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Sustainable refrigeration and HVAC systems with natural fluids: An energy efficiency case Study

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Abstract

This study explores the integration of refrigeration and HVAC systems utilizing natural refrigerants, ammonia (NH3, R717) and propane (R290), aiming to improve energy efficiency and minimize environmental impact. A case study was conducted at a commercial facility in Luanda, Angola, where the original refrigeration system was replaced and integrated with the HVAC system. The proposed system showed significant performance improvements and compliance with F-GAS regulations. The Coefficient of Performance (COP) increased from 2.29 to 3.53 in summer and from 2.21 to 3.12 in winter, reflecting substantial energy efficiency gains. Annual energy consumption decreased by 548,225 kWh (33.8%), leading to a corresponding reduction in CO_2 emissions by 53,726 kg (33.8%). These reductions align with global climate commitments such as the Kyoto Protocol and the Paris Agreement, demonstrating the potential of integrated systems to mitigate climate change. The system exhibited strong adaptability to climatic conditions, with efficient operation at a COP of 3.53 in summer and 3.12 in winter. The integration of refrigeration and HVAC systems with natural refrigerants not only enhances energy efficiency and reduces CO_2 emissions but also provides long-term economic benefits through operational cost reductions. The study emphasizes the importance of system integration as a sustainable solution in the refrigeration and HVAC industries, contributing to the advancement of energy-efficient technologies and supporting global climate mitigation goals.

Keywords: Ammonia, COP, Energy Performance Optimization, HVAC, Industrial Refrigeration, Propane.

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1. Introduction

Energy efficiency and sustainability have emerged as priority topics in nearly all sectors of the economy. In the context of refrigeration and air conditioning (HVAC), the need to develop systems that combine low energy consumption, reduced environmental impact, and economic viability has become a priority, particularly in medium- and large-sized buildings [1, 2]. Among the promising technological solutions, integrated systems stand out, which combine refrigeration, air conditioning, and domestic hot water production (DHW) functions. These systems have demonstrated significant potential for improving energy efficiency, especially when combined with the use of natural refrigerants [3, 4]. Carbon dioxide (CO_2), for example, has been one of the most widely adopted solutions in recent decades, particularly in food retail applications across Europe. Between 2008 and 2020, the number of transcritical CO_2 installations in the region increased from 140 to 29,000, representing an impressive growth of 81% between 2018 and 2020 [4]. This increase was driven by environmental policies such as the 2014 Regulation on Fluorinated Gases [5] which significantly restricted the use of hydrofluorocarbons (HFCs). The recent revision of this regulation [6] aims for even more aggressive reductions in HFC use, establishing targets for a 95% reduction by 2030 and complete elimination by 2050. These changes impact various applications, including commercial and industrial refrigeration, heat pumps, and air conditioning systems.

In this context, research has focused on the development of technical solutions that utilize natural refrigerants and deliver performance equivalent to or exceeding that of HFC-based systems. The objective is to mitigate direct greenhouse gas emissions while avoiding an increase in indirect emissions resulting from higher energy consumption. Among natural refrigerants, ammonia (NH_3, R717), propane (R290), and carbon dioxide (CO_2) stand out for their sustainable characteristics and high energy efficiency. However, their implementation still faces technical and regulatory challenges, such as safety concerns and maximum load limitations in systems [7, 8]. Integrated systems have been identified in the current literature as an efficient and promising approach for optimizing thermal systems in buildings and industrial facilities [9, 10]. These systems enable the recovery of thermal energy that would otherwise be wasted, promoting higher energy efficiency and reducing environmental impact [11]. Technologies such as multi-stage compressors, electronic control valves, thermal inertia tanks, and heat exchangers play a crucial role in maximizing the performance of these systems [12]. Despite the advances, integrated systems still face significant limitations, with economic feasibility being one of the main barriers to their widespread adoption. The high initial cost, coupled with the need for specialized maintenance, reduces accessibility for smaller installations, limiting their application primarily to commercial and service buildings with complex thermal needs [13]. Nevertheless, several gaps remain in the literature, such as challenges related to the use of natural refrigerants and the need to optimize designs to reduce installation costs. Furthermore, there is an increasing demand for innovative solutions that allow greater operational flexibility and lower environmental impact. This paper contributes to the field by proposing a practical example of a solution that improves the efficiency of an existing system by optimizing it, presenting design and operational strategies that maximize the energy efficiency of an integrated system while simultaneously addressing regulatory and economic challenges. The research includes new approaches for the integration of natural refrigerants, with an emphasis on recent advancements in safety and technical performance. Finally, this study provides a comprehensive overview of the contribution that integrated systems can offer and proposes practical guidelines that may promote their large-scale adoption, driving the transition towards more efficient and sustainable solutions in the refrigeration and air conditioning sector.

2. Objective

This article aims to analyze the implementation of an integrated refrigeration and HVAC (Heating, Ventilation, and Air Conditioning) system in commercial facilities, utilizing natural fluids, namely ammonia (R717) and propane (R290). The analysis will be conducted through a case study, which seeks to resize the existing refrigeration system and integrate it with the HVAC system, with the goal of improving the energy performance of the facility and ensuring compliance with the latest environmental regulations. It is expected that the adoption of the proposed solutions will significantly contribute to the overall performance improvement of the facility, with particular emphasis on optimizing energy consumption and enhancing thermal efficiency. The case study will address a commercial facility, both wholesale and retail, with the objective of upgrading the refrigeration system in the commercial area, seeking an integrated solution that combines the functions of refrigeration and air conditioning. In this context, the work proposes resizing the refrigeration system to allow its integration with the HVAC system, aiming to achieve better energy performance and promote synergy between the two systems. The project was developed with a focus on improving the thermal efficiency of the facility, prioritizing a sustainable approach that favors the use of natural fluids with low environmental impact, in line with the best sustainability practices currently recognized. Thus, this study aims to contribute to the advancement of sustainable technologies in commercial installations, promoting the integration of energy-efficient and environmentally responsible solutions, aligned with the objectives of reducing environmental impact and improving energy efficiency.

3. Materials and Methods

3.1. The Commercial Plant

The facility under analysis in this study refers to a large commercial area designed for wholesale and retail trade, located in Luanda, Angola, with the layout depicted in Figure 1. The building is surrounded by exterior walls with a thickness of 200 mm, consisting of 150 mm of common brick, coated with 25 mm of plaster on both sides, both on the exterior and interior. The interior space is divided into several functional areas, namely: the commercial area, dedicated to the display of products for sale; the warehouse area; the cold storage rooms and cutting zones; and the technical area. The internal partitions consist of 150 mm thick walls, constructed similarly to the exterior walls. The frozen food chambers are insulated with polypropylene sandwich panels, with a thickness of 200 mm on the exterior walls and 140 mm on the walls adjacent to the refrigerated spaces. The refrigerated areas, in turn, are insulated with 100 mm thick sandwich panels applied all around. Both the frozen food chambers and the refrigerated spaces are covered with 200 mm thick sandwich panels. The commercial area spans 4,200 m^2 and features a ceiling height of 9 meters, housing the marketed products, checkout counters, and a support office. The storage area, intended for non-perishable product storage (cream zone in the layout), covers 400 m^2 and also has a ceiling height of 9 meters. Due to the nature of the products stored, which do not require temperature control, this area is solely ventilated. The refrigerated area (blue zone in the layout) consists of various temperature-controlled zones, including chambers with different temperature ranges. These zones vary in size and purpose, including spaces for food preparation, aisles for circulation or sale of fresh products, as well as two cold storage rooms for the storage of packaged products.

3.2. Refrigeration Power

The refrigeration power required to maintain the desired temperature conditions in the various refrigerated spaces was calculated based on the thermal loads, considering the different factors that influence this calculation. These factors include the thermal load from the building envelope, the thermal load from the stored products, the thermal load caused by air infiltration, and the internal thermal loads. Each of these components will be discussed in detail below [14].

3.2.1. Thermal Load from the Envelope

The thermal load from the envelope refers to the heat entering or leaving the refrigerated space through surfaces such as walls and doors. To calculate this thermal load, several parameters must be considered, including the surface area, the temperature difference between the external and internal environments, and the overall heat transfer coefficient of the structure [15]. These parameters are essential for determining the amount of heat that needs to be removed to maintain the desired temperature inside the cold rooms. The equation used to calculate the thermal load through the envelope is as follows:

$$\dot{Q}_{COND} = U \cdot A \cdot \Delta T \ [kW] \tag{1}$$

Where:
U - Overall heat transmission coefficient [W/(m^2·ºC)];
ΔT - Temperature variation [℃] between the outside and inside environment;
A - Area of the envelope perpendicular to the imposed heat flow [m^2].
The following Equation 2 will be used to calculate the overall heat transmission coefficient. For convection, an external convection coefficient (h_{ext}) of 8.1 W/m^2 ºC and an internal convection coefficient (h_{int}) of 8 W/m^2 ºC will be considered, which corresponds to the convection value of air without velocity; for conduction, the thermal conductivity of the panel will be applied and distributed over its thicknes [14-16].

$$U = \frac{1}{R_t} = h_{ext} + \frac{k}{e} + h_{int} \tag{2}$$

Where:
R_t - Thermal resistance of the wall [m^2 ºC/W];
k - Thermal conductivity of the material [W/m-K];
e - Insulation thickness [m].

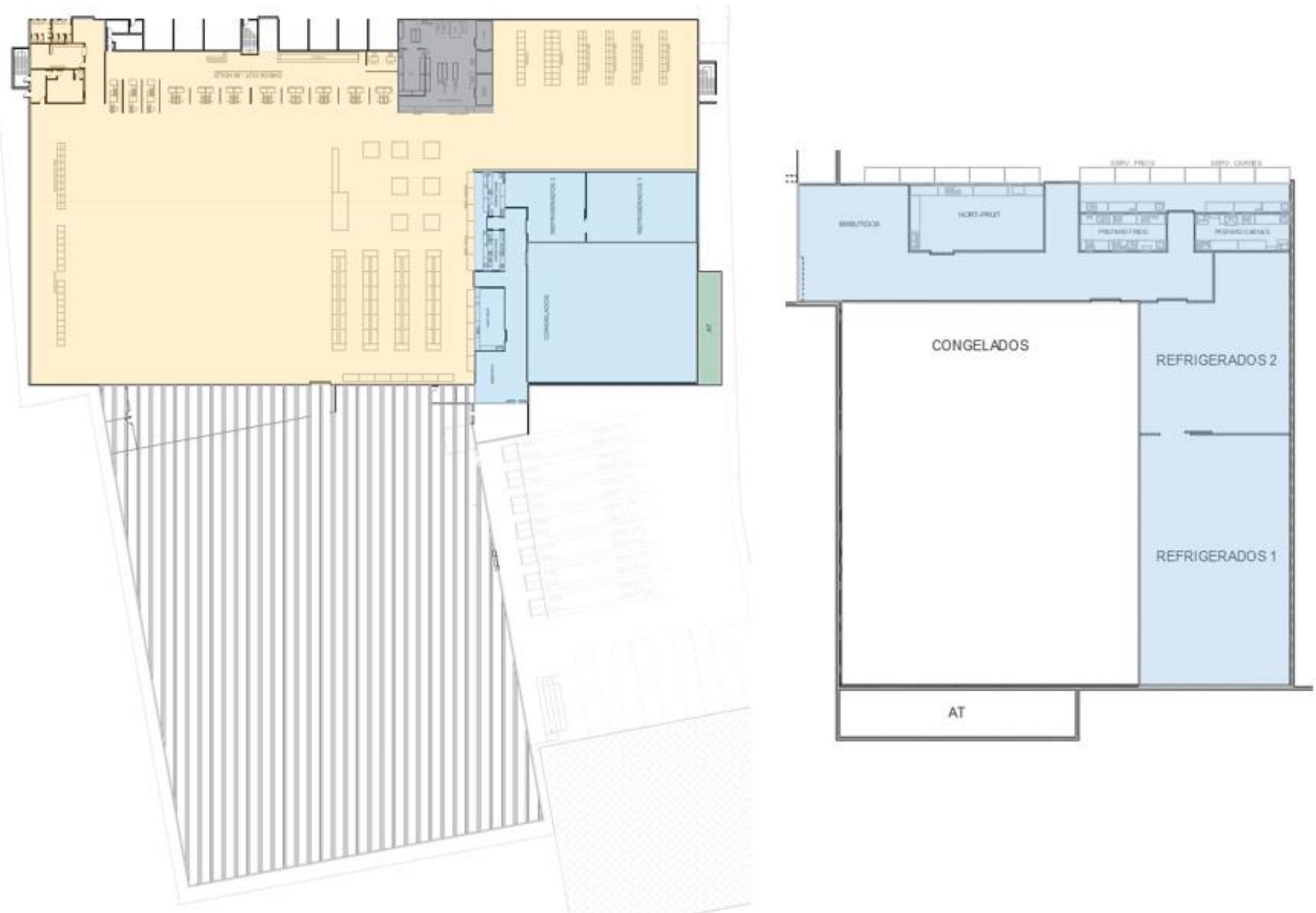

Figure 1.
Building layout.

3.2.2. Thermal Load from the Product

The thermal load from the product considers the amount of heat that must be removed to achieve and maintain the desired conservation temperature of the stored product. The equation for calculating the thermal load from the product is given by Equation 3 which considers the initial temperature of the product, the desired storage temperature, the quantity of product entering the chamber, and the specific heat of the product [14, 15]. This calculation is crucial for determining the amount of energy to be removed to achieve the required storage temperature.

$$\dot{Q}_{ARREF} = C_{pa} \times (T_i - T_a) \times \dot{m}\ [kW] \quad (3)$$

Where:
C_{pa} - Specific heat of the product [kJ/kg];
T_i - Product inlet temperature [ºC];
T_a - Product storage temperature [ºC];
$\dot{m}$ - Mass flow of product entering per day [kg/24h].

3.2.3. Thermal Load from Infiltration

Air infiltration into the refrigerated spaces occurs when external air enters the chamber. This typically happens when the cold room door is opened, when the door is not properly sealed, or when there are gaps that allow air to enter. The difference in temperature and humidity between the external air and the internal environment of the cold room results in an increase in both sensible and latent heat. The additional thermal load due to infiltration can be calculated using Equation 4 which considers the air renewal rate and the amount of heat introduced by the infiltrated air [17, 18].

$$\dot{Q}_{INF} = \dot{V}_{RENOV} \cdot E_{AR}\ [kW] \quad (4)$$

Where:
$\dot{V}_{RENOV}$ - Air renewal flow rate (=NR x Vol.Cam) [m^3/h];
E_{AR} - Amount of heat due to infiltrated air [kcal/m^3].

3.2.4. Internal Thermal Load

Internal thermal loads are generated by sources of heat within the refrigerated space, such as lighting, occupancy, and electrical equipment. The Equation 5 is used to calculate the internal thermal load, considering the number of hours of occupancy, the number of occupants, the heat dissipated by each occupant, the power of electrical equipment, and the lighting power. This thermal load is important for correctly sizing the refrigeration capacity since equipment and occupants generate additional heat that must be removed to maintain the desired temperature [3, 14, 17].

$$\dot{Q}_{INT} = \dot{Q}_{OCUP} \cdot \dot{Q}_{ILUM} \cdot \dot{Q}_{EQUIP} = \left(\frac{NHO}{24}\right)(NO \cdot E_{OCUP} + A \cdot E_{ILUM} + NE \cdot P_{MOT} \cdot ET_{MOT})[kW] \quad (5)$$

Where:
$\dot{Q}_{INT}$ - Internal thermal load [kW];
$\dot{Q}_{OCUP}$ - Thermal load due to occupancy [kW];
$\dot{Q}_{ILUM}$ - Thermal load due to lighting [kW];
$\dot{Q}_{EQUIP}$ - Internal load due to equipment [kW];
NHO - Number of hours of occupancy;
NO - Number of occupants;
E_{OCUP} - Heat dissipated by each occupant [kW];
NE - Number of devices;
P_{MOT} - Power of the equipment's electric motors.
ET_{MOT} - Thermal equivalent of the electric motor;
E_{ILUM} - Lighting power [kW].

3.2.5. Calculation of Total Refrigeration Power

Finally, the total refrigeration power required to maintain the desired thermal conditions in the various refrigerated spaces is calculated by considering the sum of all individual thermal loads, multiplied by a safety coefficient (CS) [3, 14, 15, 17, 18]. This coefficient ensures that the chosen equipment has a sufficient margin to accommodate unexpected factors or variations that may arise during system operation, such as changes in environmental conditions or energy consumption behaviour. The final equation for calculating the refrigeration power is as follows:

$$\dot{Q}_{FRIG} = (24/TFE)\left(\dot{Q}_{COND} + \dot{Q}_{ARREF} + \dot{Q}_{INF} + \dot{Q}_{INT}\right) \times cs + \dot{Q}_{Equip} \ [kW] \quad (6)$$

Where:
TFE is the evaporator operating time [h];
$\dot{Q}_{Equip}$ is the thermal load associated with refrigeration equipment [kW].

3.3. Existing Systems in the Original Installation

The original refrigeration installation comprises a configuration consisting of two independent systems, both designed to operate with the refrigerant R404A. The first system is dedicated to the preservation of frozen food, maintaining extremely low temperatures suitable for this purpose. In contrast, the second system is responsible for maintaining refrigerated conditions in the remaining chambers, ensuring temperatures tailored to the specific requirements of each space. This configuration enables efficient and adaptive refrigeration across various areas of the facility, with the thermal parameters detailed in Table 1. Additionally, the building is equipped with an original air conditioning system that employs Air Handling Units (AHUs) to process and condition fresh air within the commercial space. These units play a critical role in ensuring indoor air quality by supplying the required volume of fresh air and rigorously controlling temperature, humidity, and ventilation. The treated air is uniformly distributed through a ductwork system, ensuring optimized environmental conditions across all functional areas of the commercial facility. Furthermore, the HVAC system includes individual units installed in the support offices, specifically designed to meet the unique climatic requirements of these areas. These units provide tailored adjustments to the thermal needs of the offices, which often differ from the demands of larger commercial spaces. This integrated and adaptive approach ensures that both commercial spaces and offices benefit from appropriate and comfortable environmental conditions, thereby maximizing the overall operational efficiency of the system.

Table 1.
Set points temperatures and refrigerating power.

Area	Temperature (ºC)	Refrigerating power (kW)
Refrigerated 1	+2	33.5
Refrigerated 2	+2	31.5
Corridor	+12	9.2
Smoked meats	+12	12.6
Fruit and vegetable preparation	+15	1.9
Cold cuts preparation	+15	4.8
Meat preparation	+15	1.9
Frozen food	-25	58.6

3.3.1. Existing Frozen Food Circuit

The refrigeration circuit dedicated to freezing, integrated into the original refrigeration installation, supplies the freezer chamber, which has a total area of 545.7 m^2 and is used for the storage of packaged frozen products. This system operates as a single-stage compression cycle, functioning under a thermal regime with an evaporation temperature of -30ºC and a condensation temperature of +45ºC, with a nominal refrigeration power of 58.64 kW. The configuration of the freezing system includes the key equipment listed in Table 2, which are essential to its operation.

Table 2.
Frozen Food Circuit Equipment.

Equipment	Quantity	Model/Brand	Power (kW)
Evaporator	4	DDC/E 8M2/30 / Centauro	14.7
Compressors	4	6G-30.2Y / Bitzer	9.4
Condenser	1	ACP/M 480/280T / Centauro	167.0

The simplified functional diagram of the frozen food circuit is shown in Figure 2.

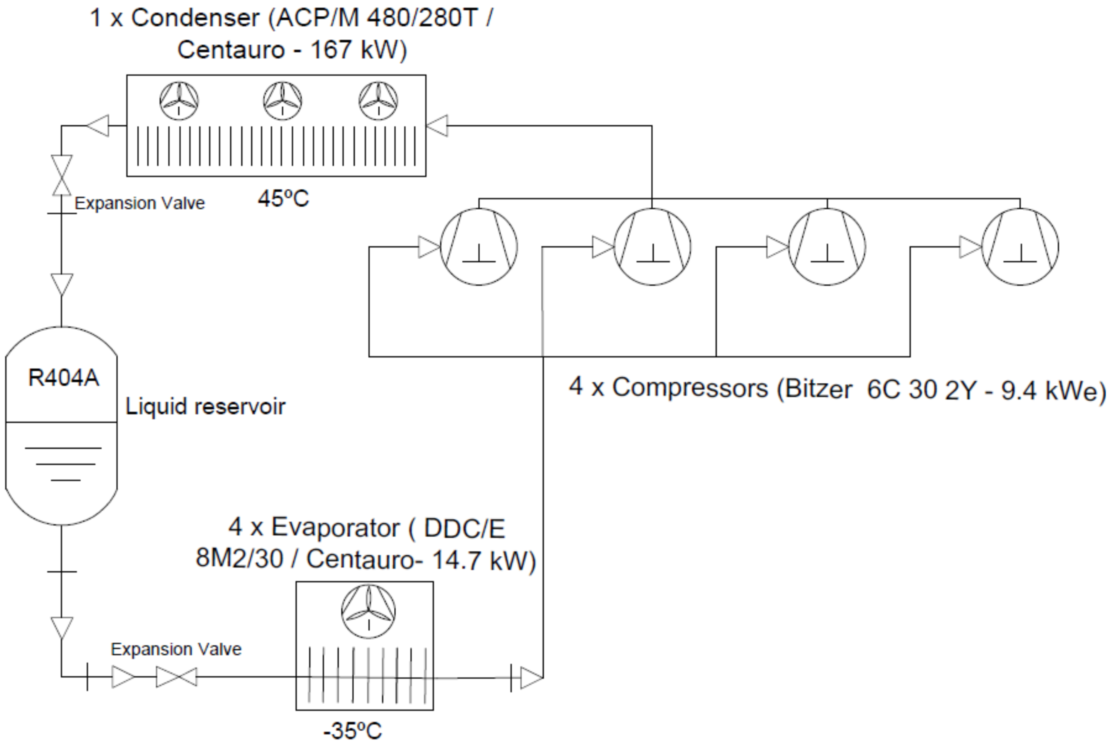

Figure 2.
Schematic diagram of the existing frozen food circuit.

3.3.2. Existing Refrigerated Food Circuit

The original chiller system is also composed of a single-stage compression cycle, operating with an evaporation temperature of -8ºC and a condensation temperature of 35ºC, as illustrated in the Mollier diagram shown in Figure 5. The chiller system includes the following main equipment, summarized in Table 3.

Table 3.
Existing Refrigerated circuit equipment.

Equipment	Quantity	Model/Brand	Power (kW)
Evaporator	2	MTB/E 6P1/25 / Centauro	14.5
Evaporator	2	MTB/E 6P1/25 / Centauro	15.7
Evaporator	3	CBK 4B2/6R / Centauro	5.7
Evaporator	1	CBK 3F3/19R / Centauro	5.7
Evaporator	2	CBK 4B1/2R / Centauro	2.6
Compressors	4	4PCS-15.2Y / Bitzer	10.65
Condenser	1	ACP/M 380/226T / Centauro	120.0

The simplified schematic diagram of the existing refrigerated circuit is shown in Figure 3.

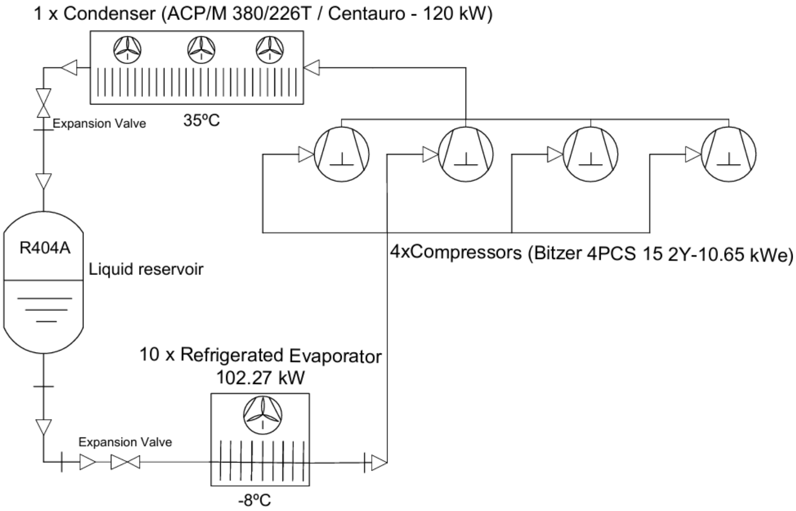

Figure 3.
Schematic diagram of the existing refrigerated circuit.

3.3.3. HVAC System

The HVAC (Heating, Ventilation, and Air Conditioning) system of this building consists of two chillers and a heat pump, designed to meet the thermal demands of the facility. The chillers, model YCWS0373SC Water Cooled Liquid Chiller Style B by Johnson Controls, have a combined thermal capacity of 510 kW and a Coefficient of Performance (COP) of 2.5, corresponding to an electrical power consumption of 204 kW.The installed heat pump, model LG Multi V III (2011), RUN160LTE4, has a thermal capacity of 50.4 kW and an electrical power consumption of 16.8 kW.

3.4. Integrated HVAC and Refrigeration System: Proposal and Analysis

3.4.1. Characterization of the Integrated System

The proposed integrated system combines the functions of heating, ventilation, and air conditioning (HVAC) with refrigeration, aiming to optimize energy performance and reduce operational costs. This integration leverages the use of two natural refrigerants: ammonia (NH_3) and propane (R290). The proposed approach capitalizes on the synergies between the two systems, fostering a sustainable and efficient utilization of thermal resources. For the HVAC system, the proposal includes recovering the thermal energy rejected by the refrigeration system, thereby eliminating the need for a dedicated chiller to produce chilled water. This approach significantly reduces initial investment and maintenance costs while optimizing the utilization of available thermal energy. The refrigeration system is designed as a two-stage cycle. The primary refrigerant, ammonia (NH_3), was selected for its superior thermodynamic properties and low environmental impact. To ensure operational safety, the system incorporates a secondary hydraulic circuit that utilizes chilled water to transfer cooling to nearby commercial areas, thereby minimizing the risk of direct exposure to NH_3 in high-traffic zones. Integrating these systems achieves improved energy efficiency and operational performance while ensuring compliance with environmental regulations and reducing long-term operational costs. Figure 4 presents the schematic diagram of the proposed new installation, illustrating the various components of the system.

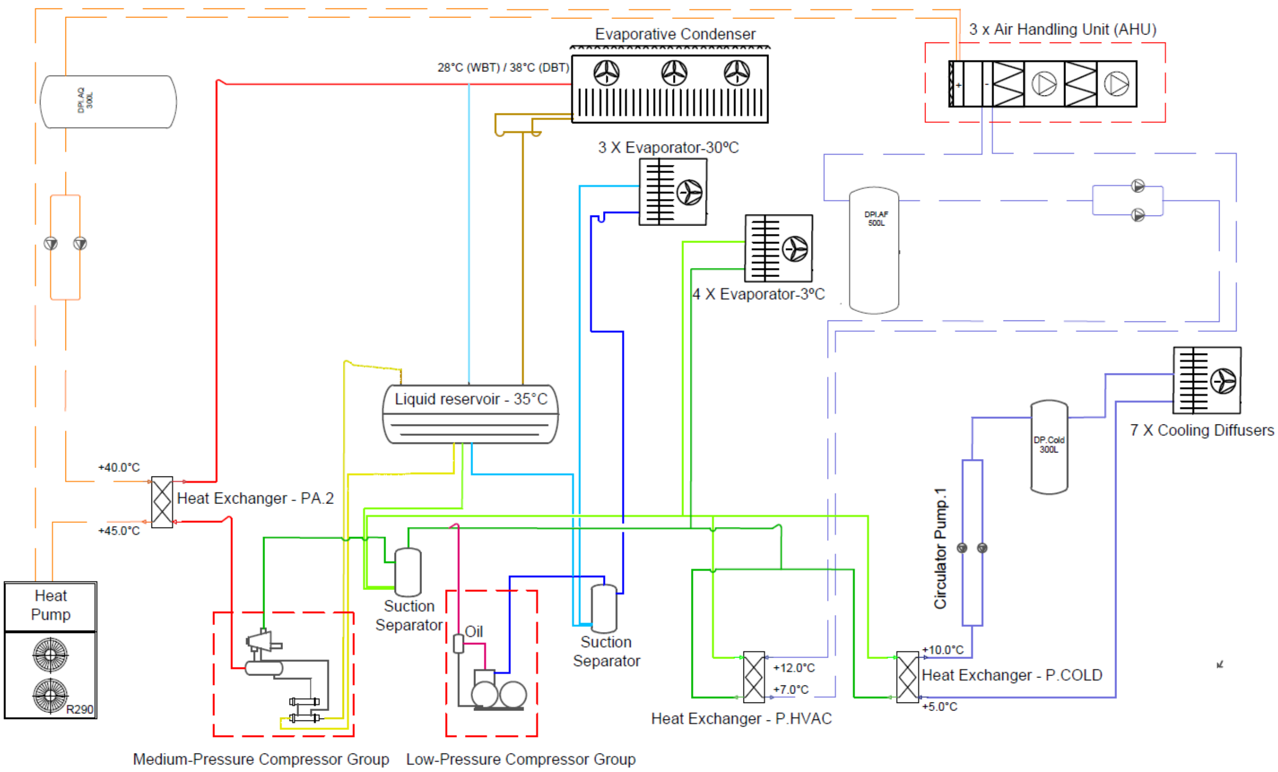

Figure 4.
Proposed Schematic diagram of the installation.

3.4.2. Technical Characterization of the Refrigeration System

The proposed refrigeration system operates on a two-stage ammonia (NH_3) cycle, with operating temperatures of -30ºC at the low stage, -3ºC at the intermediate stage, and 35ºC at condensation. This cycle, represented in the Mollier diagram (Figure 5), incorporates energy recovery strategies, including the pre-cooling of the fluid through a crossflow heat exchanger before entering the condenser. Additionally, the intermediate-pressure compressors are equipped with tubular heat exchangers to recover heat from the oil prior to cooling. Condensation occurs in an evaporative condenser, selected to account for the high ambient temperature conditions.

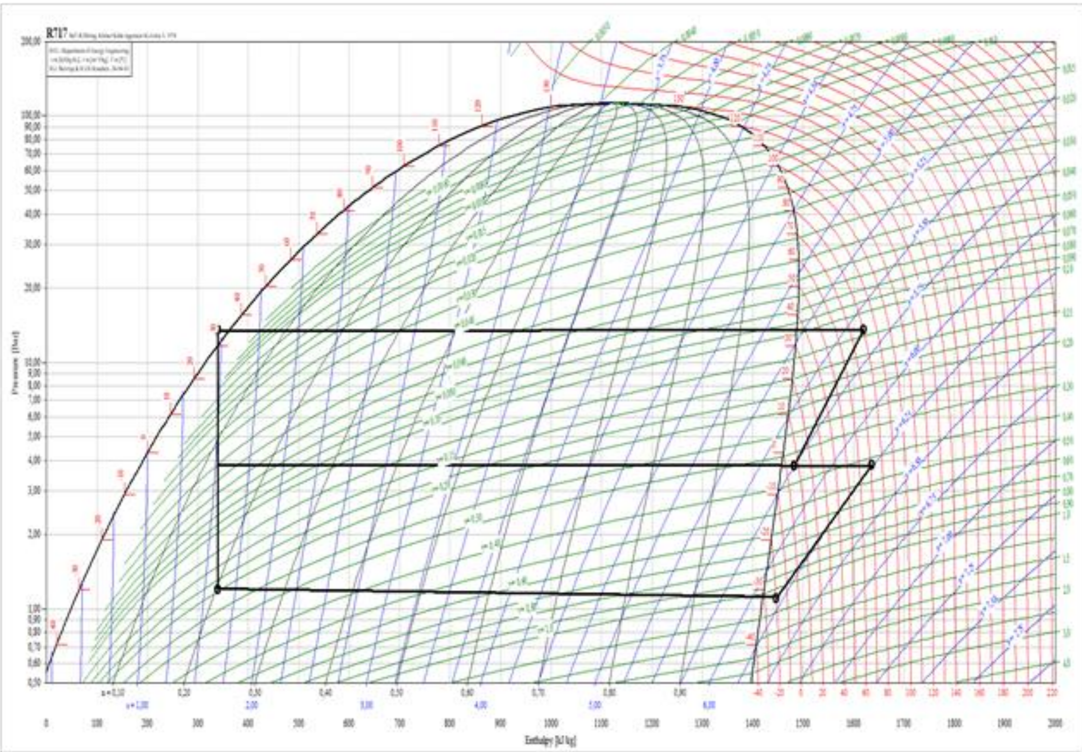

Figure 5.
Mollier diagram of the proposed refrigeration system.

The operation of the proposed system is organized into a structured process to ensure efficiency and safety. The liquid stored in the system's main tank at the condensation temperature (35ºC) can be directed to up to four distinct destinations, as depicted in Figure 6. In the low-temperature evaporators (-30ºC), ammonia (NH_3) undergoes evaporation and is routed to suction tank no. 1. This tank fulfils two essential functions: protecting the compressors from liquid shocks and subcooling the fluid before its entry into the evaporators. The evaporated fluid is subsequently compressed by the low-pressure compressors, elevating its pressure to the intermediate stage. At this stage, it combines with vapor from the medium-temperature evaporators and is directed to suction tank no. 2, which mirrors the functionality of suction tank no. 1. From suction tank no. 2, the fluid is compressed by the intermediate-pressure compressors to reach the condensation pressure (35ºC). Prior to entering the condenser, the fluid is routed through a plate heat exchanger, where it exchanges heat with water. Afterward, it is condensed and returned to the system's main tank. In the medium-temperature evaporators (-3ºC) and NH_3/water heat exchangers, the fluid follows a similar cycle, supplying a secondary hydraulic circuit designed to provide chilled water to air diffusers and refrigerated spaces. These spaces include corridors, food preparation areas, and other zones maintained at controlled temperatures ranging from 12ºC to 15ºC. The implementation of the secondary hydraulic circuit addresses the technical limitations of low-capacity evaporators in NH_3 systems, ensuring an efficient and uniform thermal distribution. Although NH_3 evaporates at -3ºC, the fluid within the hydraulic circuit returns at temperatures between 5ºC and 10ºC, depending on the thermal load. This integrated approach optimizes the system's energy efficiency and reliability, meeting both performance and environmental standards.

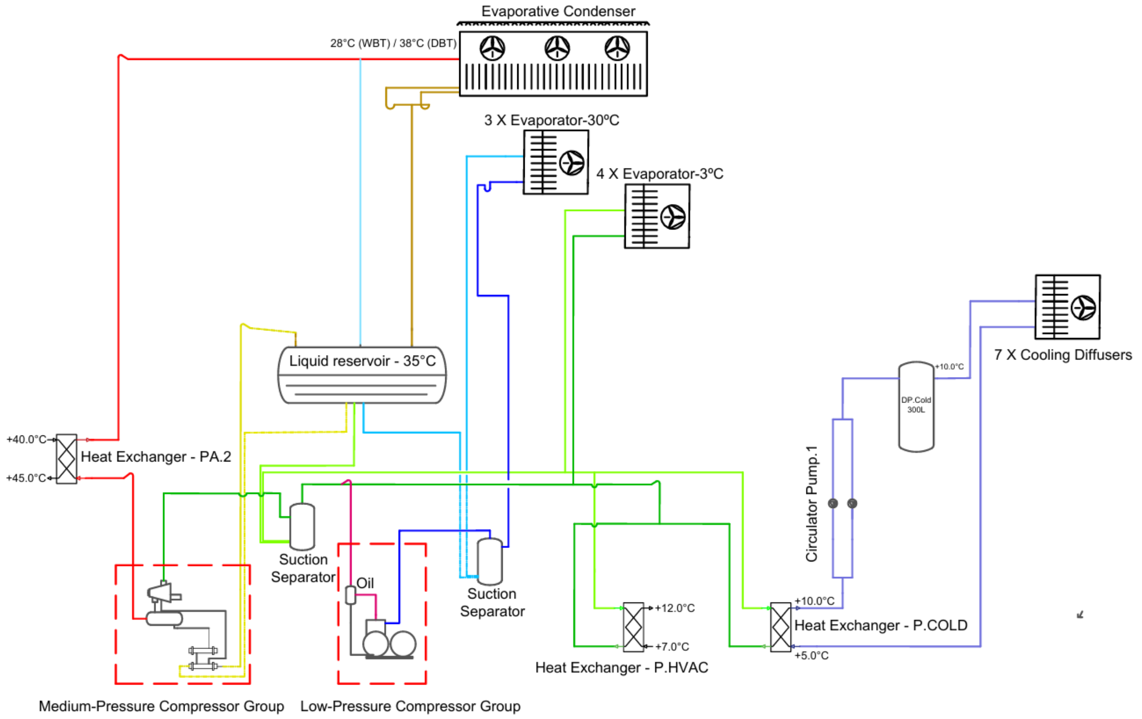

Figure 6.
Schematic diagram of the refrigeration system.

3.4.3. Proposed New Refrigeration Equipment

The selection of evaporators and air diffusers for the proposed installation was based on the thermal power requirements for each area. The refrigeration system was designed to optimize efficiency by dividing the compressors into two compression groups. Compression Group 1 is responsible for compressing the fluid from the low-temperature evaporators (-30ºC), while Compression Group 2 compresses the fluid from Group 1 along with the gas from the medium-temperature evaporators (-3ºC) to condensing pressure. The compressors selected for this installation include GEA Grasso V 300 model compressors in Group 1, with a total refrigeration capacity of 30 kW and the ability to scale power in 25% increments. In Group 2, Bitzer OSKA7472-K screw compressors were chosen, providing a total refrigeration capacity of 214 kW, with a shaft power requirement of 50.8 kW. These compressors require external oil cooling, which will be achieved through tubular heat exchangers. The oil will be cooled in two stages: first, by exchanging heat with the water from the HVAC heating system, and second, by using liquid NH_3 to reach the manufacturer-specified temperature of 35ºC.The evaporative condenser was specifically designed to operate efficiently even in high ambient temperature conditions, close to 35ºC, to ensure effective heat dissipation. The specifications of the selected evaporators and Frigo diffusers are summarized in Tables 4 and 5.

Table 4.
Proposed refrigerated evaporators.

Model	Quantity	Location
Centauro DLI 10L2/23	3	Refrigerated Chamber 1
Centauro CBBI 6P2/35	3	Refrigerated Chamber 2
Centauro BSUTI 12.187 M	3	Frozen Food Chamber

Table 5.
Frigodiffusers of refrigerated products proposed.

Model	Quantity	Location
Centauro CBK 3F2/13 R	2	Corridor
Centauro MTA 4M2/36	1	Smoked meats
Centauro CBK 4B2/6 R	1	Cold cuts
Centauro CBK 4B2/5	2	Vegetables
Centauro CBK 4B2/6 R	1	Meat

The overall layout of the proposed refrigeration system, including equipment placement, is shown in Figure 7.

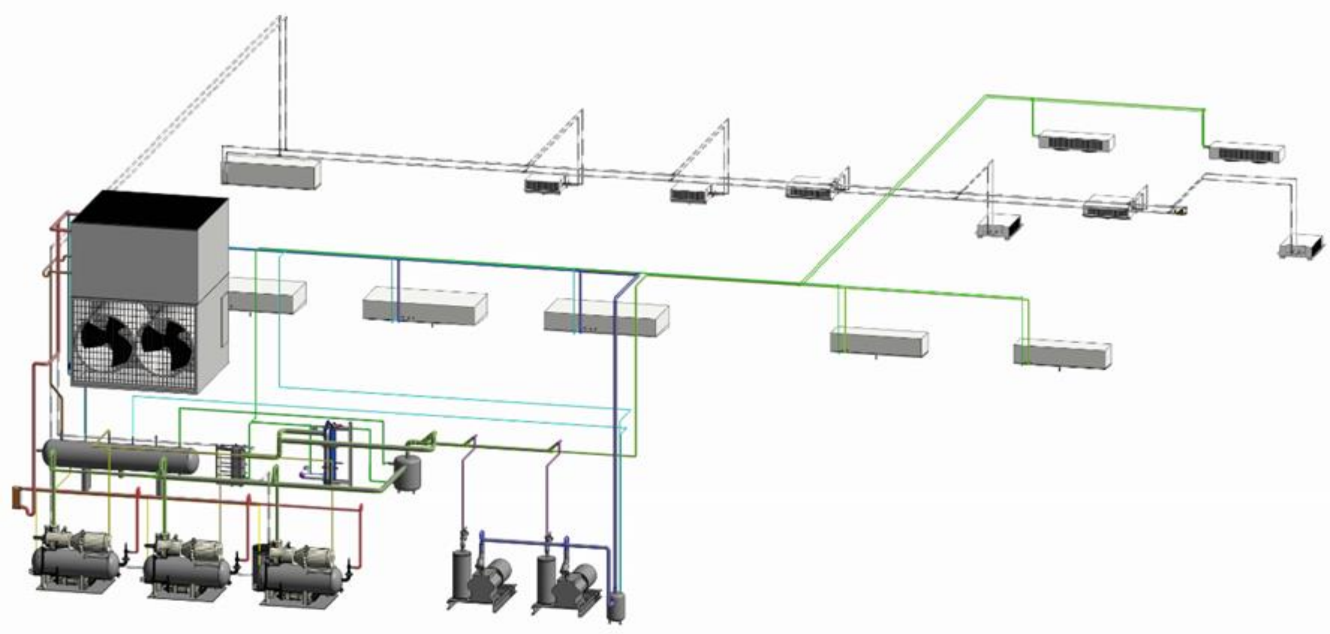

Figure 7.
Layout of proposed refrigeration system.

3.4.4. Proposed New HVAC Equipment

The proposed HVAC system for the facility includes, as key components, a heat pump for heating and Air Handling Units (AHUs) responsible for distributing conditioned air throughout the building. The function of the heat pump is to heat the water required to supply heat to the HVAC system whenever necessary. In selecting this equipment, in addition to the required power for operation, the fluid used was also taken into consideration, which in this case is propane (R290). This natural fluid was chosen due to its high efficiency and low environmental impact, making it a sustainable option for the heating system. Based on these criteria, the EMICON PAS Kap 651 unit was selected as the ideal solution for the installation. Regarding the Air Handling Units (AHUs), a configuration consisting of three identical units, model Geniox 27.13 from the manufacturer System air, was chosen, each with an air flow rate of 6,500 m^3/h, ensuring effective distribution of conditioned air throughout the facility. The general layout of the proposed new HVAC installation is shown in Figure 8.

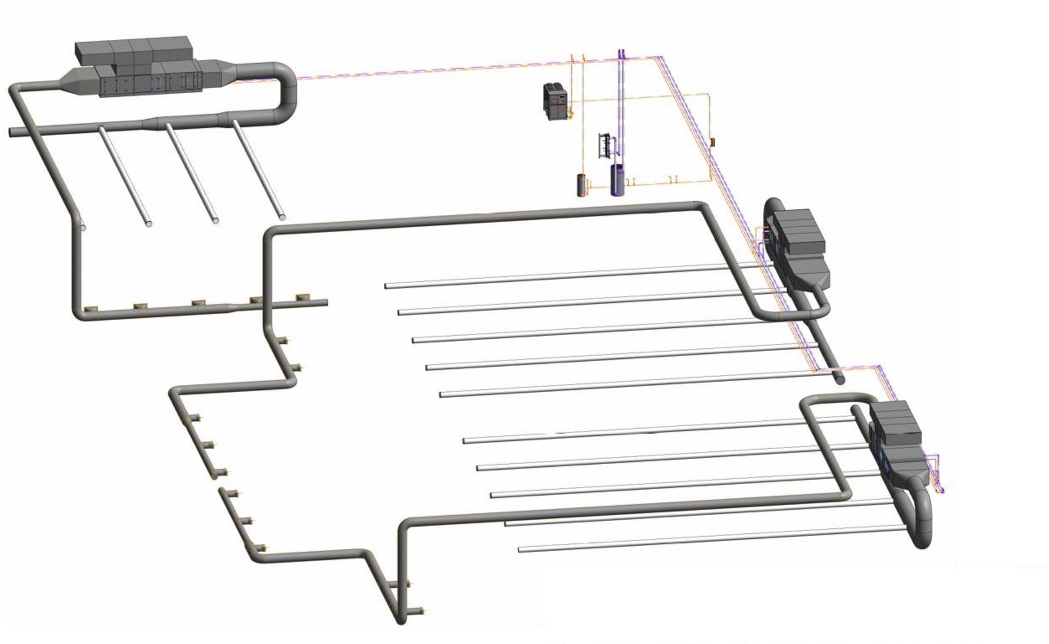

Figure 8.
Proposed HVAC installation.

3.5. System Efficiency

In order to evaluate and compare the efficiency of the systems (original and proposed), the Coefficient of Performance (COP) values for each system were determined [19]. This calculation aimed to enable a rigorous analysis between the existing and proposed solutions, examining the energy performance of each system in terms of its efficiency in utilizing energy to achieve the desired results [19, 20].The COP calculation provides an objective measure of the effectiveness of each system, serving as a crucial element for evaluating the improvements in energy performance resulting from the implementation of the proposed integrated system [20, 21].

3.5.1. Performance of the Original System

The average Coefficient of Performance (COP) of the existing installation was calculated by considering its three main components: the refrigeration system, the heat pump, and the chiller. The COP of the existing refrigeration installation was determined using the following formula:

$$COP_{friex} = \frac{\dot{Q}_{EC} + \dot{Q}_{ER}}{\dot{W}_{CC} + \dot{W}_{CR}} \tag{7}$$

Where:
COP_{friex} - Coefficient of Performance of the existing refrigeration installation.
$\dot{Q}_{EC}$ - Refrigeration capacity of the existing freezer [kW].
$\dot{Q}_{ER}$ - Refrigeration capacity of the existing refrigerated system [kW].
$\dot{W}_{CC}$ - Power absorbed by the compressors of the frozen food system [kW].
$\dot{W}_{CR}$ - Power absorbed by the compressors of the refrigerated system [kW].
The coefficient of performance of the equipment in the original installation, during nominal operation, was calculated using the following expression:

$$COP_{ESSumer} = \frac{\dot{Q}_{EC} + \dot{Q}_{ER} + \dot{Q}_{HVAC\ CO}}{\dot{W}_{CC} + \dot{W}_{CR} + \dot{W}_{CO}} \tag{8}$$

Where:
$COP_{ESSumer}$ - Coefficient of Performance of the existing installation at summer.
$\dot{Q}_{HVAC\ CO}$ - Thermal power of the original cooling batteries [kW].
$\dot{W}_{CO}$ - Power absorbed by the original chiller of HVAC system [kW].

$$COP_{ESWinter} = \frac{\dot{Q}_{EC} + \dot{Q}_{ER} + \dot{Q}_{HVAC\ HO}}{\dot{W}_{CC} + \dot{W}_{CR} + \dot{W}_{BCO}} \tag{9}$$

Where:
$COP_{ESWinter}$ - Coefficient of Performance of the existing installation at winter.
$\dot{Q}_{HVAC\ HO}$ - Thermal power of the original heating batteries [kW].
$\dot{W}_{BCO}$ - Power absorbed by the original heat pump of HVAC system [kW].

3.5.2. Performance of the Proposed System

The proposed system is centralized and utilizes thermal energy for the HVAC system through hydraulic circuits that capture rejected heat from specific areas of the installation. Due to its multifunctional nature, performance was assessed by considering various operating scenarios, enabling the calculation of adjusted Coefficients of Performance (COP), thus allowing a more accurate and realistic comparison of efficiency under different operating conditions.

3.5.3. Nominal COP

The Nominal Coefficient of Performance (COP) is employed to evaluate the performance of the integrated system (cooling and HVAC) under nominal operating conditions, without heat recovery. To enable a more precise characterization, the summer and winter months are considered separately. The COP for the summer period can be estimated using the following equation:

$$COP_{nom\ summer} = \frac{\dot{Q}_{refrigerator} + \dot{Q}_{HVAC\ Cooling}}{\dot{W}_{Clow} + \dot{W}_{Cmedium} + \sum \dot{W}_{circulator\ pumps}} \tag{10}$$

Where:

$COP_{nom\ summer}$ - COP of the system in nominal operation during the summer period;
$\dot{Q}_{refrigerator}$ - Cooling power of the evaporator and fan coil units [kW];
$\dot{Q}_{HVAC\ Cooling}$ - Thermal power of the AWU cooling batteries [kW];
$\dot{W}_{Clow}$ - Power consumed by the low-power compressors [kW];
$\dot{W}_{Cmedium}$ - Power consumed by the medium-power compressors [kW];
$\sum \dot{W}_{circulator\ pumps}$ - Sum of the electrical power consumed by the circulation pumps [kW].

Similarly, the COP for the winter period can be estimated as follows:

$$COP_{nom\ Winter} = \frac{\dot{Q}_{refrigerator} + \dot{Q}_{HVAC\ Heating} + \dot{Q}_{HVAC\ Dehumidification}}{\dot{W}_{Clow} + \dot{W}_{Cmedium} + \dot{W}_{BC} + \sum \dot{W}_{circulator\ pumps}} \tag{11}$$

Where:

$COP_{nom\ Winter}$ - COP of the system in nominal operation during the winter period;
$\dot{Q}_{HVAC\ Dehumidafication}$ - Thermal power of the AWU dehumidification batteries [kW];
$\dot{Q}_{HVAC\ Heating}$ - Thermal power of the AWU heating batteries [kW];
$\dot{W}_{BC}$ - Power consumed by the heat pump [kW];

3.5.4. Coefficient of Performance (COP) of the Refrigeration System

To facilitate a final comparison between solutions, the COP of the proposed refrigeration system was calculated, considering only the refrigeration system components (which are the known elements of the original system). This approach allows for a direct comparison, highlighting the differences from the methodology presented in 3.5.1. The calculation of the COP of the proposed refrigeration system for a global or annual period can be estimated using the following expression:

$$COP_{Refrigeration} = \frac{\dot{Q}_{refrigerator}}{\dot{W}_{Clow} + \dot{W}_{Cmedium} + \dot{W}_{circulator\ pumps}} \tag{13}$$

$COP_{Refrigeration}$ - Coefficient of performance of the refrigeration system (cooling mode) [kW].

4. Results and Discussion

4.1. Comparison of energy performance between the Original and Proposed Systems

A detailed analysis of the energy performance was conducted, encompassing both the original HVAC system and the proposed system. This evaluation considered the thermal power and electrical consumption values of the original installation, as well as the corresponding data for the proposed system, as presented in Table 6. The Coefficients of Performance (COP) were calculated for both systems, enabling a comprehensive technical comparison, as illustrated in Table 7.

Table 6.
Summary of the parameters of proposed system.

Variable	kW
$\dot{Q}_{refrigerator}$	160.91
$\dot{Q}_{HVAC\ Cooling}$	480.0
$\dot{Q}_{HVAC\ heating}$	50.0
$\dot{Q}_{HVAC\ Dehumidadication}$	250.0
$\dot{Q}_{HVAC\ recovered}$	45.0
$\dot{W}_{Clow}$	6.3
$\dot{W}_{Cmedium}$	50.8
$\dot{W}_{BC}$	16.6
$\dot{W}_{circulatorpumps}$	1.1
$\sum \dot{W}_{circulatorpumps}$	16.5

The Table 7 provides the calculated Coefficient of Performance (COP) values for both the original and proposed systems, based on the technical parameters evaluated. These indicators offer a comprehensive overview of the energy efficiency of the systems under different operational conditions, enabling a detailed comparative analysis. The results emphasize a significant improvement in the energy performance of the proposed system compared to the original. This enhancement is particularly evident during nominal speed operation and heat recovery scenarios. The proposed system demonstrates superior efficiency, reducing overall energy consumption and promoting a more sustainable solution for the thermal demands of the installation.

Table 7.
Summary of Coefficient of Performance (COP).

Indicator	Description	Installation	Considered Parameter	Period	COP
COP_{friex}	COP of the original refrigeration system	Original	Existing frozen food circuit capacity; Existing refrigerated food circuit capacity; Power absorbed by frozen food circuit compressors; Power absorbed by refrigerated food circuit compressors	Annual	2.00
$COP_{ESWinter}$	Coefficient of Performance of the existing installation during winter	Original	Thermal power of the original heating batteries; Power absorbed by the original heat pump of HVAC system; Existing frozen food circuit capacity; Existing refrigerated food circuit capacity; Power absorbed by frozen food circuit compressors; Power absorbed by refrigerated food circuit compressors	Winter	2.21
$COP_{ESSummer}$	Coefficient of Performance of the existing installation during summer.	Original	Thermal power of the original cooling batteries; Power absorbed by the original chiller of HVAC system; Existing frozen food circuit capacity; Existing refrigerated food circuit capacity; Power absorbed by frozen food circuit compressors; Power absorbed by refrigerated food circuit compressors.	Summer	2.29
$COP_{nom\ summe}$	COP of the system in nominal operation during the summer period	Proposed	Thermal power of the evaporator and fan coil units; Thermal power of AWU cooling batteries; Power consumed by the low-power compressors; Power consumed by the medium-power compressors; Sum of the electrical power consumed by the circulation pumps	Summer	3.53
$COP_{nom\ Winter}$	COP of the system in nominal operation during the winter period	Proposed	Thermal power of the evaporator and fan coil units; Thermal power of the AWU dehumidification batteries; Thermal power of the AWU heating batteries; Power consumed by the low-power compressors; Power consumed by the medium-power compressors; Power consumed by the heat pump; Sum of the electrical power consumed by the circulation pumps	Winter	3.12

$COP_{Refrigerati}$	Coefficient of performance of the refrigeration system (cooling mode)	Proposed	Thermal power of the evaporator and fan coil units; Power consumed by the low-power compressors; Power consumed by the medium-power compressors	Annual	2.49

This table summarizes the calculated Coefficients of Performance (COP) for both the original and proposed systems across various operating conditions and periods.The comparison of the Coefficient of Performance (COP) between the original and proposed systems reveals a substantial improvement in energy efficiency, driven by the integration of the refrigeration system with the heating, ventilation, and air conditioning (HVAC) system. The original system exhibits a COP of 2.0 during general refrigeration operation, with slightly improved performance during winter (COP 2.21) due to the use of the heat pump. In the summer, the COP increases to 2.29, reflecting the efficiency of the cooling batteries and the chiller. The proposed system, with its optimized design and integration with the HVAC system, demonstrates superior COP values under all operational conditions. In the summer, the COP reaches 3.53, a result of the implementation of an efficient multi-stage system, featuring low and medium-power compressors and low-energy consumption circulation pumps. During winter, the proposed system achieves a COP of 3.12, surpassing the original system's performance, despite a slight reduction compared to summer. The integration with the HVAC system allows for better adaptation to climatic variations, ensuring maintained energy efficiency. Furthermore, the refrigeration system performance in isolation (COP 2.49) also exceeds that of the original system (COP 2.00), indicating higher efficiency due to the improved utilization of compressors and pumps. In conclusion, the integration of the refrigeration system with the HVAC system, the adoption of a multi-stage configuration result in a more efficient and sustainable solution, with lower energy consumption and superior performance across all operational scenarios analysed.

4.2. Comparative Analysis of CO_2 Emissions Between the Original and Proposed Installations

The Table 8 summarizes the CO_2 emissions associated with the operation of the original installation, encompassing both the refrigeration and HVAC systems during summer and winter periods. Calculations were based on the daily energy consumption of the system components, using an emission factor of 98 g CO_2/kWh, projected for Angola in 2025 [22]. This factor reflects the carbon intensity of electricity generation in the country, enabling an accurate assessment of the environmental impact linked to energy consumption. The results include daily and annual emissions for each operating scenario (refrigeration only, summer operation, and winter operation), as well as the total annual emissions for the entire system. These data provide a detailed perspective on the energy and environmental performance of the installation, serving as a foundation for operational optimization strategies and greenhouse gas emission mitigation efforts.

Table 8.
Annual CO_2 Emissions from the Original Installation (Including Refrigeration and HVAC Systems).

Operational scenarios	Total Power (kW)	Operating Hours (h)	Daily Power Consumption (kWh)	Daily CO_2 Emissions (kg)	Annual Power Consumption (kWh)	Annual CO_2 Emissions (kg)
Refrigeration System Only	84.2	18 (Compressors)	1515.60	148.53	553194.00	54213.01
Original Installation (Summer)	84.2 + 165.5	18 (Compressors) + 16 (Chiller)	4779.60	468.40	1338288.00	131152.22
Original Installation (Winter)	84.2 + 11.4	18 (Compressors) + 12 (Heat Pump) + 16 Chiller	3336.80	327.01	283628.00	27795.54
Annual Total (Refrigeration + HVAC Systems)	-	-			1621916.00	158947.78

The Table 8 provides a summary of the CO_2 emissions from the original installation, considering the refrigeration system and HVAC operation modes during both summer and winter periods. When compared to similar installations, the energy consumption and emissions of the original installation align with conventional systems that employ compressors, chillers, and heat pumps in continuous operation [23, 24]. However, the high carbon intensity associated with the emission factor of 98 g CO_2/kWh, projected for Angola in 2025, underscores the necessity for adopting more efficient technologies and optimization strategies to mitigate greenhouse gas emissions.

Table 9 presents the technical specifications and performance of various proposed electric-powered cooling systems, including their energy consumption and associated carbon dioxide emissions. The systems are evaluated for their daily and annual performance under different operational scenarios, such as summer and winter conditions, with a detailed assessment of components like compressors, circulator pumps, and heat pumps. The calculations are based on an emission factor of 98 g CO_2/kWh projected for Angola in 2025.

Table 9.
Annual CO_2 Emissions from the proposed Installation (Including Refrigeration and HVAC Systems).

Operational scenarios	Total Power (kW)	Operating Hours (h)	Daily Power Consumption (kWh)	Daily CO_2 Emissions (kg)	Annual Power Consumption (kWh)	Annual CO_2 Emissions (kg)
Proposed Refrigeration System only	63.4 + 1.1	18h (Compressors +16h (circulator pumps)	1158.8	113.56	422962.00	41450.28
Proposed Installation (Summer)	165.1 + 16.5	18 (Compressors + 16 (circulator pumps)	3235.80	317.11	808950.00	79277.10
Proposed Installation (Winter)	147.3 +16.6 + 16.5	18 (Compressors) + 12 (Heat Pump) + 16 (circulator pumps)	3114.60	305.23	264741.00	25944.62
Proposed Annual Total (Refrigeration + HVAC Systems)	-	-			1073691.00	105221.72

The analysis of Table 9 reveals significant differences in energy consumption and CO_2 emissions, highlighting the importance of integrating the HVAC system with the refrigeration system in the proposed scenario, compared to the original system. In the proposed standalone refrigeration system, the annual consumption is 422,962kWh, with 41,450.28kg of CO_2 emitted, a typical value for this type of installation. Although the refrigerant used has low CO_2 emissions, the high carbon footprint of the electrical grid in Luanda significantly contributes to this value. On the other hand, in the proposed integrated system, the annual consumption is 1,073,691 kWh, with 105,221 kg of CO_2 emitted. This increase in energy consumption is expected due to the country's climatic conditions, which involve high temperatures and humidity, especially during the summer. However, there is an observed improvement in efficiency and a reduction in CO_2 emissions in the integrated scenario, underscoring the importance of the integration between the refrigeration and HVAC systems for more efficient energy management and environmental impact reduction.

4.3. Financial Analysis of the Original and Proposed Installations

The following table presents a comparative analysis between the original system and the proposed system, considering the annual operational costs and the associated CO_2 emissions for each system. The electricity cost in Luanda is assumed to be 0.012 EUR/kWh, while the cost of CO_2 emissions is calculated based on a price of 80 EUR per ton of CO_2, in line with typical values observed in the European Union Emissions Trading System (EU ETS). This analysis not only assesses the direct operational costs but also the environmental impact in terms of CO_2 emissions, highlighting the cost savings generated by the proposed system relative to the original system.

Table 10 .
Comparison of operational costs between the original and proposed systems, including energy consumption and CO_2 emissions.

Aspects	Original System	Proposed System	Savings (Proposed vs original)
Annual Energy Consumption (kWh)	1,621,916	1,073,691	548,225
Annual Operational Cost (€)	19,462.99 €	12,884.29 €	6,578.70 €
Annual CO_2 Emissions (kg)	158,947.78	105,221.72	53,726.06
CO_2 Emissions Cost (80 EUR/ton)	12,715.82 €	8,417.74 €	4,298.08 €

This study demonstrates that the proposed system is not only more economically viable, with a significant reduction in operational costs, but also more environmentally sustainable, offering a substantial decrease in CO_2 emissions.

5. Conclusions

The present study highlighted the significance of integrating refrigeration systems with HVAC systems using natural fluids, demonstrating the operational and environmental benefits resulting from this approach compared to the use of separate systems for refrigeration and HVAC. The analysis revealed a considerable increase in the Coefficient of Performance (COP) of the proposed system, which rose from 2.29 in the original system to 3.53 in the integrated system during summer conditions, and from 2.21 to 3.12 during winter conditions, indicating a substantial improvement in energy efficiency. This increase in COP reflects a more efficient use of energy, enabling superior performance with lower consumption. Regarding energy consumption, the proposed integrated system demonstrated an annual reduction of 548,225 kWh compared to the original system, resulting in a significant energy saving of approximately 33.8%. This reduction in energy consumption also led to a decrease of 53,726 kg in CO_2 emissions, corresponding to a 33.8% savings in annual greenhouse gas emissions. These results are particularly relevant in the context of global emission reduction commitments, such as the Kyoto Protocol and the Paris Agreement, which aim to limit global warming and promote sustainable practices across various industries. The integration of refrigeration and HVAC systems, utilizing natural fluids, significantly contributes to the reduction of CO_2 emissions, aligning with the objectives of these international agreements and promoting a shift towards more sustainable solutions. This study, by highlighting the reduction of environmental impact and improvement in energy efficiency, underscores the importance of system integration in the engineering field. The adoption of solutions such as the proposed one can serve as a foundation for future innovations and the development of more sustainable technologies in the refrigeration and HVAC industries. Furthermore, the medium- and long-term economic benefits derived from reduced operational costs and increased energy efficiency make the integrated system not only an environmentally responsible solution but also an economically advantageous one, establishing it as a cost-effective and sustainable alternative. Therefore, the importance of this study lies in its contribution to advancing practices and technologies in the engineering field, offering an integrated solution that not only improves operational performance but also supports global climate change mitigation goals and the promotion of a more sustainable energy future.

References

[1] F. Giunta and S. Sawalha, "Techno-economic analysis of heat recovery from supermarket's CO2 refrigeration systems to district heating networks," *Applied Thermal Engineering,* vol. 193, p. 117000, 2021. https://doi.org/10.1016/j.applthermaleng.2021.117000

[2] 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. https://doi.org/10.1016/j.egyr.2024.04.033

[3] J. Garcia and A. Semedo, "Sustainable CO2 refrigeration system for fish cold storage facility using a renewable integrated system with solar, wind and tidal energy for cape verde—analyzing scenarios," *Sustainability,* vol. 16, no. 10, p. 42259, 2024. https://doi.org/10.3390/su16104259

[4] P. D'Agaro, M. A. Coppola, M. Libralato, and G. Cortella, "HVAC coverage in integrated refrigeration systems at various climate and building conditions," *Applied Thermal Engineering,* vol. 266, p. 125503, 2025. https://doi.org/10.1016/j.applthermaleng.2025.125503

[5] European Union, "Regulation (EU) No 517/2014 of the European parliament and of the council of 16 April 2014 on fluorinated greenhouse gases and repealing regulation (EC) No 842/2006 (OJ L 150, pp. 195–230)," European Union, 2014.

[6] European Union, "Regulation (EU) 2024/573 of the European parliament and of the council of 7 February 2024 on fluorinated greenhouse gases, amending Directive (EU) 2019/1937 and repealing Regulation (EU) No 517/2014. Official Journal of the European Union, L (publication number not specified)," European Union, 2024. http://data.europa.eu/eli/reg/2024/573/oj

[7] A. Arteconi and F. Polonara, "Demand side management in refrigeration applications," *International Journal of Heat and Technology,* vol. 35, no. 1, pp. 58-63, 2017. https://doi.org/10.18280/ijht.35Sp0108

[8] Y. Lu and E. Huo, "Viewpoints on the refrigeration by renewable energy," *Frontiers in Energy Research,* vol. 10, p. 928187, 2022. https://doi.org/10.3389/fenrg.2022.928187

[9] S. Smitt, I. Tolstorebrov, and A. Hafner, "Integrated CO2 system with HVAC and hot water for hotels: Field measurements and performance evaluation," *International Journal of Refrigeration,* vol. 116, pp. 59-69, 2020. https://doi.org/10.1016/j.ijrefrig.2020.03.021

[10] E. Fabrizio, F. Seguro, and M. Filippi, "Integrated HVAC and DHW production systems for Zero energy buildings," *Renewable and Sustainable Energy Reviews,* vol. 40, pp. 515-541, 2014. https://doi.org/10.1016/j.rser.2014.07.193

[11] G. Maouris, E. J. Sarabia Escriva, S. Acha, N. Shah, and C. N. Markides, "CO2 refrigeration system heat recovery and thermal storage modelling for space heating provision in supermarkets: An integrated approach," *Applied Energy,* vol. 264, p. 114722, 2020. https://doi.org/10.1016/j.apenergy.2020.114722

[12] S. O. Banjo *et al.*, "Performance assessment of a refrigeration system with an integrated condenser under different environmental conditions," *Heliyon,* vol. 10, no. 7, p. e29226, 2024. https://doi.org/10.1016/j.heliyon.2024.e29226

[13] A. K. Vuppaladadiyam *et al.*, "Progress in the development and use of refrigerants and unintended environmental consequences," *Science of The Total Environment,* vol. 823, p. 153670, 2022. https://doi.org/10.1016/j.scitotenv.2022.153670

[14] H. M. Hashim, E. Sokolova, O. Derevianko, and D. B. Solovev, "Cooling load calculations," *IOP Conference Series: Materials Science and Engineering,* vol. 463, no. 3, p. 032030, 2018. https://doi.org/10.1088/1757-899X/463/3/032030

[15] M. D. Pereira and A. S. de Oliveira, "Analysis of the thermal load calculation of a beef carcass cooling chamber with a capacity of 42 tons," *The Journal of Engineering and Exact Sciences,* vol. 6, no. 5, pp. 0777-0782, 2020. https://doi.org/10.18540/jcecvl6iss5pp0777-0782

[16] A. Alaidroos, "Transient behavior analysis of the infiltration heat recovery of exterior building walls," *Energies,* vol. 16, no. 20, p. 7198, 2023. https://doi.org/10.3390/en16207198

[17] J. M. Dias *et al.*, "Impact of environmental conditions on the ripening of Queijo de Évora PDO cheese," *Journal of Food Science and Technology,* vol. 58, no. 10, pp. 3942-3952, 2021. https://doi.org/10.1007/s13197-020-04856-x

[18] N. Ricardo Costa and J. Garcia, "Applying design of experiments to a compression refrigeration cycle," *Cogent Engineering*, vol. 2, no. 1, p. 992216, 2015. https://doi.org/10.1080/23311916.2014.992216
[19] N. Costa and J. Garcia, "Using a multiple response optimization approach to optimize the coefficient of performance," *Applied Thermal Engineering*, vol. 96, pp. 137-143, 2016.
[20] S. Thanasoulas, J. Arias, and S. Sawalha, "Investigating the heating and air conditioning provision capability of a supermarket to neighboring buildings: Field measurement analysis and economic evaluation," *Applied Thermal Engineering*, vol. 230, p. 120782, 2023.
[21] J. Bak, J. Koo, S. Yoon, and H. Lim, "Thermal draft load coefficient for heating load differences caused by stack-driven infiltration by floor in multifamily high-rise buildings," *Energies*, vol. 15, no. 4, p. 1386, 2022.
[22] Republic of Angola Ministry of Energy and Water, "ANGOLA ENERGIA 2025: Long-term vision for the electricity sector / angola power sector long term vision. Luanda, Angola," 2025. https://angolaenergia2025.gestoenergy.com/pt-pt/conteudo/geracao-0?utm_source=chatgpt.com
[23] F. Alan, B. Tim, and E. Judith, "Carbon emissions from refrigeration used in the UK food industry," *International Journal of Refrigeration*, vol. 150, pp. 297-303, 2023.
[24] Z. Zhu, S. Zhao, J. Liu, M. Wang, and X. Lv, "Evaluation of carbon emissions for fresh food last-mile delivery: A comparative analysis of two refrigeration methods," *Case Studies in Thermal Engineering*, vol. 60, p. 104787, 2024.