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Research Article

Utilization of the Refrigeration Effect from LPG Expansion for a Refrigerated Display Cabinet (RDC): An Experimental Study of a Half-Cycle System

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Abstract

Commercial refrigeration is energy-intensive and still relies mainly on electrically driven compressors. LPG, a propane–butane mixture widely used as cooking fuel, cools as it is throttled before combustion, but this effect is normally wasted. This study experimentally examines whether the effect, used in a half-cycle refrigeration system, can cool a small (192 L) beverage refrigerated display cabinet (RDC). Liquid LPG was throttled through an expansion valve into an evaporator inside the cabinet, and the outlet stream was burned. Temperatures, pressures and relative humidity were logged every minute for 60 minutes. The cabinet air temperature fell from 26.3 °C to 4.7 °C, and the humidity ratio fell by about 78%, showing continuous dehumidification. Mixture properties calculated with the Peng–Robinson equation of state show that the evaporator outlet (− 1.6 °C, 2.42 bar absolute) lay inside the two-phase region of this zeotropic mixture (dew point ≈ 7 °C). The 3.7 K temperature rise along the evaporator therefore reflects temperature glide, not superheat. At the measured LPG flow rate of 3.07 kg/h, the estimated refrigerant-side cooling rate was about 80 W (≈ 50–155 W across the uncertainty in composition and measurements). Only a small part of the absorbed heat appeared as a change in the energy of the cabinet air; the rest could not be apportioned without a full cabinet energy balance. The final air temperature lay within the range of product surface temperatures measured in a commercial minimarket RDC (5.4–8.3 °C), which is used here only as an illustrative reference. The results, obtained from a single 60-minute test, demonstrate preliminary cooling feasibility in the tested cabinet. Practical use as an auxiliary cooling source would require loaded-cabinet and longer-duration tests, and depends on LPG consumption coinciding with cooling demand.

Keywords: LPG; Half-cycle refrigeration system; Refrigerated display cabinet; Cooling effect; Zeotropic mixture

1. Introduction

Global cooling demand continues to increase alongside population growth, urbanization, and the expansion of food cold chains, while the refrigeration and air-conditioning sector accounts for a significant share of global primary energy consumption [1]. Meanwhile, heat pump technology and vapor-compression refrigeration systems continue to be developed to meet energy efficiency targets in both industrial and commercial sectors [2]. International regulatory pressures, particularly through the phasedown of high-Global Warming Potential (GWP) refrigerants under regulations such as the European Union's F-gas Regulation, are accelerating the transition from synthetic refrigerants (HFCs/HCFCs) toward alternatives with substantially lower environmental impacts [3].

Hydrocarbon (HC) refrigerants, particularly propane (R-290) and butane/isobutane (R-600/R-600a), have emerged as the most promising natural refrigerant candidates to replace synthetic refrigerants, owing to their zero Ozone Depletion Potential (ODP) and very low GWP (≈ 3), alongside competitive thermodynamic properties and compatibility with conventional lubricants [4], [5]. Recent literature reviews confirm that heat pumps and vapor-compression systems utilizing pure HCs can match or even surpass the energy performance of HFC-based systems in domestic and small-scale commercial applications [6], [7], although the flammability of these refrigerants remains a primary safety concern that limits the allowable charge size in sealed systems.

Reviews of refrigerant development [8] and experimental studies [9], [10] show that the performance of HC mixtures depends strongly on composition, while flammability and safe deployment remain the main constraints [11], [12]. HCs have also been applied in eutectic systems [13], water coolers [14] and isobutane refrigeration

systems [15], and several studies have addressed commercial coolers and low-temperature HC systems specifically [16], [17], [18], [19].

In terms of applications, commercial refrigerated display cases (RDCs) are among the equipment with the highest energy consumption intensity in the food retail sector, primarily because their designs are generally open-type, making them susceptible to warm air infiltration from the surrounding environment [20]. Various recent studies have focused on improving RDC energy efficiency through air curtain design optimization [21], [22], [23]. Demirpolat [21] reported an energy consumption reduction of up to approximately 40% through a multi-airflow air curtain design. Meanwhile, Suamir *et al*. [24], in a context relevant to the present study, evaluated the thermal and energy performance of a commercial refrigerated display case specifically integrated with hydrocarbon refrigerants. A recent systematic review of the commercial food sector further emphasized that natural refrigerant strategies and closed-door RDCs represent key directions in the energy transition of food retail buildings [25]. In addition, the use of phase change materials (PCMs) as thermal energy storage media has been investigated to shift peak cooling loads in RDCs [26].

LPG, as a mixture of propane and butane, is required by Indonesian regulations to contain at least 97% propane and butane, based on the specifications issued by the Directorate General of Oil and Gas [27], although the specific mixture ratio may vary among commercial products [28]. Since LPG is essentially a hydrocarbon refrigerant, the pressure drop it experiences when passing through a regulator or expansion valve before combustion produces a cooling effect similar to the throttling process in a conventional vapor-compression refrigeration cycle. However, in household and commercial LPG installations, this cooling effect has traditionally been allowed to dissipate without being utilized.

The concept of utilizing this cooling effect was first systematically developed by the research group at Universitas Muhammadiyah Magelang through what they termed a half-cycle refrigeration system for LPG-fueled vehicles**.** Setiyo *et al*. [29] initiated this investigation through numerical analysis to evaluate the cooling potential of an LPG vehicle vaporizer, followed by laboratory-scale investigations into mini cooler applications [30] and calculations of the system's refrigeration effect and energy efficiency ratio (EER) [31]. Subsequent studies evaluated evaporator effectiveness [32] and the cooling power characteristics of a system equipped with an auxiliary chiller as a heat exchanger, reporting an actual cooling effect of up to 1.2 kW at LPG mass flow rates exceeding 3 g/s [33]. Waluyo *et al*. [34] further characterized the cooling effect of the half-cycle refrigeration system in LPG-fueled vehicles, achieving up to 373 W, while Ryastama *et al*. [35] confirmed similar findings across variations in engine speed and blower velocity. Collectively, these studies consistently demonstrate that LPG half-cycle refrigeration systems provide significant cooling effects and have potential for use as secondary air-conditioning systems operating alongside the primary cooling system in vehicles.

Meanwhile, for non-vehicle applications, the concept of a *zero-cost refrigeration system* was introduced by Mohan [36] in 2013 through the utilization of LPG as both a refrigerant and a fuel. This concept was subsequently developed through the design of LPG refrigeration systems incorporating capillary tubes and evaporators [37], [38]. In 2021, Shahare *et al*. [39] fabricated a low-cost LPG refrigeration system, while Ali *et al*. [40] demonstrated the capability to achieve cooling temperatures of up to 0°C in a 30 L compartment within 30 minutes in 2023. Furthermore, Manohar *et al*. [41] evaluated the design of a low-cost LPG refrigeration system in 2020. Collectively, these studies demonstrate the potential for utilizing LPG as both a refrigerant and a fuel in alternative cooling systems.

Published work on LPG half-cycle refrigeration has focused on vehicle cabins and small domestic compartments. Two questions remain open for a stationary, closed display cabinet. First, can the cooling available at a typical commercial LPG consumption rate pull down a cabinet of practical size into the temperature range used for chilled beverages, and how quickly? Second, how much of the LPG refrigeration effect is actually available inside the cabinet? LPG is a zeotropic propane–butane mixture that evaporates over a temperature glide, so the state at the evaporator outlet cannot be inferred from temperature alone; treating the outlet as superheated vapour when it is still two-phase leads directly to an overestimated cooling capacity. A further practical constraint is that a half-cycle system cools only while LPG is being burned. Its usefulness therefore depends on fuel use coinciding with cooling demand, as may occur in kitchens, restaurants and food businesses that burn LPG for many hours a day [4], [5], [20], [21].

This study addresses these questions experimentally. We measured the pull-down of a closed 192 L RDC cooled only by LPG expansion, together with the refrigerant pressures and temperatures and the humidity of the cabinet air. Using mixture properties from the Peng–Robinson equation of state, we determined the phase at the evaporator inlet and outlet and estimated the refrigerant-side cooling rate and its uncertainty. We then compared the change in energy stored in the cabinet air with the estimated evaporator heat removal, and set the final cabinet temperature against product surface temperatures in a commercial RDC as an illustrative reference. The results are intended as a preliminary feasibility assessment, not a performance rating.

2. Methods

This study was conducted experimentally using a laboratory-designed and assembled half-cycle refrigeration system test rig, with LPG serving as the refrigerant. In this experimental setup, LPG functions not only as the refrigerant but also as the fuel for the burner.

2.1. Test Object and Experimental Apparatus Design

The research object was a half-cycle refrigeration system test rig using LPG as the working fluid (assumed to consist of 50% propane and 50% butane), mounted on a wheeled frame for ease of mobility. The LPG cylinder was positioned upside down on a digital weighing scale to allow LPG to exit the cylinder in a saturated liquid phase (T_1, P_1). The liquid LPG was then expanded isenthalpically through an expansion valve into the inlet evaporator (T_2, P_2), which was installed inside a 192 L refrigerated display case (RDC). The LPG leaving the evaporator (T_3, P_3) was subsequently directed to the burner for combustion, allowing continuous LPG flow through the evaporator throughout the testing period. The process flow diagram, measurement points, and data acquisition system architecture are presented in **Figure 1**.

The RDC is a showcase unit, typically used for displaying chilled beverages, designed as an upright, single-door cabinet with an internal volume of 192 liters, featuring a glass door and foam-insulated walls. Testing was conducted with the unit empty and the door kept closed throughout the entire test sequence. The evaporator is an aluminium finned-tube heat exchanger repurposed from the condenser of a passenger-car air conditioner. Its face area matches the internal cross-section of the cabinet, so it spans the entire ceiling of the cabinet interior. The expansion valve was a diaphragm-type acetylene regulator of the kind used in oxy-acetylene welding. It was set to an outlet pressure of 1.5 barg, consistent with the measured evaporator inlet pressure of 1.40–1.49 barg, and this setting was kept constant throughout the test. Because the regulator holds the evaporator pressure, the LPG flow rate is set by the consumption of the burner downstream. A low-power laptop cooling fan was installed to circulate air across the evaporator and operated continuously throughout the test. A high-pressure burner was used as a heat source, positioned about 2 m from the cabinet without a heat shield.

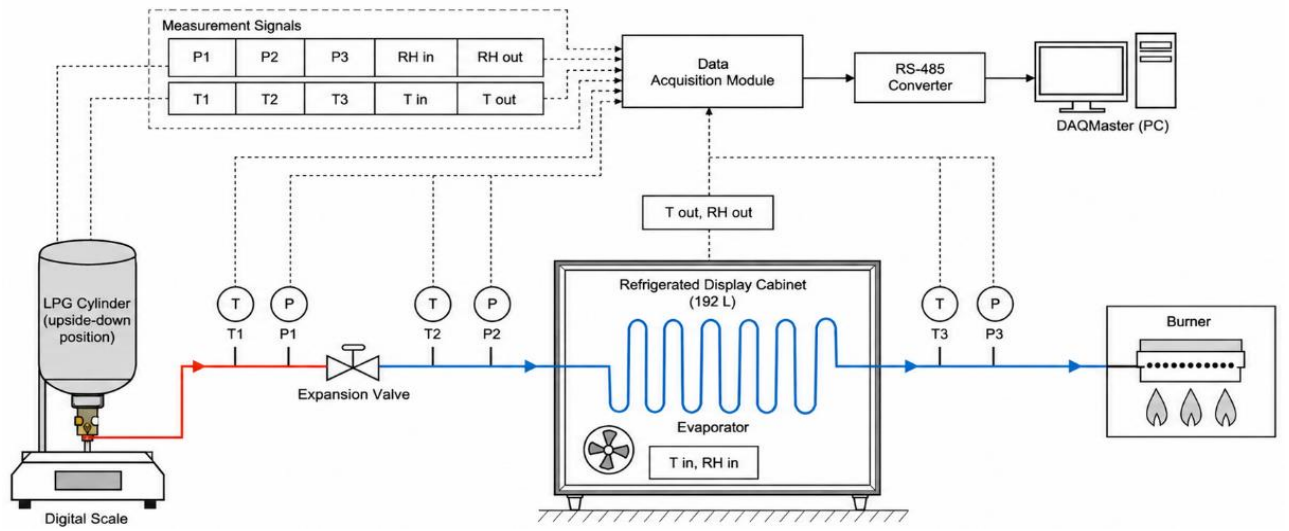


Figure 1. Schematic of the LPG half-cycle refrigeration system test rig, including measurement points and data acquisition system architecture

2.2. Instrumentation and Data Acquisition

Temperature was measured at five points (T_1, T_2, T_3, T_{in}, and T_{out}) using PT100 RTD sensors. Pressure was measured at three points (P_1, P_2, and P_3) using PSAN pressure transducers with a measurement range of 0–1000 kPa (0–10 bar), while relative humidity was measured at two points (RH_{in} and RH_{out}) using RH sensors, as presented in **Figure 1**. All sensors were connected to Autonics data acquisition modules (TM4 Basic, TM4 Expand, and TK4S), powered by a 24 VDC power supply, and connected to a computer through an RS-485 communication converter, as shown in the upper section of **Figure 1**. Data from all sensors were automatically recorded using DAQMaster software (Autonics) at 1-minute intervals over a 60-minute testing period, generating 60 time-series data points for each measured variable. The PT100 sensors had an accuracy of $\pm(0.15 + 0.002|T|)$ °C, the PSAN pressure transducers had an accuracy of 1% of full scale according to the manufacturer's datasheet, the relative humidity (RH) sensors had an accuracy of ±2% RH, and the digital scale had a resolution of 0.01 kg. The temperature sensors at measurement points T_1–T_3 were installed either directly in the flow or on the pipe surface with thermal insulation. The locations of all temperature, pressure and humidity sensors, including T_{in} inside the RDC and T_{out} in the surrounding air, are shown in **Figure 1**.

2.3. Testing Procedure

The experiment began by ensuring that all sensors and data acquisition modules were properly installed and that the DAQMaster software was ready to record data. Before the test, all PT100 sensors were left at the same ambient temperature until their readings stabilised, and their offsets were adjusted in DAQMaster so that all sensors showed the same reading. The pressure transducers were zeroed in the same way at atmospheric pressure. This single-point offset adjustment removes differences between sensors, but it does not verify their linearity over the measured range; the manufacturer's accuracies were therefore used in the uncertainty analysis. The expansion valve was then opened to initiate the flow of liquid LPG from the cylinder to the evaporator. Data recording at all measurement points commenced simultaneously at 1-minute intervals over a 60-minute testing period. LPG exiting the evaporator was directed to and combusted in the burner throughout the test duration to maintain continuous LPG flow through the evaporator.

The average LPG mass flow rate was determined gravimetrically by dividing the difference in the mass of the LPG cylinder and its accessories placed on the digital weighing scale between the beginning and end of the test by the test duration (60 minutes). This method yielded an average LPG mass flow rate of 3.07 kg/h, which was used to calculate the cooling power associated with the phase change of LPG. To provide comparative data on the operating conditions of commercial RDCs, surface temperature measurements were also performed on a conventional electrically compressor-driven RDC unit at a minimarket using a TG165-X thermal camera. Because the scale was read only at the start and end of the test, this flow rate is a time average; variation during the test, including the initial transient, is not resolved. The test was performed only once, without repetition. Measurement uncertainties were propagated to the derived quantities (mass flow rate, specific refrigeration effect and cooling rate) with the root-sum-square method of Kline and McClintock [42]. Because the evaporator outlet lies within the two-phase region (Section 3.4), the cooling-rate uncertainty is dominated by the assumed LPG composition and by the outlet pressure and temperature, so these were varied explicitly. These uncertainties reflect instrument accuracy and property assumptions only; test-to-test repeatability could not be assessed and is a limitation of this study.

2.4. Data Analysis Method

The time-series data were used to analyse the pressure, temperature and relative-humidity behaviour in the RDC. Thermodynamic properties of the LPG mixture (bubble and dew points, vapour quality and enthalpy) were calculated with the Peng–Robinson equation of state [43], using standard critical constants and ideal-gas heat capacities [44]. These were used to determine the phase at each measurement point and the specific refrigeration effect. All pressures used in these calculations are absolute (gauge + 1.013 bar). Heat removed from the RDC air was calculated with the ASHRAE psychrometric relations [45], with the humidity ratio obtained from the Buck saturation vapour-pressure equation [46]. Transient behaviour was characterised by fitting a first-order (lumped-capacitance) model to the cabinet air temperature.

3. Results and Discussion

The LPG used is Bright Gas by Pertamina, assuming a mixture composition of 50% propane and 50% butane by mass; it should be noted that official LPG specifications in Indonesia only mandate a minimum propane-plus-butane content of 97% without stipulating a fixed ratio between the two [27], so the 50:50 ratio used here refers to the specific product tested and may differ for other LPG products [28]. A single test was conducted over a 60-minute period with data recorded at 1-minute intervals. The LPG cylinder was positioned upside down to ensure the LPG exited the cylinder as a saturated liquid at conditions T_1 and P_1. The LPG then underwent isoenthalpic expansion through an expansion valve into an evaporator placed inside a 192-liter RDC, entering the evaporator at conditions T_2 and P_2. The LPG exiting the evaporator at conditions T_3 and P_3 was subsequently directed to the burner. Measurement data indicate an initial transient period of approximately 10 minutes before the system reached a quasi-steady state, as described in the following subsection. The measured cylinder-outlet pressure (6.89 bar absolute at 26.4 °C) is close to the bubble pressure calculated for a 50:50 mixture (6.54 bar) and corresponds to a propane mass fraction of about 55% (53–56% within the ±0.1 bar accuracy of the pressure transducer). Mass fractions of 50% and 55% were therefore both used in the property calculations (Section 3.4).

3.1. LPG Pressure Behavior

Figure 2 shows that the pressure upstream of the expansion valve (P_1) remained relatively stable in the range of 5.84–6.05 barg, with an average of 5.88 barg under quasi-steady-state conditions. This value represents the saturation pressure of the liquid LPG mixture within the cylinder at a temperature of $T_1 \approx 26.4$ °C. The slight downward trend (from 6.05 to 5.86 barg) is not caused by the falling amount of LPG, because the pressure of a saturated liquid depends on its temperature and composition, not on the quantity in the cylinder. A more likely cause is a change in liquid composition: as liquid is withdrawn, the growing vapour space is filled mainly by evaporation of the more volatile propane, so the remaining liquid becomes slightly richer in butane. Slight cooling of the cylinder wall may also contribute. At points P_2 and P_3 (evaporator inlet and outlet), the pressure dropped sharply from initial values of 1.59–1.60 barg (at the 1-minute mark, before the expansion valve had fully opened the flow) to a stable range of 1.40–1.49 barg in less than 2 minutes after full LPG flow through the expansion valve was established. The average P_1/P_3 pressure ratio, calculated from absolute pressures (6.89 and 2.42 bar), was approximately 2.85 under quasi-steady conditions, confirming a significant pressure drop caused by the adiabatic throttling process at the expansion valve. The difference between P_2 and P_3 was negligible (averaging only 0.02 bar); consequently, the pressure drop along the evaporator piping was considered negligible given the relatively low LPG mass flow rate used in this test.

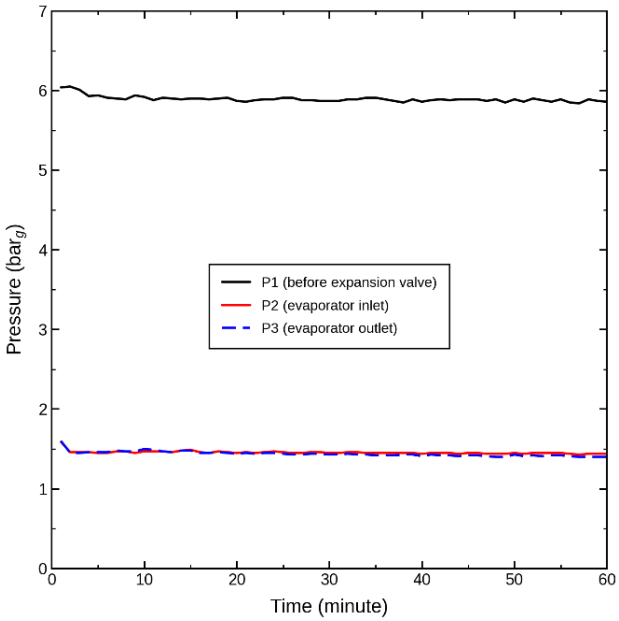


Figure 2. LPG pressure profile at three measurement points during the 60-minute test

3.2. LPG Temperature Behavior

Temperature T_1 remained relatively constant within the 26.2–26.9 °C range throughout the test, confirming that the LPG cylinder acts as a saturated liquid reservoir with a temperature determined by equilibrium with the surrounding environment. Temperature T_2 dropped drastically from 25.2 °C (at the 1-minute mark) to approximately -5 °C in just two minutes, subsequently stabilizing between -5.0 and -5.6 °C (averaging -5.33 °C over the final 20 minutes). This sudden drop was a direct consequence of the isenthalpic expansion process; as a portion of the liquid LPG underwent instantaneous vaporization (flash evaporation) while passing through the expansion valve, it absorbed latent heat from the remaining liquid, causing the mixture's temperature to fall to the new saturation temperature corresponding to pressure P_2.

Interestingly, T_3 did not immediately mirror T_2 at the start of the test; instead, T_3 stood at 27.1 °C during the first minute and only dropped significantly after the fourth minute, gradually approaching T_2 by the 9th or 10th minute. This pattern indicates a transient period driven by the thermal inertia of the evaporator's metal mass and connecting piping. Initially, these components were at room temperature, so the flowing LPG absorbed heat from them, exiting the evaporator in a highly superheated state. Once the evaporator components had sufficiently cooled (after approximately 10 minutes), the system reached a quasi-steady state. During the final 20 minutes, the difference (T_3–T_2) averaged 3.7 °C. This difference is not superheat. LPG is a zeotropic mixture: at the outlet pressure (2.42 bar absolute), its bubble and dew points are about − 7.6 °C and 7.0 °C for a 50:50 composition (Section 3.4). An outlet temperature of − 1.6 °C is therefore well below the dew point, so the LPG leaving the evaporator was still a two-phase mixture, and the temperature rise along the evaporator reflects the temperature glide of the evaporating mixture. Only during the first minutes, when T_3 was above about 7 °C, was the outlet genuinely superheated. The air temperature inside the RDC (T_{in}) dropped continuously and nearly monotonically from 26.3 °C to 4.7 °C (a total decrease of 21.6 °C) over a 60-minute period, with the rate of cooling slowing down over time—a pattern characteristic of a first-order cooling process (Newtonian cooling) approaching a new equilibrium between the heat absorbed by the evaporator and the heat entering from the surroundings. Conversely, the ambient temperature (T_{out}) surrounding the RDC rose from 27.8 °C to 30.1 °C (an increase of 2.3 °C) because the burner was operating only about 2 m from the RDC without a heat shield. Consequently, the temperature difference (T_{out} - T_{in}), which acts as the driving force for heat transfer into the RDC, widened from approximately 1.5 °C at the start of the test to over 25 °C by the end, as shown in **Figure 3**. The burner thus imposed an additional heat load on the cabinet, both through the warmer surrounding air and by thermal radiation, particularly through the glass door. This load would be absent in a typical installation, where the burner and the display cabinet are in separate areas, so in this respect the measured pull-down is conservative. Future tests should place a heat shield between the burner and the cabinet or increase their separation.

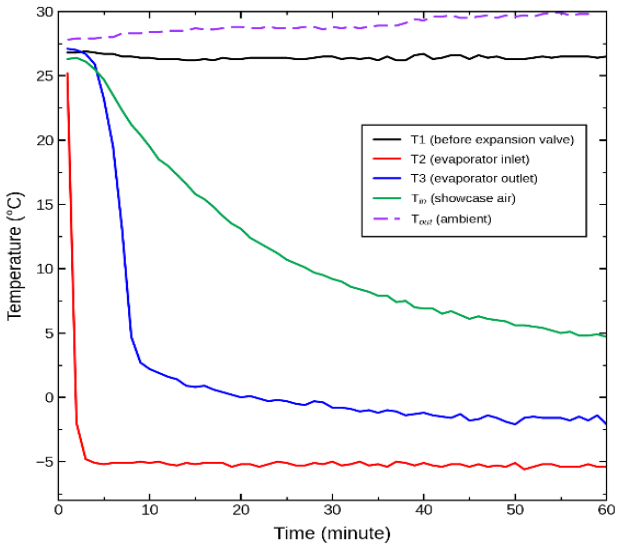


Figure 3. LPG and air temperature profiles during the 60-minute test

3.3. Relative Humidity Behavior Inside the RDC

The relative humidity (RH) inside the RDC exhibits a characteristic non-monotonic pattern associated with cooling accompanied by dehumidification, as shown in **Figure 4**. During the first three minutes, RH_{in} remained within the range of 71.6–74.4%, reflecting the RDC air conditions, which were still in equilibrium with the surrounding environment before the cooling effect became significant. Subsequently, RH_{in} decreased sharply, reaching a minimum of 40.2% at minute 7. This coincided with the period when the evaporator surface became sufficiently cold to cool the surrounding air below its dew point, causing water vapor to condense on the evaporator surface and be removed from the air through dehumidification.

After minute 7, RH_{in} gradually and consistently increased, reaching 66.5% at the end of the test (minute 60), even though T_{in} continued to decrease significantly from 22.3 °C to 4.7 °C over the same period. The increase in RH during continuous cooling appears counterintuitive, since cooling at a constant water vapor content should cause RH to increase toward saturation. To confirm what actually occurred in terms of water vapor content, the humidity ratio (W) was calculated from the T_{in} and RH_{in} data using the Buck saturated vapor pressure equation [46] and standard psychrometric relationships [45]. The results (right axis of **Figure 4**) show that W decreased monotonically throughout the 60-minute test, from 15.7 g water vapor/kg dry air in the first minute to only 3.5 g/kg at minute 60, representing an approximately 77.7% reduction in absolute water vapor content. This demonstrates that dehumidification (condensation/freezing of water vapor on the evaporator surface) occurred continuously throughout the test. The increase in RH_{in} after minute 7 was primarily attributed to the decrease in saturated vapor pressure as air temperature declined, consistent with the Clausius–Clapeyron relationship. This

decrease occurred more rapidly than the reduction in the actual water vapor content, causing the relatively low remaining moisture content to be expressed as an increase in RH.

For comparison, RH_{out} in the environment surrounding the RDC decreased gradually from 57.1% to 48.7%, in conjunction with the burner-induced increase in T_{out} and relatively stable environmental water vapor content. This pattern was not associated with LPG evaporation and reinforces the interpretation that the strong dehumidification trend observed inside the RDC was primarily a localized effect of evaporator cooling, rather than a general change in the surrounding room air conditions.

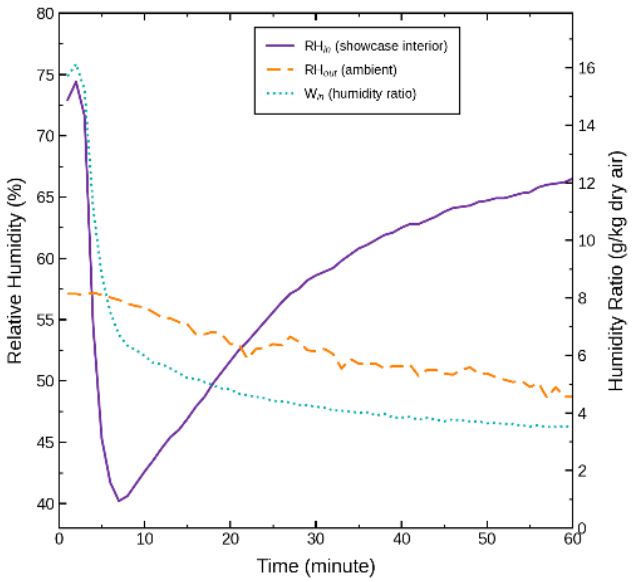


Figure 4. Relative humidity inside the RDC and the surrounding environment, along with the humidity ratio of the RDC air

3.4. Refrigeration Effect and Cooling Rate of the LPG Mixture

The specific refrigeration effect (cooling effect) associated with the phase change of LPG in the evaporator was calculated using the classical energy balance approach for refrigeration cycles [47]. Since the expansion valve operates adiabatically without work, the specific enthalpy remains constant (isenthalpic process). Therefore, the specific enthalpy of LPG entering the evaporator (h_2) is equal to the saturated liquid enthalpy before the expansion valve (h_1). The specific refrigeration effect absorbed by LPG throughout the evaporator, from the inlet condition (h_1) to the outlet condition (h_3), is expressed in Eq. (1):

$$q_{evap} = h_3 - h_2 = h_3 - h_1 \quad (1)$$

Representative quasi-steady-state conditions were derived from the average of the final 20 minutes of testing (minutes 41–60), as summarized in **Table 1**.

Table 1. Thermodynamic conditions of LPG at three measurement points (average of minutes 41–60; values in parentheses are absolute pressures, bar)

Point	Location	*T* (°C)	*P* (barg)	Phase
1	Before expansion valve	26.44	5.88 (6.89)	Saturated liquid
2	Evaporator inlet	-5.33	1.44 (2.45)	Liquid-vapor mixture (flashed) (X ≈ 0.20)
3	Evaporator outlet	-1.64	1.41 (2.42)	Two-phase mixture (X ≈ 0.42; dew point ≈ 7.0 °C)

Thermodynamic properties of the propane–n-butane mixture were calculated with the Peng–Robinson equation of state [46] (binary interaction parameter k_{ij} = 0.003), with ideal-gas heat capacities from [44]. For the pure components the model reproduces reference saturation pressures and latent heats between −5 °C and 26 °C to within about 1%. Unlike a mass-weighted average of pure-component properties, this approach accounts for the temperature glide of the zeotropic mixture and gives the vapour quality of each state directly. The results for propane mass fractions of 50% (base case) and 55% (fitted to the measured P_1) are listed in **Table 2**.

Table 2. LPG mixture states calculated with the Peng–Robinson equation of state (quasi-steady averages, minutes 41–60)

Quantity	50:50 (base case)	55:45 (fitted to P1)	Measured
Bubble pressure at T_1 = 26.44 °C (bar abs.)	6.54	6.89	6.89
Inlet temperature after throttling, T_2 (°C)	−4.7	−6.6	−5.33
Inlet vapour quality, X_2 (–)	0.20	0.21	–
Bubble/dew temperature at P_3 = 2.42 bar abs. (°C)	−7.6 / 7.0	−9.3 / 5.0	T_3 = −1.64
Outlet vapour quality, X_3 (–)	0.42	0.57	–
Specific refrigeration effect, $q_{evap} = h_3 - h_1$ (kJ/kg)	93	148	–
q for complete evaporation to the dew point (kJ/kg)	329	325	–
Cooling rate at $\dot{m}$ = 3.07 kg/h (W)	80	127	–

Because the outlet lies within the two-phase region, the specific refrigeration effect must be evaluated from the enthalpies of the actual states (Eq. 1), not as the sum of a latent heat and a superheat term, which would assume

complete evaporation. The phase of each state is described by its vapour quality X, obtained from the lever rule (Eq. 2), where h_L and h_V are the enthalpies of the coexisting liquid and vapour at the same temperature and pressure:

$$X = \frac{h - h_L}{h_V - h_L} \tag{2}$$

$$X_2 \approx 0.20\ (inlet, h = h_1),\ X_3 \approx 0.42\ (outlet)$$

For the 50:50 base case, about 20% of the liquid flashes to vapour across the expansion valve, and only about 42% of the mixture is vapour at the evaporator outlet (57% for the 55:45 mixture). The inlet quality agrees with the simplified estimate $c_{pl}(T_1 - T_2)/h_{fg} = 77.9/380.3 \approx 0.205$. The specific refrigeration effect then follows from Eq. (3):

$$q_{evap} = h_3(T_3, P_3) - h_1(T_1, P_1) \tag{3}$$

$$q_{evap} \approx 93\ kJ/kg\ (50{:}\,50);\ \ 148\ kJ/kg\ (55{:}\,45)$$

This is less than one-third of the ≈ 329 kJ/kg that would be obtained if the mixture evaporated completely. About 58% of the LPG mass therefore leaves the cabinet still liquid and is carried to the burner. Two consequences follow. First, much of the available refrigeration effect is not used inside the cabinet; because the evaporator already spans the full internal ceiling of the cabinet, a higher outlet quality would require additional heat-transfer area elsewhere in the cabinet (for example, on the rear wall), stronger air circulation than the small laptop fan used here provides, or a lower LPG flow rate, and subcooling the liquid before the valve would reduce flash gas. Second, liquid LPG reaching the burner is a safety concern, so a practical system must ensure complete evaporation (for example, with a downstream vaporiser) before combustion. Based on the measured average flow rate of 3.07 kg/h (8.53×10^{-4} kg/s), the refrigerant-side cooling rate is given by Eq. (4):

$$Q_{evap} = \dot{m} \times q_{evap} \tag{4}$$

$$Q_{evap} = 8.53 \times 10^{-4}\ kg/s \times 93\ kJ/kg \approx 0.080\ kW \approx 80\ W$$

This value (≈ 80 W for the 50:50 base case) is an estimate of the refrigerant-side cooling rate during the quasi-steady period, not a measured capacity. It is very sensitive to the outlet state. Because T_3 lies within the glide, a change of 0.2 K in T_3 alters the estimate by about 5 W, and moving from 50% to 55% propane raises it from 80 W to 127 W. Including the sensor accuracies (±0.16 K in T_3 and ±0.1 bar, i.e. 1% of full scale, in P_2 and P_3) gives a range of roughly 50–155 W. The pressure accuracy has the largest effect: an error of ±0.1 bar in P_3 shifts the bubble and dew points and changes the estimate by about ±25 W. The calculated states also apply only to the quasi-steady period (minutes 41–60). During the first ≈ 10 minutes the outlet was superheated and the flow rate was not resolved in time, so no single value of q_{evap} applies to the whole test.

3.5. Air Heat Absorption in the RDC

The heat absorbed from the air inside the RDC is calculated using a psychrometric energy balance approach [45], treating the RDC (192 L volume) as a closed control volume containing a fixed mass of air (a simplified assumption excluding infiltration or air exchange with the environment, given the lack of infiltration rate data). The enthalpy of the moist air is calculated from the temperature (T_{in}) and humidity ratio (W_{in}) using standard ASHRAE psychrometric equations [45]:

$$h = 1.006 \cdot T + W \cdot (2501 + 1.86 \cdot T)\ \ [kJ/kg\ dry\ air] \tag{5}$$

The dry air mass within the RDC was calculated based on the RDC volume (192 L) and the specific volume of moist air at initial conditions, yielding a dry air mass of 0.221 kg. The calculation results for the initial (1st minute) and final (60th minute) test conditions are summarized in **Table 3**.

Table 3. RDC air conditions at the beginning and end of the test

Parameter	Minute 1	Minute 60
Temperature, T_{in} (°C)	26.3	4.7
Relative humidity, RH_{in} (%)	72.9	66.5
Humidity ratio, W (g/kg)	15.7	3.5
Enthalpy of moist air, h (kJ/kg)	66.50	13.53

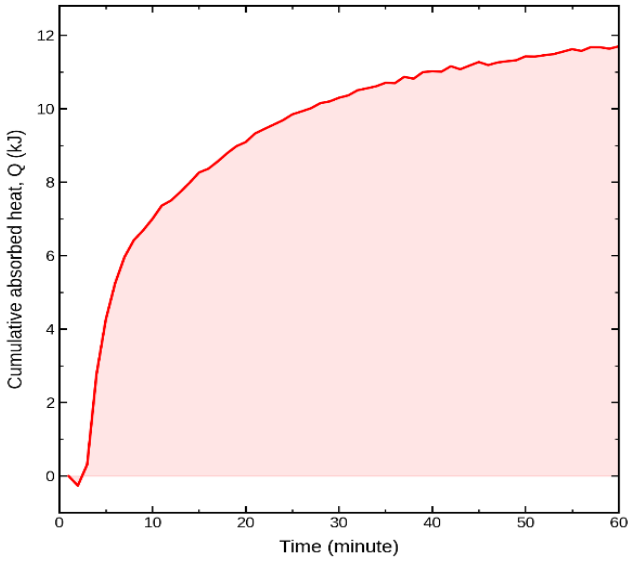


Figure 5. Cumulative heat absorbed from the RDC air (192 L) over the 60-minute test duration

The total heat absorbed from the RDC air during the 60-minute test was $\Delta Q = m \times \Delta h = 0.221$ kg × (66.50 − 13.53) kJ/kg ≈ 11.69 kJ, comprising a sensible heat component of ≈4.88 kJ (±41.7%) due to the drop in air temperature and a latent heat component of ≈6.73 kJ (±57.6%) due to water vapor condensation (equivalent to ±2.7 grams of water condensed from the RDC air). REF _Ref240716081 \h The average rate of change of the air energy over the test was approximately 3.25 W. As shown in **Figure 5**, the rate of heat absorption is not constant over time; during the first 10 minutes, the average rate reaches approximately 11.7 W, significantly higher than the ±0.45 W observed during the final 10 minutes. This decline is consistent with two factors: (1) a first-order cooling process approaching a new equilibrium, resulting in a reduced temperature difference driving heat transfer; and (2) the absorption of the bulk of the readily extractable latent heat (derived from the initial high moisture content) during the early phase of the test, causing the latent heat contribution to diminish substantially in the final period.

The value of 3.25 W is the average rate of change of the energy stored in the cabinet air. It is not the evaporator heat removal. Besides the air, the evaporator also cools the cabinet walls, shelves and glass door and its own metal mass, condenses (and possibly freezes) moisture, removes the heat released by the laptop fan (of the order of a few watts), and offsets heat gains through the walls, by infiltration and from the nearby burner, which grew as $T_{out} - T_{in}$ rose from ≈ 1.5 °C to over 25 °C.

Over the 60-minute test, the change in air energy (11.7 kJ) was small compared with the estimated refrigerant-side heat removal. Applying the quasi-steady value of q_{evap} to the whole test gives roughly 290 kJ (50:50) to 460 kJ (55:45), although this extrapolation is uncertain for the transient period. The ratio between the two should not be read as a cooling utilisation efficiency. The 192 L cabinet holds only about 0.22 kg of dry air, so the energy stored in the air is necessarily a small part of the total, even in a perfectly insulated cabinet.

The present data do not allow the remaining heat to be divided between the cabinet structure, the evaporator mass, the condensate, the fan heat and the heat gains from the surroundings (including the burner), so we do not attribute it to any single cause such as poor insulation or infiltration. A complete cabinet energy balance, $Q_{evap} = \Delta U_{air} + \Delta U_{structure} + \int UA(T_{out} - T_{in})dt + Q_{latent}$, would require the overall heat-transfer coefficient UA of the cabinet (for example, from a reverse heat-leak test with an internal electric heater), the thermal mass of the cabinet interior, the fan power and the mass of condensate collected, with the burner shielded or moved away so that the surrounding temperature stays constant. These measurements are recommended for future work.

3.6. Comparison with commercial RDCs and potential applications

To provide a practical benchmark against real-world operating conditions, surface temperature measurements were conducted on a conventional electric compressor-based commercial refrigerated display cabinet (RDC) at a convenience store using a handheld thermal camera (model TG165-X). The measurements revealed that the surface temperatures of products on the packaged beverage shelves ranged from 5.4 °C to 8.3 °C (**Figure 6**), representing the typical operating temperature range for ready-to-sell packaged beverages in modern Indonesian retail outlets. These are surface temperatures of stocked products in a cabinet at steady operation, not air temperatures.

Based on the first-order fit in **Figure 7**, the cabinet air in this study reached 8.3 °C after 34 minutes and 5.4 °C after 53 minutes. This comparison is illustrative only. Our values are air temperatures in an empty cabinet during pull-down, whereas the commercial values are product surface temperatures at steady operation; product temperatures lag the air temperature, and a loaded cabinet would cool more slowly. The result shows that LPG expansion alone, without a compressor, brought the air in this cabinet to a temperature typical of beverage display, but it does not demonstrate equivalent refrigeration performance. The estimated cooling rate (≈ 80 W; 50–155 W)



Figure 6. Thermal camera images (TG165-X) of a commercial refrigerated display case (RDC) at a minimarket, showing a minimum measured temperature of 5.4 °C (a) and a maximum of 8.3 °C (b)

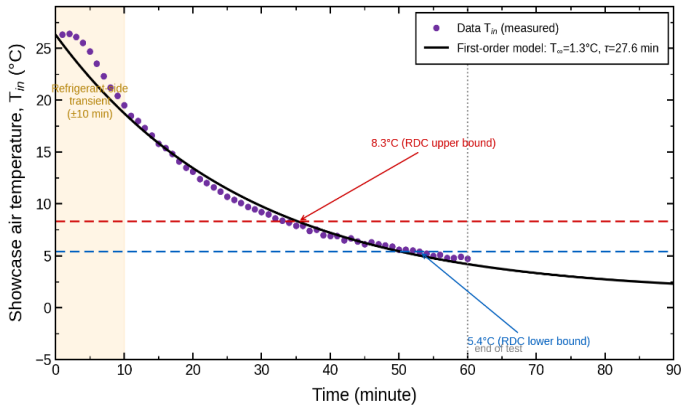


Figure 7. Cooling curve of T_in along with the first-order model ($\tau \approx 27.6$ minutes), and a comparison with the operating temperature range of commercial RDCs in convenience stores (5.4–8.3 °C)

is lower than values reported for LPG half-cycle systems in vehicles, for example 373 W [35], up to 1.2 kW at flow rates above 3 g/s [33], and 0.89 kW at 0.407 g/s [34]. Direct comparison is difficult because those systems operated with vehicle HVAC airflow, different evaporator geometries and different definitions of the cooling effect. In the present system, the main reason for the lower value is that much of the LPG left the evaporator still liquid; improving evaporation inside the cabinet is therefore the most direct way to increase the useful cooling.

4. Conclusion

This study experimentally assessed whether the refrigeration effect of LPG expansion, used in a half-cycle system before combustion, can cool a 192 L refrigerated display cabinet. At an average LPG flow rate of 3.07 kg/h, the air in the empty, closed cabinet cooled from 26.3 °C to 4.7 °C (a reduction of 21.6 °C) in 60 minutes, and its humidity ratio fell from 15.7 to 3.5 g/kg dry air (≈ 78%), showing continuous dehumidification. Mixture property calculations showed that the LPG left the evaporator as a two-phase mixture (vapour quality ≈ 0.4–0.6), so the temperature rise along the evaporator was temperature glide, not superheat. The estimated refrigerant-side cooling rate was about 80 W (≈ 50–155 W), well below the value obtained if complete evaporation is assumed, and only a small part of the absorbed heat appeared as a change in the energy of the cabinet air. The final air temperature fell within the range of product surface temperatures measured in a commercial minimarket RDC (5.4–8.3 °C), but this is an illustrative comparison, not evidence of equivalent performance. Overall, this single 60-minute test demonstrates preliminary cooling feasibility in the tested cabinet. Practical use as an auxiliary cooling source will require more complete evaporation inside the cabinet with safe handling of the outlet stream, repeated tests to establish repeatability, tests with loaded cabinets and longer operation, a full cabinet energy balance, and an assessment of how often LPG consumption coincides with cooling demand in real kitchens and food businesses.

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References

[1] M. S. Stephen, S. A. Valsalan, and H. N. Ramanathan, "Energy consumption in seafood freezing: An analysis on production capacity and freezer type," *Next Energy*, vol. 12, p. 100620, 2026, doi: 10.1016/j.nxener.2026.100620.

[2] S. Bobbo, G. Lombardo, D. Menegazzo, L. Vallese, and L. Fedele, "A Technological Update on Heat Pumps for Industrial Applications," 2024. doi: 10.3390/en17194942.

[3] A. Mota-Babiloni and P. Makhnatch, "Predictions of European refrigerants place on the market following F-gas regulation restrictions," *International Journal of Refrigeration*, vol. 127, pp. 101–110, 2021, doi: 10.1016/j.ijrefrig.2021.03.005.

[4] O. A. A.-M. Ibrahim *et al.*, "Hydrocarbon refrigerants as sustainable alternatives to high-GWP refrigerants: a systematic review of R600 and R600a," *Journal of Thermal Analysis and Calorimetry*, vol. 150, no. 21, pp. 17185–17206, 2025, doi: 10.1007/s10973-025-14831-3.

[5] O. A. A.-M. Ibrahim, S. A. Kadhim, K. A. Hammoodi, F. L. Rashid, and A. H. Askar, "Review of hydrocarbon refrigerants as drop-in alternatives to high-GWP refrigerants in VCR systems: The case of R290," *Cleaner Engineering and Technology*, vol. 23, p. 100825, 2024, doi: 10.1016/j.clet.2024.100825.

[6] D. Wu, B. Hu, and R. Z. Wang, "Vapor compression heat pumps with pure Low-GWP refrigerants," *Renewable and Sustainable Energy Reviews*, vol. 138, p. 110571, 2021, doi: 10.1016/j.rser.2020.110571.

[7] F. Alsouda, N. S. Bennett, S. C. Saha, F. Salehi, and M. S. Islam, "Vapor Compression Cycle: A State-of-the-Art Review on Cycle Improvements, Water and Other Natural Refrigerants," 2023. doi: 10.3390/cleantechnol5020030.

[8] R. Yıldırım, A. Akyüz, K. Kumaş , A. D. Tuncer, and A. Gungor, "A comprehensive overview of refrigerants from the past to the modern day: energy, exergy, and environmental perspectives," *Journal of Thermal Analysis and Calorimetry*, vol. 150, no. 5, pp. 3201–3223, 2025, doi: 10.1007/s10973-025-14050-w.

[9] G. H. Kang, S.-I. Na, and M. S. Kim, "Reasonable performance comparison for a refrigeration system using different refrigerants: Case with propane and isobutane mixtures with several compositions," *International Journal of Refrigeration*, vol. 135, pp. 41–50, 2022, doi: 10.1016/j.ijrefrig.2021.12.006.

[10] M. Ozsipahi, H. A. Kose, H. Kerpicci, and H. Gunes, "Experimental study of R290/R600a mixtures in vapor compression refrigeration system," *International Journal of Refrigeration*, vol. 133, pp. 247–258, 2022, doi: 10.1016/j.ijrefrig.2021.10.004.

[11] D. Cai, Z. Hao, H. Xu, and G. He, "Research on flammability of R290/R134a, R600a/R134a and R600a/R290 refrigerant mixtures," *International Journal of Refrigeration*, vol. 137, pp. 53–61, 2022, doi: 10.1016/j.ijrefrig.2022.02.015.

[12] M. Wei *et al.*, "Benefits and challenges in deployment of low global warming potential R290 refrigerant for room air conditioning equipment in California," *Sustainable Energy Technologies and Assessments*, vol. 70, p. 103937, 2024, doi: 10.1016/j.seta.2024.103937.

[13] L. Vaitkus, L. Prakopavič ius, and A. Balč ius, "Refrigerants for eutectic refrigerating systems–Experimental results and future consideration," *International Journal of Refrigeration*, vol. 164, pp. 105–116, 2024, doi: 10.1016/j.ijrefrig.2024.04.012.

[14] A. M. Ashour, S. A. Kadhim, H. M. Jaffar, O. A. A.-M. Ibrahim, A. Mota-Babiloni, and A. Bouabidi, "Experimental investigation of the performance of vapor compression cycle in a water cooler system with low-GWP alternative refrigerants," *Thermal Science and Engineering Progress*, vol. 62, p. 103611, 2025, doi: 10.1016/j.tsep.2025.103611.

[15] Z. Hou, Z. Yang, H. He, Y. Zhao, and S. Zhang, "Experimental study on the thermal performance of the isobutane in lubricants for refrigeration and air conditioning applications," *Applied Thermal Engineering*, vol. 302, p. 131956, 2026, doi: 10.1016/j.applthermaleng.2026.131956.

[16] D. Calleja-Anta, D. Sánchez, L. Nebot-Andrés, R. Cabello, and R. Llopis, "Alternative mixtures to R-600a. Theoretical assessment and experimental energy evaluation of binary mixtures in a commercial cooler," *International Journal of Refrigeration*, vol. 152, pp. 83–92, 2023, doi: 10.1016/j.ijrefrig.2023.05.009.

[17] S. A. Kadhim, "Thermodynamic and environmental analysis of hydrocarbon refrigerants as alternatives to R134a in domestic refrigerator," *International Journal of Thermodynamics*, vol. 27, no. 2, pp. 10–18, 2024, doi: 10.5541/ijot.1368985.

[18] S. G. Hacıpaş aoğ lu and İ . T. Öztürk, "Methods to increase the coefficient of performance of refrigeration cycles in low-temperature applications : a review," *Engineering Research Express*, vol. 7, 2025, doi: 10.1088/2631-8695/adc875.

[19] B. Ş ensoy *et al.*, "Experimental analysis of energy-saving and eco-friendly refrigeration system design with condensate water management," *Applied Thermal Engineering*, vol. 271, p. 126315, 2025, doi: 10.1016/j.applthermaleng.2025.126315.

[20] A. Mota-Babiloni, J. Navarro-Esbrí, Á. Barragán-Cervera, F. Molés, B. Peris, and G. Verdú, "Commercial refrigeration – An overview of current status," *International Journal of Refrigeration*, vol. 57, pp. 186–196, 2015, doi: 10.1016/j.ijrefrig.2015.04.013.

[21] H. Demirpolat, "The energy efficiency and environmental analysis of open-type commercial display cabinet with a multi-flow air curtain design," *Sustainable Energy Technologies and Assessments*, vol. 60, p. 103555, 2023, doi: 10.1016/j.seta.2023.103555.

[22] Y. Wang, Y. Zheng, X. Hu, S. Liu, X. Li, and Q. Wang, "Impact of condensate water recycling on the performance of open refrigerated display cabinet," *Case Studies in Thermal Engineering*, vol. 54, p. 104022, 2024, doi: 10.1016/j.csite.2024.104022.

[23] T. Vengalis and V. Mokšin, "Experimental and Numerical Study of the Performance of an Open-Type Multi-Deck Refrigerated Cabinet with Single and Dual Air Curtain," 2023. doi: 10.3390/app13169080.

[24] I. N. Suamir, M. E. Arsana, and K. M. Tsamos, "Experimental study on the influences of air flow in an integral hydrocarbon display cabinet to its temperature and energy performances," in *IOP Conference Series: Materials Science and Engineering*, IOP Publishing, 2019, p. 12017. doi: 10.1088/1757-899X/494/1/012017.

[25] C. Piselli, C. Balocco, S. Forastiere, A. Silei, F. Sciurpi, and F. Cotana, "Energy efficiency in the commercial sector. Thermodynamics fundamentals for the energy transition," *Energy Reports*, vol. 11, pp. 4601–4621, 2024, doi: 10.1016/j.egyr.2024.04.033.

[26] R. A. Kishore *et al.*, "Experimental characterization and analysis of phase change material-based thermal energy storage system for refrigerated display case," *Energy Conversion and Management*, vol. 353, p. 121143, 2026, doi: 10.1016/j.enconman.2026.121143.

[27] Direktorat Jenderal Minyak dan Gas Bumi, "Spesifikasi Bahan Bakar Gas Jenis Liquified Petroleum Gas No. 26525.K/10/DJM.T/2009," 2009, *Kementerian Energi dan Sumber Daya Mineral Republik Indonesia, Jakarta*.

[28] R. A. Fauzi, "Pendeteksi kebocoran gas menggunakan sensor MQ-2 berbasis Arduino Uno," *Jurnal Teknik Informatika*, vol. 12, no. 2, pp. 101–110, 2024.

[29] M. Setiyo, S. Soeparman, and N. Hamidi, "Numerical Study on Cooling Effect Potential from Vaporizer Device of LPG Vehicle," *Journal of Engineering Science and Technology*, vol. 12, no. 7, pp. 1766–1779, 2017.

[30] M. Setiyo, N. Widodo, B. C. Purnomo, S. Munahar, M. A. Rahmawan, and A. Luthfi, "Harvesting cooling effect on lpg-fueled vehicles for mini cooler: A lab-scale investigation," *Indonesian Journal of Science and Technology*, 2019, doi: 10.17509/ijost.v4i1.12834.

[31] M. Setiyo, B. C. Purnomo, B. Waluyo, D. R. B. Syaka, and N. Hamidi, "Refrigeration effect and energy efficiency ratio (EER) calculation of 1/2 cycle refrigeration system on LPG-fueled vehicles," in *IOP Conf. Series: Materials Science and Engineering*, IOP, 2018. doi: 10.1088/1757-899X/403/1/012087.

[32] M. Setiyo, B. Waluyo, and N. Hamidi, "Analysis of Evaporator Effectiveness on 1/2 Cycle Refrigeration Systems: A Case Study on LPG Fueled Vehicles," *Jurnal Teknologi*, vol. 82, no. 3, pp. 53–59, 2020, doi: 10.11113/jt.v82.13386.

[33] M. Setiyo *et al.*, "Cooling power characteristics of half-cycle refrigeration system in LPG fuelled vehicles by auxiliary chiller as heat exchanger," *Thermal Science and Engineering Progress*, vol. 27, p. 101145, 2022, doi: 10.1016/j.tsep.2021.101145.

[34] B. Waluyo *et al.*, "Cooling effect characteristic of the novel half-cycle refrigeration system on a liquefied petroleum gas (LPG) fueled vehicle," *Thermal Science and Engineering Progress*, vol. 34, p. 101405, 2022, doi: 10.1016/j.tsep.2022.101405.

[35] P. N. Ryastama, D. S. Prakasa, M. L. Rochman, and M. Setiyo, "Eksperimen Kinerja Sistem Refrigerasi Setengah Siklus pada Kendaraan Berbahan Bakar LPG," *Jurnal Rekayasa Mesin*, vol. 14, no. 1, pp. 181–194, 2023, doi: 10.21776/jrm.v14i1.1197.

[36] M. Mohan, "Zero Cost Refrigeration and Air Conditioning Using LPG," 2013, *Tech Briefs, Chennai*. [Online]. Available: http://contest.techbriefs.com/2013/entries/sustainable-technologies/3792

[37] I. H. Shah and K. Gupta, "Design of LPG Refrigeration System and Comparative Energy Analysis with Domestic Refrigerator," *International Journal Of Engineering Sciences & Research Technology*, vol. 3, no. 7, pp. 206–213, 2014, [Online]. Available: http://www.ijedr.org/papers/IJEDR1502237.pdf

[38] P. Srinivas, P. R. Chandra, M. R. Kumar, and N. Y. M. Reddy, "Experimental Investigation of LPG as Refrigerant in a Domestic Refrigerator," *Journal of Mechanical Engineering Research and Technology*, vol. 2, no. 1, pp. 470–476, 2014.

[39] U. Shahare, K. Yadav, V. Dongre, A. Umre, K. Gajbhiye, and Y. Bhaisare, "Fabrication of Low Cost Refrigeration System by using LPG," *International Journal of Advanced Research in Science, Communication and Technology*, vol. 8, no. 1, pp. 282–286, 2021, doi: 10.48175/IJARSCT-1846.

[40] G. Ali, A. Muzaffar, T. A. Cheema, and S. F. Shah, "Refrigeration Potential Investigation of Half-Cycle Refrigeration System Using Liquefied Petroleum Gas †," *Engineering Proceedings*, vol. 45, no. 1, 2023, doi: 10.3390/engproc2023045044.

[41] M. Manohar, P. K. Sahu, P. Sahu, D. Singh, and A. Chandra, "Design Analysis and Performance of Low Cost Refrigeration System using LPG," *International Journal of Engineering Science and Computing*, vol. 10, no. 5, pp. 25633–25636, 2020.

[42] S. J. Kline, "Describing uncertainties in single-sample experiments," *Mech. Eng.*, vol. 75, pp. 3–8, 1963.

[43] D.-Y. Peng and D. B. Robinson, "A New Two-Constant Equation of State," *Industrial & Engineering Chemistry Fundamentals*, vol. 15, no. 1, pp. 59–64, Feb. 1976, doi: 10.1021/i160057a011.

[44] B. E. Poling, J. M. Prausnitz, O. John Paul, and R. C. Reid, *The properties of gases and liquids*, vol. 5. Mcgraw-hill, New York, 2001.

[45] ASHRAE, *ASHRAE Handbook—Fundamentals, Chapter 1: Psychrometrics*. Atlanta: American Society of Heating, Refrigerating and Air-Conditioning Engineers, 2021.

[46] A. L. Buck, "New equations for computing vapor pressure and enhancement factor," *Journal of Applied Meteorology and Climatology*, vol. 20, no. 12, pp. 1527–1532, 1981, doi: 10.1175/1520-0450(1981)020<1527:NEFCVP>2.0.CO;2.

[47] Y. A. Çengel and M. A. Boles, *Thermodynamics: An engineering approach*, Eight. McGraw-Hill, 2015.