar)	N/A	350	207	50

HP – High Pressure, MP – Medium Pressure, LP – Low Pressure

3. Numerical modelling

The hydrogen refueling station is modelled as a zero-dimensional (0D) time-dependent control-volume system. The configuration consists of a three-level cascade storage (HP, MP, LP), a vehicle tank, a heat exchanger for pre-cooling and pressure-switch control logic, as presented in Figure 1. State variables are assumed to be uniformly distributed in each container.

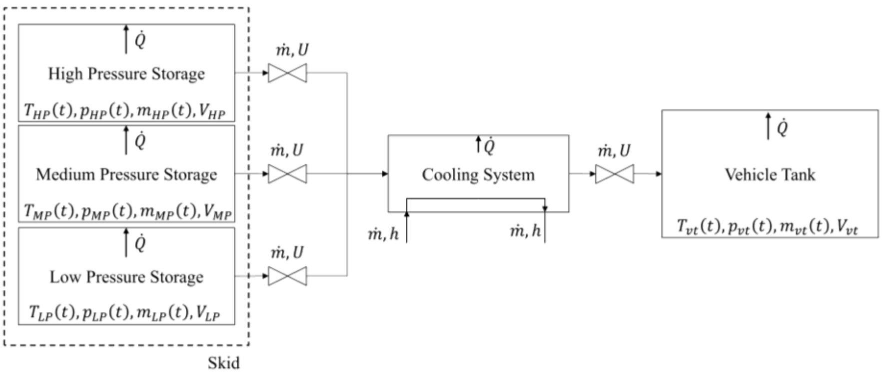

Figure 1. *Schematic of mass and energy relationship of the hydrogen refueling station.*

3.1. Thermodynamic model

Thermodynamic simulation of hydrogen behavior is modelled through the conservation of mass, in Eq. (1) and Eq. (2). While conservation of energy is modelled through the first law of thermodynamics for open systems in Eq. (3). Due to their minor impact on the results, the kinetic and potential energy of mass flows are neglected.

$$\frac{dm}{dt} = \sum \dot{m}_{in} - \sum \dot{m}_{out} \tag{1}$$

$$\frac{dm}{dt} = V \frac{d\rho}{dt} \tag{2}$$

$$\frac{dU}{dt} = Q - W + \sum \dot{m}_{\mathrm{in}} \mathrm{h}_{\mathrm{in}} - \sum \dot{m}_{\mathrm{out}} \mathrm{h}_{\mathrm{out}} \tag{3}$$

where, m is mass, t is time, $\dot{m}_{in}$ is the inlet mass flow, $\dot{m}_{out}$ is the outlet mass flow, $\frac{dU}{dt}$ is the change in internal energy, h_{in} is the inlet enthalpy, h_{out} is the outlet enthalpy, Q heat exchange of the gas with the exterior, and W is the work done by the gas.

Due to their minimal influence on the refueling process, both pressure losses and pipe heat transfer were excluded from the analysis [14]. Considering the supercritical state imposed by the operation conditions, state and thermodynamic properties are determined through by the NIST REFPROP database [15].

The dynamic hydrogen model follows a similar general main step: mass iteration, internal energy update and state variable from thermodynamic database. At each timestep, the mass of each tank or control volume is updated explicitly based on the calculated mass flow during the current step for the next iteration. Using this updated mass, the energy balance equation is then calculated to obtain the new internal energy derivative of each component. Thus, the iteration of internal energy is determined explicitly using the previously determined derivative and time step. Once both the mass and internal energy have been updated, the remaining thermodynamic state variables are reconstructed by querying the hydrogen property tables. Density together with internal energy provides the necessary inputs for retrieving pressure, temperature, enthalpy, and other relevant properties.

During the first iteration step, the mass is steadily increased until the calculated pressure for that time step corresponds to the established by APRR value. Therefore, once mass transfer (Δm) is iterated, the system will update the internal energy of the tank through Eq. (3), which for the tank is reduced to Eq. (4). By considering the inflow enthalpy as the enthalpy outflowing the cooler, the equation can be rearranged for the iterated internal energy of the tank as expressed in Eq. (5).

$$\frac{dU}{dt} = \sum \dot{m}_{\mathrm{in}} \mathrm{h}_{\mathrm{in}} = m_{tank}^{i} u_{tank}^{i} - m_{tank}^{i-1} u_{tank}^{i-1}, \qquad \dot{m}_{in} = \Delta m \tag{4}$$

$$u_{tank}^{i} = \frac{m_{tank}^{i-1} u_{tank}^{i-1} + \dot{m}_{in} h_{in}}{m_{tank}^{i}}, \quad h_{in} = h_{cooler} \tag{5}$$

where m_{tank} is the H_2 mass in the tank, u_{tank} is the H2 specific internal energy in the tank, $\dot{m}_{in}$ mass inflow into the tank, h_{in} the enthalpy inflowing in the tank, h_{cooler} the enthalpy outflowing in the cooler, ^{i} property in the current timestep, and $^{i-1}$ property in the previous timestep.

Real-gas properties are therefore obtained from accurate thermodynamic databases (CoolProp). Therefore, initial thermodynamic states are retrieved from the database using the initial pressure and temperature conditions (ex: $h = f(P,T)$). During the iteration routines, every other thermodynamic variable is retrieved from the updated internal energy and density ($P, T, h = f(u, \rho)$). Once the mass flow is validated according to the ARRP, state variables corresponding to the skid levels can be updated as well as the necessary heat removal from the cooler. Similar to before, these can be expressed by Eq. (6) and Eq. (7):

$$\frac{dU}{dt} = -\sum \dot{m}_{\text{out}}\, \text{h}_{\text{out}} \tag{6}$$

$$u_{skid}^{i} = \frac{m_{skid}^{i-1} u_{skid}^{i-1} - \dot{m}_{out} h_{out}}{m_{skid}^{i}}, \quad h_{out} = h_{skid} \tag{7}$$

where m_{skid} is the H2 mass in the skid, u_{skid} is the H2 specific internal energy in the tank, $\dot{m}_{out}$ mass outflow into the tank, h_{out} the enthalpy outflowing in the skid.

All thermal effects arise from gas compression, expansion, and mixing processes. The cooling system is modelled as an isobaric process, where hydrogen is cooled at constant pressure to reach the target outlet temperature before entering the vehicle tank. Considering a steady-flow conditions, heat removed from the hydrogen gas is then calculated through Eq. (8). It is assumed that the connections between the skid and heat exchanger are isenthalpic, thus inflow enthalpy can be considered the same as the one on the skid. While outflowing enthalpy is derived from the thermodynamic database, assuming that the heat exchanger is isobaric and that it is capable of removing heat until hydrogen reaches −40 °C. The same enthalpy is used as the inflow enthalpy, meaning, it is updated at each time step, according to Eq. (9).

$$Q = \dot{\text{m}}(\text{h}_{\text{out}} - \text{h}_{\text{in}}), \qquad \dot{m}_{\text{in}} = \dot{m}_{\text{out}}\,, \qquad \frac{dU}{dt} = 0 \tag{8}$$

$$h_{in} = h_{skid}, \qquad h_{out} = h(P_{skid}, T = -40\ ºC) \tag{9}$$

where Q is the heat exchange of the gas with the exterior, $\dot{\text{m}}$ is the mass flow, h_{in} is the inlet enthalpy, h_{out} is the outlet enthalpy, h_{skid} is the Skid enthalpy, P_{skid} is the Skid Pressure.

3.2. Numerical implementation

The numerical implementation is formulated as a time-dependent zero-dimensional (0D) explicit model developed in the MATLAB programming language. Each control volume (tanks, pipes, or bus storage modules) is treated as a lumped, well-mixed node with uniform thermodynamic properties. All state evolution therefore occurs only in time, and no spatial discretization is required.

The time integration is carried out using the explicit Euler method, which updates all state variables based solely on information from the current time step. An adaptive time-stepping scheme is employed to maintain accuracy and stability, with the time step automatically reduced as the bus tank approaches its maximum capacity. This enables the correct characterization of the filling procedure for each individual vehicle. Other criteria for Adaptive Time-Stepping were deemed optional as the time step selected from the convergence study already reduced numerical error.

Beyond the thermodynamic modelling of hydrogen behavior, some operational and system control are also assumed. During refueling, a pressure switch will swap the skid's level that's refueling the tank according to the skid level pressure limits, and the pressure difference between the skid level and the vehicle tank (Δp). Considering that mass flow decreases with a smaller pressure difference, author established that there should be a pressure difference of at least 50 bar. If the condition is not met, the algorithm will select the next skid level to continue refueling.

3.3. Time step convergence study

Given its strong impact on the accuracy of time-dependent variables, the time step was selected as a key parameter for the convergence analysis. Since one of the objectives of this work is to predict the refueling time of a bus to approximately 3 minutes, particular attention must be paid to avoiding excessively coarse time steps that could distort the estimated filling duration. For the present study, a convergence criterium of 5% was established.

Figure 2 shows the filling time of three bus vehicles for a time step of 5 min to 0.05 min (3 sec). It is shown that with the time step refinement, the filling time of each vehicle will tend to the 3 min, despite the initial disparity in courser time steps. It is verified that the bus nº 1, 3 and 5 achieve a percentage variation below the criterium at different time steps (0.5, 2 and 0.2, respectively). With this in mind, every simulation model uses a maximum time step of 0.2 min along with an adaptive refinement scheme when reaching the end of the refueling.

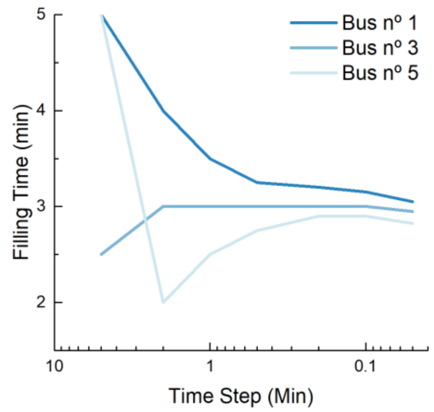

Figure 2. *Filling time of several BUS in function of the simulation time step.*

4. Results and discussion

Current model was developed from a set of simplifications and assumptions, explained in the previous section but without any form of cooling which will influence the current results. Figure 3 illustrates the transient pressure and temperature behavior of the hydrogen refueling process.

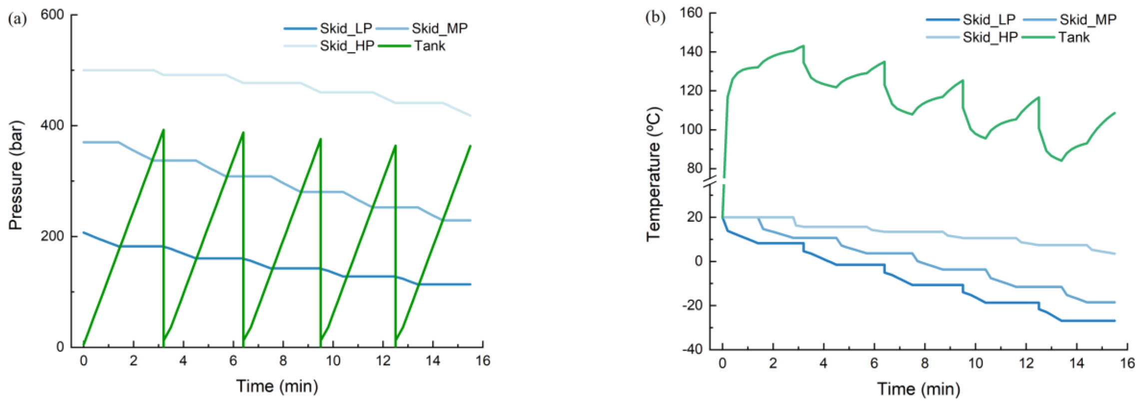

Figure 3. *(a) Simulated pressures and (b) temperature of skid storage banks (HP, MP, LP) and vehicle tank.*

The pressures of the three skid storage levels decrease gradually in a stepwise manner as hydrogen is withdrawn as it depletes each skid tank. As seen in Figure 3 (a), the tank pressure shows a linear pattern, as the model is designed to respect the set ARRP value. As the tank pressure reaches the pressure objective of 350 bar, pressure is reset to the initial pressure of the next vehicle, hence the sharp pressure in the graph. As skid levels are depleted with each refueling, the pressure available at each level is lower for the next vehicle, and the pressure switch is triggered sooner. This is evident because, with each refueling, the Low-pressure skid level gets used less. Example of this is the last bus barely uses the LP Skid.

Figure 3 (b) shows the same information but accompanied by the temperature in the vehicle tank. As pressure rises in the tank, temperature in the tank sharply increases in first moments and stabilizes before each skid tank switch. Within the first vehicle refilling, a maximum temperature of 109 ºC is reported, which far exceeds the safety threshold of 85 °C, according to SAE J2601 guidelines [16, 17]. This behavior occurs as a result of the compression heating and the Joule-Thomson effect, as well as the adiabatic boundary conditions. With no form of heat transfer, any compression process and mass transfer results in an internal energy increase, which results in the sharp increase of hydrogen temperature. For this reason, the introduction of a cooling elements is mandatory.

Having verified the sharp increase in temperature, the current model was developed in order to reduce the hydrogen temperature as it reaches the vehicle tank by means of heat exchanger unit. This last element is configured to cool the gas until −40 ºC at the outlet. Since the skid and tank are considered adiabatic, any heat removal derives solely from the heat exchange. Figure 4 illustrates the transient pressure, temperature behavior and heat removed of the hydrogen refueling process.

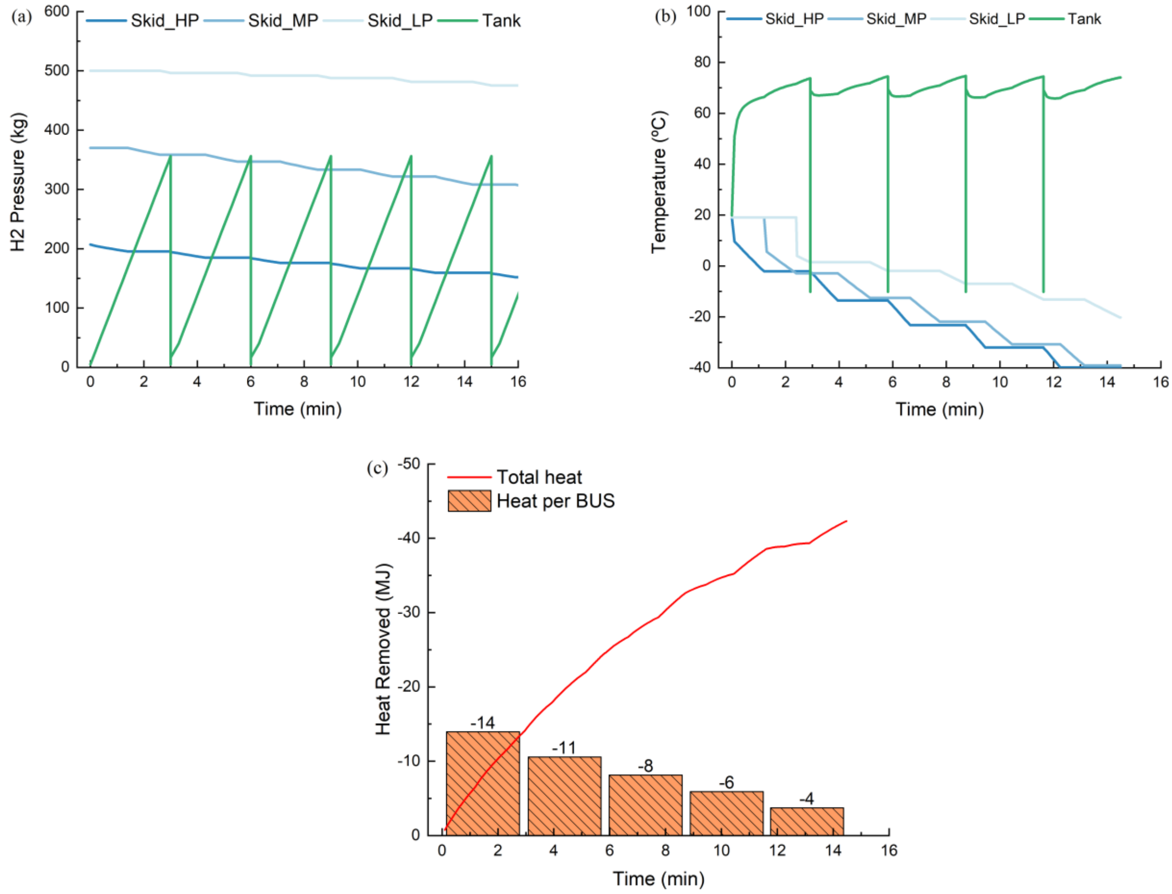

Figure 4. *(a) Simulated pressures, (b) temperature and (c) heat of skid storage banks (HP, MP, LP) and vehicle tank with active cooling.*

Figure 4 (a) shows the transient pressure evolution of the storage system and vehicle tank during refueling. Similar to before, behavior maintains the same sequential depletion of the three storage banks, as well as the linear pressure filling pattern, which reflects on the impact of pressure control scheme implemented. Figure 4 (b) shows the thermal behavior of the skid storage and the vehicle tank during refueling. The temperatures of the skid banks decrease down to −20 ºC and lower in steps as hydrogen is discharged, reflecting the cooling associated with expansion and depletion of the storage vessels. Conversely, the tank temperature increases throughout the cyclic filling process, reaching a maximum temperature of 74 ºC. Although this is below the 85 ºC maximum temperature established by the SAEJ2601, this is still considered an unsafe condition to manipulate high pressure hydrogen gas. This heating is mainly due to compression effects and the Joule–Thomson contribution.

Figure 4(c) show the heat flow removed from the hydrogen throughout all refiling and the total heat removed per individual refilling. The cumulative heat removed rises steadily, representing the cooling demand necessary to offset the thermal load generated by gas compression inside the tank. As the temperature of the skid decreases with the fueling, the gap between the gas influx and the target temperature of −40 ºC of the heat exchanger is reduced. This is reflected in lesser heat removal as more vehicles are refueled. If we compare the total heat removed for each vehicle, we observe that the first requires 14 MJ to be removed, while the last only requires 4 MJ.

With the development of current model, came the necessity to evaluate the impact of the ambient temperature on the performance of the heat exchanger and how would impact subsequent bus refueling. Figure 5 shows the amount of heat removed from the hydrogen for each at different ambient temperatures.

As expected, one of first conclusions to be drawn is that with a lower ambient temperature, the heat exchanger will have less energy to be removed in order for hydrogen to enter the vehicle at −40 ºC. It should be noted that below −10 ºC ambient temperature, heat needs to be supplied (and not removed) to the hydrogen gas in order to reach the temperature goal.

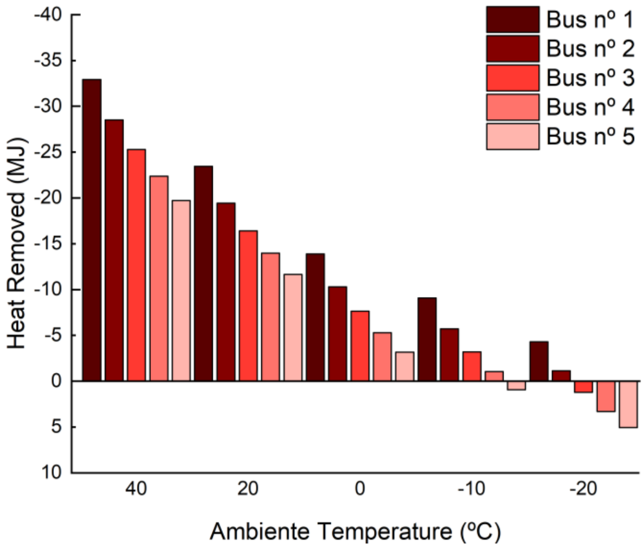

Figure 5. *Heat removal in function of ambient temperature.*

Also, as the skid levels transfer hydrogen mass to the tank, their pressure levels will quickly drop. Thus, to ensure the thermodynamic equilibrium in a constant volume, the hydrogen will decrease its own temperature. The same effect is also reported throughout the vehicle refueling, as any subsequent bus will require less heat to be removed than the previous. In sum, the energy to be removed by the heat exchanger is heavily dependent on the ambient temperature as well as the temperature drop caused by the depressurizing of each skid level.

5. Conclusions

This study developed a transient zero-dimensional model to simulate the thermodynamic behavior of a hydrogen refueling station for HDVs. The model integrates cascade storage dynamics, pressure control strategies and a pre-cooling heat exchanger in order to analyze pressure, temperature and cooling requirements during fast refueling. Compared to modelling approaches reported in the literature, which often simplify thermal effects or focus primarily on light-duty applications, the present work analysis the role of transient thermal behavior and real-gas effects.

Results showed that neglecting thermal effects leads to unrealistic predictions, with tank temperatures reaching up to 109 °C under adiabatic conditions, exceeding the 85 °C safety limit defined in SAE J2601. The integration of a heat exchanger pre-cooling hydrogen to −40 °C significantly reduced the maximum tank temperature to approximately 74 °C, allowing the refueling process to remain within safe operating limits. However, even under these controlled conditions, temperatures remain relatively high, reinforcing that compression heating remains a key aspect in high-pressure hydrogen dispensing.

The simulations also demonstrated that the thermodynamic state of the cascade storage system affects the cooling demand, with heat removal decreasing during successive refueling events as storage temperatures drop. This outcome highlights that the coupling between storage and heat exchanger performance has direct implications for system design and operational optimization. In addition, ambient temperature was found to strongly influence the heat exchanger load, reinforcing the need for climate-dependent design strategies, particularly for stations operating under variable environmental conditions.

Overall, the proposed model provides a useful tool for analyzing the performance and thermal behavior of hydrogen refueling stations and supports future optimization of heavy-duty hydrogen dispensing systems.

Nevertheless, some limitations need to be addressed: the current model is based on several simplifying assumptions, including adiabatic tank behavior, neglect of pressure losses and the use of a lumped parameter (0D) approach. While these assumptions enable computational efficiency and system-level analysis, they may limit the accuracy of localized thermal gradients and flow dynamics.

Future work should include a dedicated sensitivity analysis of the skid-tank pressure difference to quantify its impact on mass flow rate, refueling time and thermal behavior, enabling the identification of optimal pressure switching thresholds that balance fast filling performance with thermal safety and energy efficiency.

Acknowledgments

This work has been supported by the project "*Hi-MOV - Via de mobilidade tecnológica transfronteiriça com hidrogénio renovável, Referência 0160_HI_MOV_1_E, financiado pela CCDR-N - Comissão de Coordenação e Desenvolvimento Regional Norte, no âmbito do Cooperação Territorial Europeia - INTERREG, programa operacional Programa de Cooperação Transfronteiriça Espanha-Portugal (POCTEP) 2021-2027*". Also, authors would like to acknowledge the support from FCT – Fundação para a Ciência e Tecnologia within the R&D Unit Project Scope UID/00319/2025 – Centro ALGORITMI (ALGORITMI/UM) https://doi.org/10.54499/UID/00319/2025; as well as R&D Unit Project Scope UIDB/04077/2025 – Centro METRICS (METRICS/UM).

Nomenclature

Latin symbols

h	specific enthalpy, J/kg
m	mass, kg
$\dot{m}$	mass flow rate, kg/s
Δm	mass variation, kg
Q	heat transfer, J
Q	heat transfer rate, W
t	time, s
u	specific internal energy, J/kg
U	internal energy, J
V	volume, m³
W	work, J
p	pressure, Pa
T	temperature, °C

Greek symbols

ρ	density, kg/m³

Subscripts and superscripts

in	inlet condition
out	outlet condition
$tank$	vehicle tank
$skid$	storage system
HP	high pressure level
MP	medium pressure level
LP	low pressure level
i	current time step
$i-1$	previous time step

References

[1] Masnadi MS, Grace JR, Bi XT, et al. From fossil fuels towards renewables: Inhibitory and catalytic effects on carbon thermochemical conversion during co-gasification of biomass with fossil fuels. *Appl Energy* 2015; 140: 196–209.

[2] Cunanan C, Tran M-K, Lee Y, et al. A Review of Heavy-Duty Vehicle Powertrain Technologies: Diesel Engine Vehicles, Battery Electric Vehicles, and Hydrogen Fuel Cell Electric Vehicles. *Clean Technologies* 2021; 3: 474–489.

[3] Rosen MA, Koohi-Fayegh S. The prospects for hydrogen as an energy carrier: an overview of hydrogen energy and hydrogen energy systems. *Energy Ecol Environ* 2016; 1: 10–29.

[4] Zhang Y, Davis D, Brear MJ. The role of hydrogen in decarbonizing a coupled energy system. *J Clean Prod* 2022; 346: 131082.

[5] Benvenuti N, Milhe M, Guewouo T, et al. Levelized cost of hydrogen: A dynamic simulation-based methodology for heavy-duty refuelling stations. *Int J Hydrogen Energy* 2026; 206: 153374.

[6] Ludwig-Bölkow-Systemtechnik. Milestone reached: over 1,000 hydrogen refuelling stations in operation worldwide in 2024. *17th Annual assessment of H2stations.org by LBST*, https://www.h2stations.org/press-release-2025-milestone-reached-over-1000-hydrogen-refuelling-stations-in-op-eration-worldwide-in-2024/ (2025, accessed 11 February 2025).

[7] Nistor S, Dave S, Fan Z, et al. Technical and economic analysis of hydrogen refuelling. *Appl Energy* 2016; 167: 211–220.

[8] Genovese M, Cigolotti V, Jannelli E, et al. Current standards and configurations for the permitting and operation of hydrogen refueling stations. *Int J Hydrogen Energy* 2023; 48: 19357–19371.

[9] Bosu S, Rajamohan N. Recent advancements in hydrogen storage - Comparative review on methods, operating conditions and challenges. *Int J Hydrogen Energy* 2024; 52: 352–370.

[10] Zheng J, Ye J, Yang J, et al. An optimized control method for a high utilization ratio and fast filling speed in hydrogen refueling stations. *Int J Hydrogen Energy* 2010; 35: 3011–3017.

[11] Martorelli M, Genovese M, Fragiacomo P. Enhancing heavy duty vehicle hydrogen refueling by alternative approach to SAE J2601/2 protocol and flow dynamics. *Int J Hydrogen Energy* 2025; 101: 234–249.

[12] Caponi R, Monforti Ferrario A, Bocci E, et al. Thermodynamic modeling of hydrogen refueling for heavy-duty fuel cell buses and comparison with aggregated real data. *Int J Hydrogen Energy* 2021; 46: 18630–18643.

[13] Tatti R, Klopčič N, Radner F, et al. Thermodynamic Modelling and Sensitivity Analysis of a 70 MPa Hydrogen Storage System for Heavy Duty Vehicles. *Hydrogen* 2026; 7: 8.

[14] Ferreira AC, Ferreira JP, Teixeira SF, et al. Development of a Numerical Optimization Model for Sizing Hydrogen Refuelling Stations: Application to a Case Study. In: Blanco-Marigorta A, Gamero B, Martel N, et al. (eds) *36th International Conference on Efficiency, Cost, Optimization, Simulation and Environmental Impact of Energy Systems (ECOS 2023)*. Las Palmas De Gran Canaria, Spain: ECOS 2023, pp. 999–1010.

[15] Lemmon E, Huber M, McLinden M. NIST Standard Reference Database 23: Reference Fluid Thermodynamic and Transport Properties-REFPROP. Version 8.0: Natl Std. Ref. Data Series (NIST NSRD.

[16] Fueling Protocol for Gaseous Hydrogen Powered Heavy Duty Vehicles. Epub ahead of print 24 September 2014. DOI: 10.4271/J2601/2_201409.

[17] Reddi K, Elgowainy A, Rustagi N, et al. Impact of hydrogen SAE J2601 fueling methods on fueling time of light-duty fuel cell electric vehicles. *Int J Hydrogen Energy* 2017; 42: 16675–16685.