Analysis and optimization of water-loop systems with low-GWP multipurpose heat pumps

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Abstract

Heat pumps are among the most promising technologies to reduce greenhouse gas emissions, especially when combined with low-GWP refrigerants. In this context, the combination of heat pumps and water loops for heat dissipation and valorization has recently gained interest. This study evaluates the performance of heat pumps employing different refrigerants (R454B and R290) within the water-loop system of a supermarket in London, where a water loop is used for heat dissipation from refrigerated counters. The water loop operates at a nominal flow rate of 11.5 l/s and a supply temperature of 18 °C. Peak heat loads reach 95 kW in winter and 270 kW in summer. The existing setup includes a dry cooler, two R454B reversible air-to-water heat pumps, and an R454B water-to-water heat pump active in winter for building climatization by exploiting the water-loop heat. Two retrofit scenarios are compared. The first configuration includes the pre-existing dry cooler, two R290 air-to-water heat pumps and an R290 multipurpose heat pump, which is able to release the condensation heat either to air or to water for building climatization. The second solution involves only R290 multipurpose heat pumps, used to thermally stabilize the water loop, satisfying the building heating demand. In order to compare the different solutions, energy analyses are performed through hourly simulations. Results show that both retrofit alternatives outperform the current system in terms of energy efficiency, environmental impact, and economic feasibility, with the second configuration offering the best overall performance (seasonal efficiency up to +20% of the current setup). The study demonstrates the viability of the proposed system and suggests directions for further technological development of water loops with multipurpose heat pumps using natural refrigerants in commercial, industrial, and district heating/cooling applications.

Keywords: Water loop; Low-GWP refrigerant; Multipurpose heat pump; Refrigeration; Heating; Air-to-water heat pump; Water-to-water heat pump.

1. Introduction

In recent years, cl[1]imate change has emerged as one of the most urgent and important challenges of our time. Refrigerant fluids, primarily used in the climatization and product refrigeration sector, are among the main contributors to this phenomenon, which has environmental, economic, and social impacts that directly affect people's quality of life.

Heat pumps are well-established, widely adopted technologies which, thanks to their high energy efficiency and use of renewable energy sources, are excellent alternatives to traditional air conditioning generators. However, it is essential to pay attention to the refrigerants used in these devices [1], as many have high Global Warming Potential (GWP) values. In this context, R454B and R290 (the latter also known as propane) stand out for their low GWP and excellent thermodynamic properties, making them particularly suitable for use in heat pumps.

Globally, there is a push towards this type of technology through both national and international regulations. Through the Green Deal [2] and the 'Fit for 55' legislative package [3], the European Union has set a target to

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reduce greenhouse gas emissions by 55% by 2030 compared to 1990 levels and to achieve climate neutrality by 2050. The usage of heat pumps employing natural refrigerants has also been encouraged through the RED II [4] and F-Gas [5] regulations.

Recently, there has been growing interest in water-loop system solutions for rejecting condensation heat, combined with heat recovery systems. Indeed, dynamic simulation studies on Water Loop Heat Pump (WLHP) systems demonstrated the feasibility of this integrated approach in buildings with simultaneous heating and cooling loads, in terms of energy consumption, operating costs, and environmental impact [6]. A typical application of WLHP systems is in supermarkets, where there are opposite requirements for air conditioning, domestic hot water production, and product refrigeration. Recovering heat from certain processes and utilizing it in others through reversible machines has proven to lead to energy and economic savings, thereby increasing the overall efficiency of the system. Polzot et al. [7] analyzed the energy-saving potential of a WLHP system with HFO (hydrofluoroolefin) heat pumps connected to a CO_2 booster refrigeration unit in a supermarket, where heat recovery is obtained from the refrigeration circuit during the heating season, and demonstrated significant improvements in overall efficiency compared to separated systems. Coccia et al. [8] investigated different control strategies based on real-time pricing to maximize electricity cost savings in a supermarket integrated HVAC, refrigeration, and WLHP system, including heat pumps with HFO as low-GWP refrigerant. Recently, Huang et al. [9] investigated the techno-economic feasibility of heat recovery in a supermarket refrigeration system integrated with a district heating system, developing dynamic control strategies that demonstrated annual energy cost savings ranging from 41% to 93%.

Applications of water-loop systems employing low-GWP heat pumps have shown great potential for energy and economic benefits. However, there are still few published results on water-loop systems with multipurpose heat pumps, namely with refrigeration units able to work not only as Air-to-Water (A-W) machines, but also as Water-to-Water (W-W) ones, satisfying opposite loads. In this regard, this work aims to analyze the performance of a water-loop system with multipurpose heat pumps using low-GWP refrigerants, evaluating their contribution to achieving environmental and energy efficiency.

The study focuses on a supermarket in London with an installed system comprising two A-W heat pumps and a W-W heat pump, connected via a water loop at the service of the supermarket refrigerated counters. In addition to the existing system, this study examines two possible alternatives, both involving the replacement of the R454B refrigerant with propane, which has a lower GWP (3 for R290 vs 466 for R454B). The difference between the proposed solutions is in the number and the type of machines installed. The study aims to determine which system offers the highest energy and economic savings, while also accounting for differences in the environmental impact.

For the performance analysis conducted on the heat pumps, the REF.BOX software [10] was used, which provided all the data needed to simulate the heat pumps operation as the external temperature and the number of compressors varied. In order to calculate the system efficiency, the seasonal energy performance ratio (SEPR) and the seasonal coefficient of performance (SCOP) were calculated by means of an hourly analysis of the three case studies, concluding with an environmental impact assessment and an economic analysis.

2. Case Study

The supermarket object of the present study is equipped with a hot circuit serving the air handling units (AHUs) heating coils for winter space heating, and a cold loop that removes heat from the refrigerated counters throughout the year. In this study, both the AHUs and the refrigerated counters are treated as black boxes which require fixed temperature differences between the supply and the return of the water circuits. The cold water loop has a nominal flow rate of 11.5 l/s in both summer and winter; the heat transfer fluid consists of a water-glycol mixture containing 15% glycol. The cold water loop load varies slightly with the external air temperature within a season and has a summer peak load of 261 kW at an external design temperature of 30 °C, and a winter peak load of 95 kW. The supply/return temperatures to the refrigerated counters are 18/20 °C in winter and 18/24 °C in summer, with minor flow rate adjustments through inverter-driven pumps due to the outdoor air temperature variations. The hot circuit designated for the AHUs operates during the winter months only, with fixed supply and return temperatures of 50 and 45 °C, respectively, and variable mass flow rate on the basis of the building thermal load, depending on the external air temperature value; the thermal load has a peak of 343 kW at a design outdoor temperature of –4 °C and decreases linearly until an external air temperature value of 16 °C.

To ensure the system operates correctly, the temperature of the two water circuits must be stabilized by means of heat pumps/chillers to meet the users' energy demand under all load conditions. In particular, different water loop systems are analyzed in order to determine which configuration yields the highest energy and economic savings, also evaluating the environmental impact:

- System A: currently installed system with two air-to-water heat pumps and a water-to-water heat pump, all with R454B;
- System B: retrofit scenario with two air-to-water heat pumps and a multipurpose heat pump, all with R290;
- System C: retrofit scenario with three multipurpose heat pumps, all with R290.

2.1. System A

The devices serving the existing plant (system A) are two PLE122EC chillers and a WLE274C heat pump, manufactured by Galletti SpA [11], and a dry cooler. The PLEs are single-circuit reversible A-W machines that operate with R454B as refrigerant. These devices are equipped with two uneven compressors each and have a cooling capacity of 144 kW at an outdoor air temperature of 30 °C and water temperatures of 24/18 °C. They guarantee the possibility of producing water from -10 to 55 °C with outside air ranging from -12 to 46 °C. The WLE is a W-W dual-circuit heat pump operating with R454B as well. It has four compressors distributed unevenly in pairs across the two different refrigerant circuits and a heating capacity of 346 kW with water temperatures of 20/18 °C on the cold side and 45/50 °C on the hot side.

In summer (see Fig. 1a), the two PLE122EC chillers must stabilize the cold water loop to maintain the supply temperature stable at 18 °C. The AHUs are not used during summer, so the hot water circuit is not employed, and the WLE274C heat pump is switched off. Also the dry cooler is practically not used, because it can dissipate heat only if the external air temperature is below 15 °C.

In winter (see Fig. 1b), the heating demand has priority and is satisfied by the WLE274C heat pump, which discharges heat to the hot water circuit and extracts heat from the cold water loop (W-W unit). The two reversible PLE122EC units act as chillers or heat pumps to restore the cold water loop supply temperature depending on the heat extracted from the loop by the WLE274C unit. If the outside temperature is below 15 °C and the loop must be cooled, the dry cooler is used with priority, as its power consumption is much lower than that of the refrigeration machines.

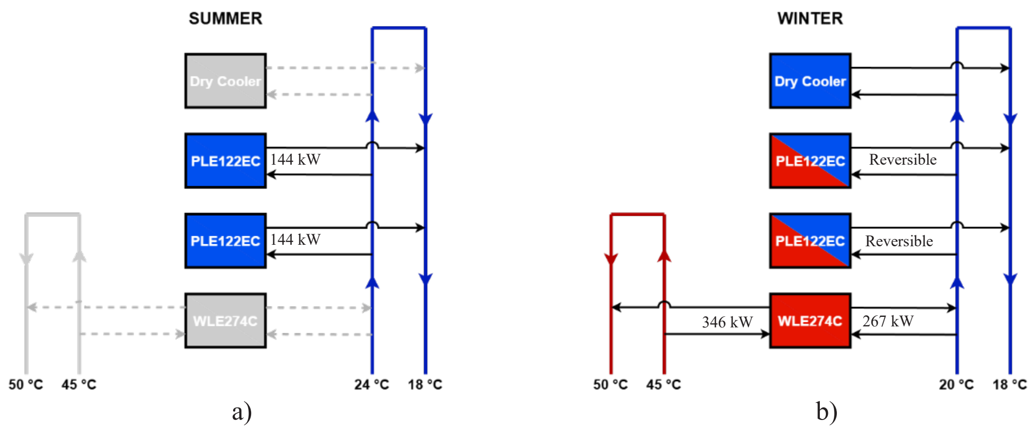

Figure 1: System A operating diagram during summer (a) and winter (b), with maximum capacities at design conditions.

2.2. System B

This system configuration involves installing three different units: a PLN-P174, a PLN-H174, and a PLN-H144 [11]. The PLN-P174 unit is a four-pipes monobloc A-W heat pump which uses R290 (propane) as refrigerant. It is a multipurpose heat pump, namely it can work not only as an A-W machine, but it is also able to recover the condensation heat, thus working as a W-W unit to satisfy contemporary cooling and heating loads. The machine, which has a cooling capacity of 210 kW with external air at 30 °C and water at 24/18 °C, is equipped with four scroll compressors, configured in even mode and arranged in pairs on two independent circuits. The unit can produce water from 5 to 55 °C with outside air from -15 to 45 °C. The two PLN-H units are A-W monobloc reversible heat pumps operating with R290 and able to produce water from -10 to 55 °C with external temperatures from -15 °C to 45 °C. The PLN-H174, with the same capacity as the P model, has four even compressors arranged in pairs on two different circuits, whereas the PLN-H144 has a cooling capacity at summer design conditions of 171 kW and uneven compressors.

During summer (see Fig. 2a), space heating is not required inside the supermarket. As a consequence, the PLN-H174 unit, which is connected only to the hot circuit serving the AHUs, is not active, and the PLN-P174 multipurpose heat pump operates as an A-W chiller condensing on the finned coil and evaporating on the plate heat exchanger to refrigerate the cold loop. In addition, the PLN-H144 unit is used as a chiller to dissipate the excess heat produced by the refrigerated counters. The dry cooler connected to the cold loop is hardly ever active due to the high external air temperatures.

During winter (see Fig. 2b), the multipurpose PLN-P174 unit is operated in W-W mode to remove thermal power from the cold water loop and release thermal power to the AHUs hot circuit at the same time. However, the PLN-P174 alone cannot meet the AHUs thermal demand, which is given priority, so the PLN-H174 A-W unit is operated in heat pump mode to serve the air handling unit coils as well. Since the use of the PLN-P174 unit would yield in many conditions a too low supply temperature to the refrigerated counters, this must be re-stabilized at 18 °C by the PLN-H144 unit operating in heat pump configuration. However, in the few winter conditions of elevated external air temperature, the thermal load of the AHUs circuit is low, and so is the heat extracted from the cold loop by the PLN-P174 unit. As a consequence, the PLN-H144 may be required to cool the cold water loop in these conditions, thus working a chiller (anyway, if the external air temperature is below 15 °C, the dry cooler is activated first).

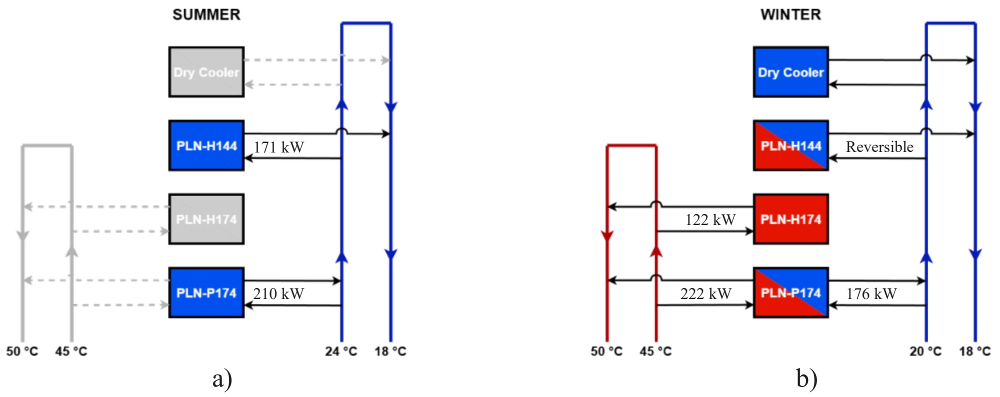

Figure 2: System B operating diagram during summer (a) and winter (b), with maximum capacities at design conditions.

2.3. System C

System C involves three multipurpose heat pumps: one PLN-P114 and two PLN-P174 [11]. The multipurpose model is the same as in the previous section, but two units have a cooling capacity at design conditions of 210 kW and one of 130 kW. This configuration is designed to ensure greater redundancy and energy versatility for the system. Indeed, these multipurpose devices can operate both as A-W and W-W units, allowing to effectively satisfy opposite loads simultaneously through heat recovery, and to enhance the system efficiency also during defrost cycles in winter (not considered in the present work). In addition, system C represents a plug-and-play solution on the current system, which is a 4-pipe system (cold water loop and hot circuit), where employing standard 2-pipe heat pumps would require higher complication diverting the flow to the two circuits with several motorized 3-way valves, which in turn communicate with the machines.

During summer (see Fig. 3a), one of the two PLN-P174 heat pumps is not active, as the PLN-P114 unit and one PLN-P174 unit are sufficient to meet the refrigeration load of the cold water loop. Since the hot circuit to the AHUs is not used during summer, both the PLN-P174 and the PLN-P114 multipurpose units operate as chillers in A-W configuration, condensing on the finned coil to dissipate to the external air the heat produced by the refrigeration counters.

During winter (see Fig. 3b), there is a fundamental difference compared to the previous two systems (systems A and B): priority is no longer given to satisfying the AHUs thermal load, but to satisfying the heat removal demand of the cold water loop. The cooling power to be provided to the cold water loop is normally supplied entirely by the PLN-P114 unit, which operates in heat recovery configuration (W-W), satisfying at the same time a part of the thermal power required by the AHUs. Since the PLN-P114 alone cannot meet entirely the AHUs thermal demand in most situations, the two PLN-P174 units operate as heat pumps in A-W configuration, serving the AHUs coils. If, on the contrary, the AHUs thermal demand is particularly low (high external air temperatures in winter), the two PLN-P174 units are switched off and part of the condensation heat of the PLN-P114 machine is released to the external air.

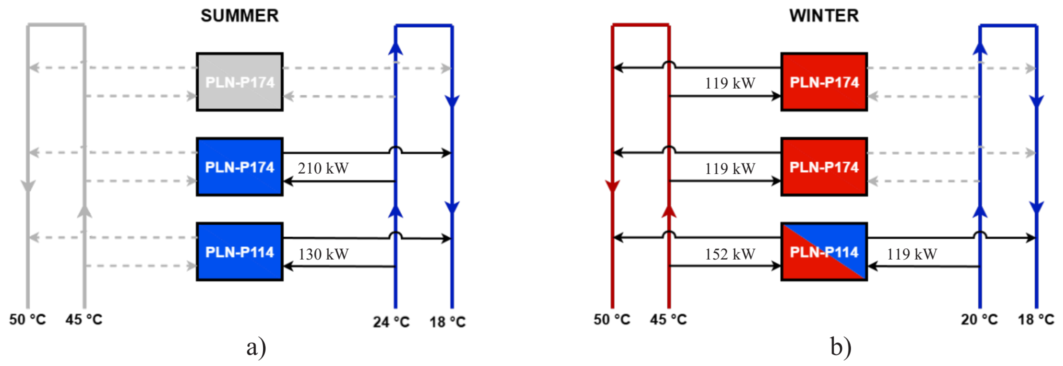

Figure 3: System C operating diagram during summer (a) and winter (b), with maximum capacities at design conditions.

3. Key performance indicators

To assess the most efficient configuration, an analysis based on hourly energy simulations during the supermarket operating periods is presented. Additionally, the environmental impact and an economic comparison were also examined.

3.1. Energy performance

The key performance indicator evaluated for the summer period is the SEPR (seasonal energy performance ratio), which assesses the seasonal energy efficiency of refrigeration process units accounting for load variations and variable climatic conditions throughout the summer period [12]. The SEPR is defined as:

$$SEPR = \frac{\sum_{j=1}^{n} h \cdot P_c(T_j)}{\sum_{j=1}^{n} h \cdot \frac{P_c(T_j)}{EER(T_j)}} \tag{1}$$

where n is the total number of hours of the simulation, h is the hour duration, T_j is the outdoor air temperature of the j-th hour, and $P_c(T_j)$ and $EER(T_j)$ are the cooling load and chiller energy efficiency ratio associated to T_j, respectively.

Similarly, the parameter evaluated for the winter period is the SCOP (seasonal coefficient of performance), which assesses the seasonal energy efficiency of heat pumps in heating mode [12]:

$$SCOP = \frac{\sum_{j=1}^{n} h \cdot P_h(T_j)}{\sum_{j=1}^{n} h \cdot \frac{P_h(T_j)}{COP(T_j)}} \tag{2}$$

where $P_h(T_j)$ and $COP(T_j)$ are the heating load and the heat pump coefficient of performance associated with the external air temperature T_j, respectively.

The hourly values of outdoor air temperature in London were obtained from the test reference year (TRY) of Meteonorm database in TRNSYS [13] (see Fig. 4). A dedicated VBA code was developed to calculate, for each hour of the summer (winter) season: (i) the heat pump capacity ratio, (ii) the number of compressors in operation, (iii) the cooling (thermal) power supplied, and (iv) the consequent EER (COP) value. The SEPR (SCOP) obtained on a seasonal basis was then employed to perform the comparison between the three system configurations.

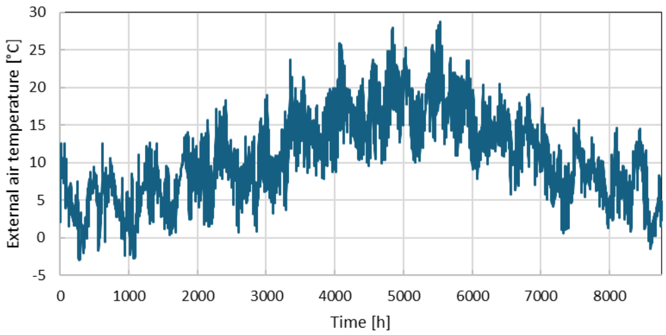

Figure 4: Hourly temperature values from the Meteonorm TRY of London.

3.2. Environmental impact

The environmental impact analysis was conducted using the total equivalent warming impact (TEWI) index [$kgCO_{2eq}$], as defined in Eq. (3):

$$TEWI = c_m \cdot GWP \cdot (f \cdot \tau + p) + E \cdot \beta \cdot \tau \quad (3)$$

where c_m is the refrigerant charge [kg], GWP is the global warming potential (3 for R290 and 466 for R454B), f is the annual refrigerant leakage rate (estimated at 3% in this case), τ is the estimated operating life of the system (10 years for the studied case), p is the percentage of refrigerant lost and not recoverable at the end of the system life (here 10%), E is the annual electricity consumption of the system, and β is the CO_2 emission factor for the generation of electricity used by the system (0.21 $kgCO_2$/kWh [14]).

3.3. Economic comparison

The economic analysis was performed by comparing the three systems with an additional reference configuration (system D), which employs a traditional heat generator. In particular, the system D layout is the same as that of system A, but the thermal load required by the AHUs is met by a condensing boiler (model Vitocrossal 300 CI3 by Viessmann [15]). As a consequence, during summer system D operates like system A, whereas in winter no W-W heat pumps are connected to the cold loop and the gas boiler provides the required heat for the hot circuit.

By interpolating the values of the manufacturer data sheets, the boiler operating conditions were obtained, and the gas consumption values were determined for each condition.

The economic analysis was assessed by calculating the net present value (NPV) and the payback period (PBP), by considering the total investment costs of the different system configurations and the costs of electricity and natural gas in the United Kingdom.

4. Results

4.1. Summer period

During summer, the external design temperature is 30 °C; the cold water loop supply temperature is 18 °C and the return one is 24 °C. The hourly analysis is conducted over the period from July 15 to September 15.

4.1.1. System A

In the installed system, only the two A-W chillers are active during this period, which are required to dissipate the heat generated in the cold water loop by the supermarket refrigerated counters. Each of the two chillers has three possible partialization steps, obtained by activating or deactivating one of the two compressors, resulting in a total of eight possible operative configurations: for each outdoor air temperature, the operating configuration was selected by means of REF.BOX [10], identifying the one that satisfies the cooling load with the minimum excess power. The consequent EER values were determined for every outdoor air temperature, obtaining the trend shown in Fig. 5, which refers to the sum of both chillers.

The value of SEPR yielded by the hourly simulation is 5.0.

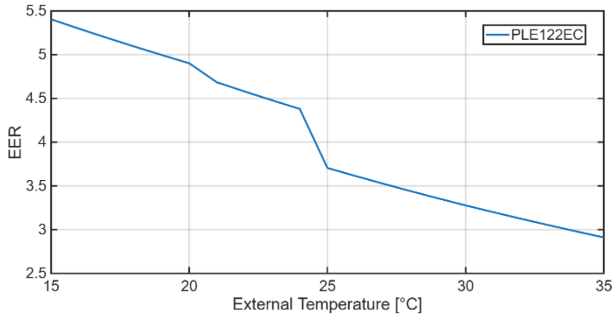

Figure 5: EER values of system A units as a function of the external air temperature.

4.1.2. System B

In system B, during summer only the PLN-P174 and the PLN-H144 units are active; both operate as A-W chillers to handle the cooling load in the water loop. For each outdoor temperature value, the following data were obtained for each chiller at each partialization step: refrigeration power, electrical power consumption, and EER. As there are two active machines, each with four possible partialization steps, a total of 16 operating configurations were to be analyzed. The obtained EER values as a function of the external air temperature are reported in Fig. 6. Indeed, also in this case, for each outdoor temperature, the operating configuration that provides the minimum cooling capacity satisfying the water loop load was selected, and the SEPR was calculated accordingly, resulting in a value of 5.65.

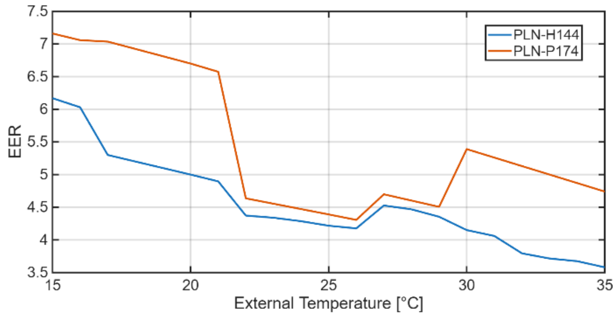

Figure 6: EER values of system B units as a function of the external air temperature.

4.1.3. System C

In system C, only one of the two PLN-P174 units and the PLN-P114 unit are active during the summer period, as they are sufficient to satisfy the cooling load of the cold water loop. Both multi-purpose machines operate in A-W mode. Also in this case, the total number of operative configurations through machine partialization is 16. Once the optimal operative configuration for each outdoor temperature was identified, the SEPR value was calculated, obtaining a value of 5.94. In Fig. 7, the EER values for the chillers of system C are reported as functions of the external temperature.

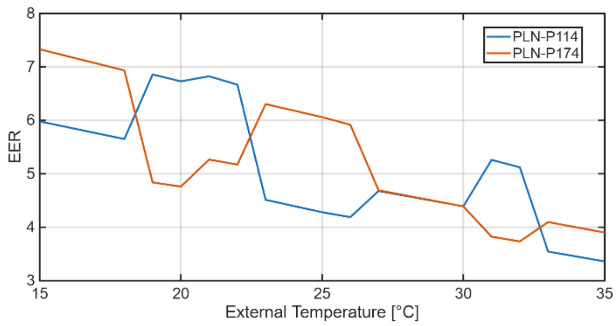

Figure 7: EER values of system C units as a function of the external air temperature.

4.2. Winter period

In winter, the design outdoor temperature is -4 °C; the cold water loop has a return temperature from the refrigerated counters of 20 °C and requires a supply temperature of 18 °C. At the same time, the AHUs that provide space heating to the supermarket require thermal power. The hourly analysis is conducted over the period from January 1 to June 1.

4.2.1. System A

During the winter period, to meet the AHUs thermal load, the WLE unit is activated, extracting heat from the cold water loop at the same time. Since the source temperature is fixed at 20 °C, the only degree of freedom in the machine operation lies in partialization. In this case, six different operating configurations are possible. Starting from the thermal power requirement of the AHUs under design conditions, the thermal power required under other external temperature conditions was calculated. Consequently, the most efficient compressor configuration was determined for every outdoor air temperature. Once the thermal power of the WLE was determined, it was possible to calculate its effect on the cold water loop temperature, which the PLEs or dry cooler must restore, whether cooling or heating the loop. Once the actual load on the cold water loop was determined, it was possible to evaluate the most efficient operating configuration for the two PLEs too, which have eight different partialization options. In Fig. 8 the COP trend versus the external air temperature is shown for the different units (the two PLEs are considered in heat pump mode and reported as a single curve). As shown in the figure, the WLE COP is not particularly affected by the outside temperature, as the machine does not exchange heat directly with the outdoor air (W-W unit). However, the outdoor air temperature influence is not completely negligible, because the WLE unit must satisfy the supermarket space heating demand, which is influenced by the external air value. On the contrary, the PLEs COP is highly dependent on the outdoor temperature: as it increases, so does the COP. The PLEs stop working when the outside temperature is higher than 11 °C, as there is no longer a need for them to heat the cold water loop. The obtained SCOP value is equal to 4.8.

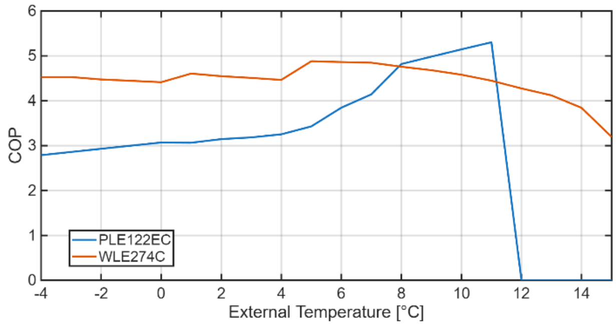

Figure 8: COP values of system A units as a function of the external air temperature.

4.2.2. System B

During the winter season, with system B, the thermal load required by the AHUs is met by the PLN-P174 and PLN-H174 heat pumps, while the PLN-H144 unit is used mainly as a heat pump to re-stabilize the cold water loop temperature. Unlike in system A, here there are two units connected to the hot water circuit: an A-W and a W-W. The former varies its operating conditions based on the outside air temperature and partialization, while the latter varies only based on partialization. There are 18 operative configurations available through partialization to provide the thermal energy to the hot water circuit: the one that satisfies the load with the minimum thermal power was selected. Once the number of active compressors was determined for each external air temperature value, it was also possible to calculate the electricity consumption and the associated COP. By balancing the thermal power removed from the cold water loop by the PLN-P174 unit for AHUs heating and the thermal power that the supermarket counters need to be removed from the cold water loop, it was possible to evaluate the operation of the PLN-H144 A-W unit and the dry cooler to stabilize the cold water loop. From this balance, the optimal configuration for the entire system operation was determined and the associated COP values are reported in Fig. 9. As evident from Fig. 9, at the lowest external temperatures all three machines are switched on, as the AHUs thermal load is at its maximum and, consequently, also the need to restabilize the cold water loop temperature. At milder external temperatures, the first to switch off is the PLN-H144. The PLN-H174 also shuts down when the external temperature exceeds 13 °C. System B reached a SCOP of 5.85.

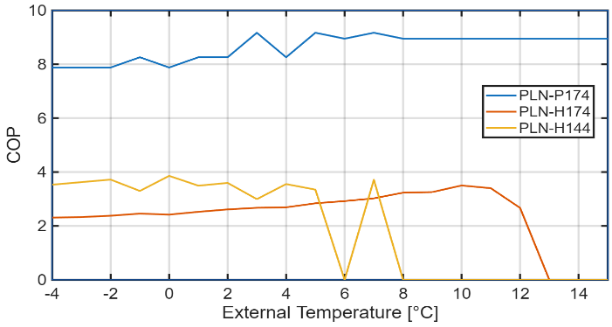

Figure 9: COP values of system B units as a function of the external air temperature.

4.2.3. System C

During winter operation, the cold water loop is balanced by the PLN-P114 unit, and the two PLN-P174 machines operate as A-W heat pumps, providing the missing thermal load to the AHUs. By comparing the performance maps of the PLN-P114 unit and the cooling demand of the water loop serving the supermarket refrigeration counters, the ideal operative configuration for the PLN-P114 was identified, and its consequent effect on the hot water circuit serving the AHUs was determined, allowing to evaluate the missing thermal load to maintain the AHUs target temperature. Then, to determine the optimal configuration of the two PLN-P174 machines, the 16 partialization possibilities were compared for each external temperature. Fig. 10 shows the COP trends of the refrigeration machines involved. The COP of the PLN-P114 unit (working in W-W mode) is practically unaffected by the external temperature. In contrast, the two PLN-P174 units (working in A-W mode) have greater efficiency at higher temperatures. The SCOP based on the outdoor temperature data provided by the TRY for the city of London is equal to 5.91.

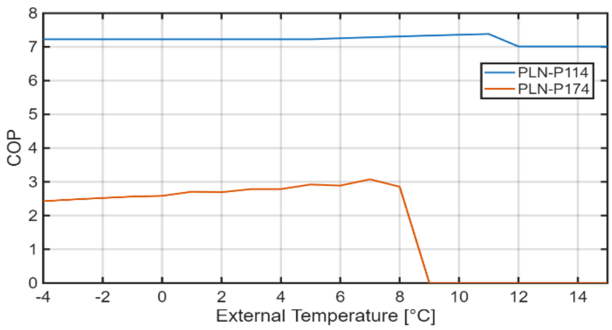

Figure 10: COP values of system C units as a function of the external air temperature.

4.3. Energy comparison

Table 1 presents the summer and winter energy performance coefficients of the considered systems, along with the percentage improvements of the retrofit systems B and C compared to the currently installed system A. Both systems B and C outperform system A. One of the reasons lies in the higher evaporation heat of R290 with respect to R454B (390 kJ/kg vs 180-200 kJ/kg): this difference allows a greater amount of thermal energy exchanged for the same mass of fluid, increasing the volumetric efficiency of the refrigeration cycle. Additionally, R290 performs better at partial loads, especially in multipurpose configurations, due to its higher thermal conductivity (0.0183 W/(m K) vs 0.0130 W/(m K) at 1 atm and 25 °C), which enables more efficient heat exchange within the evaporators and condensers for the same exchanger size. This results in generally higher EER and COP values for the machines in systems B and C compared to system A. A comparison between the seasonal coefficients of systems B and C shows that system C is more efficient than B because of the higher EER and COP values of the PLN-P114 (system C) with respect to the PLN-H144 (system B). In particular, in the winter analysis with system C, the use of the two PLN-P174 units in A-W configuration is severely limited by the presence of the PLN-P114, which, operating in heat recovery mode, can satisfy both the cold water loop load and most of the AHUs demand for the outdoor air temperature values of the TRY analyzed, leading to this configuration being the most efficient, ultimately consuming less electricity. System 3 is, in fact, the only studied configuration in which, during the winter period, the thermal load extracted from the cold water loop has priority. As a result, the cold water loop never needs to be thermally re-stabilized, and the other two heat pumps supply the residual thermal load to the AHUs. In contrast, in the first two systems, in order to meet the AHUs demand which has priority, a thermal power greater than the actual needs is extracted from the cold water loop; as a result, the other machines are needed to thermally re-stabilize the water loop, resulting in less efficient configurations.

Table 1: Energy performance coefficients of the three systems analyzed.

Performance coefficient	System A	System B	System C	System B vs A	System C vs A
SEPR	5	5.65	5.94	+13%	+19%
SCOP	4.81	5.85	5.91	+22%	+23%

4.4. Environmental impact comparison

The values of refrigerant charge for the different refrigeration machines needed to evaluate the TEWI index are reported in Table 2. The annual energy consumption of each refrigeration machine in each studied configuration is shown in Fig. 11. It is worth mentioning that equal units (same model) may have different consumptions depending on how they are operated (number of compressors switched on): see for instance the values of the two PLN-P174 units of system C.

The TEWI values obtained for each system are shown in Table 3, where for systems A and B also the emissions due to the electric consumption of the dry cooler were taken into account. System A has the highest environmental impact due to the use of a refrigerant with a GWP approximately 155 times that of propane. Systems B and C both use the same fluid, but system C contains a higher mass (+12 kg). Nevertheless, despite the greater mass of refrigerant present, system C has a slightly lower TEWI. This is due to the lower electricity consumption of system C compared to system B (226 vs 236 MWh).

Table 2: Mass of refrigerant fluid in the different heat pumps.

Heat pump	PLE-122EC	WLE-274C	PLN-P174	PLN-H174	PLN-H144	PLN-P114
R454B	12 kg	20 kg	/	/	/	/
R290	/	/	25 kg	17 kg	10 kg	14 kg

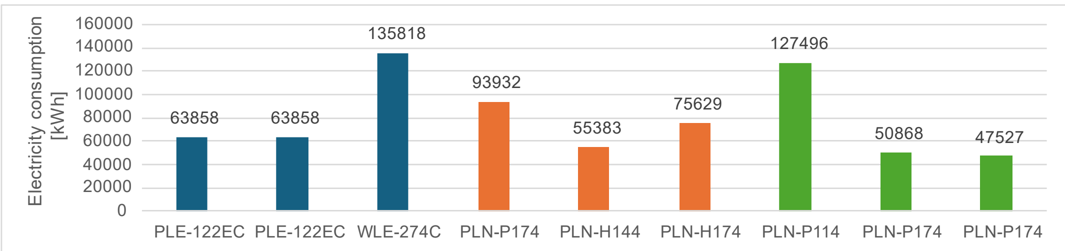

Figure 11: Annual electricity consumption of each device: blue for system A, orange for system B, and green for system C units.

Table 3: TEWI values for the three different systems.

System	TEWI [10^3 kgCO_2eq]
A	564
B	495
C	474

4.5. Economic comparison

The economic analysis was performed by comparing systems A-C to a fourth system (system D) that includes a gas boiler. The presence of a gas boiler leads to a differentiation in the energy sources used during the winter season by system D with respect to system A, resulting in the consumption of $95.41 \cdot 10^3$ kWh of electric energy and $46.4 \cdot 10^3$ Sm^3 of natural gas. An electricity cost of 0.3 €/kWh [16] and a natural gas cost of 1.2 €/Sm^3 [17] were considered. Table 4 shows the relative investment costs with respect to system D, and the annual energy and maintenance costs. The latter costs, provided by the manufacturer service manager, do not vary between the different solutions due to the presence in all cases of flammable refrigerants (A2L or A3), and to the discounts applied to solutions with more than two heat pumps installed.

Table 4: Detailed costs, net present value, and payback period for the three systems with respect to system D.

System	Relative investment cost	Energy cost [€/y]	Maintenance cost [€/y]	NPV [€]	PBP [y]
A	-11%	79407	6000	29594	14
B	+37%	70701	6000	166052	7.82
C	+55%	67767	6000	210932	7.27
D	-	84339	6000	-	-

System C investment is the most expensive due to the refrigeration machines used, but it has the highest annual energy savings, which is a key factor in the long term. Assuming an interest rate of 5%, an annual rate of increase in the cost of energy resources of 5% and an analysis period of 20 years, it is possible to calculate the net present value NPV and the payback period PBP. The obtained values are shown in Table 4, where each system performance is compared to system D. System A does not appear to be economically advantageous compared to the boiler alternative due to the high PBP. Nevertheless, system A resulted to be a valid solution as well, given the lower investment cost of the machines compared to systems B and C. It is worth mentioning that the benchmark boiler is a latest-generation model manufactured by a leading company in the sector, offering performance significantly higher than average. Furthermore, the analysis does not account for any financial incentives that can be obtained when purchasing heat pumps in revamping situations such as the one described, and the environmental impact must also be considered when comparing the boiler solution to systems A-C. Systems B and C, on the other hand, are economically advantageous despite the significantly higher initial investment costs, since they reach short payback periods. In particular, system C has the highest NPV value and a PBP of 7.27 years, making it the most cost-effective option.

5. Conclusions

This article investigated the energy performance, environmental impact, and economic savings of three system solutions to refrigerate the counters and to provide space heating for a supermarket in London. The different configurations are based on a cold water loop to the counters and a hot circuit to the AHUs. The aim was to compare the solutions to identify the most effective one from various perspectives. The analysis showed that system configurations with well-integrated multipurpose machines with R290 as refrigerant can achieve significant improvements in energy efficiency compared to more traditional refrigeration systems. In particular, system C, equipped with three multipurpose refrigeration machines, achieved an average EER 19% higher than the currently installed system and an average COP 23% higher. System B achieved values similar to those of system C, but slightly lower. As a consequence, from an energy perspective, system C configuration proved to be the most efficient, with lower electricity consumption. The environmental impact was assessed using the TEWI index. The calculation of this value revealed that system C has approximately 16% less impact in terms of equivalent CO_2 emissions than system A. System B has a TEWI value slightly higher than C but still lower than A. The economic analysis confirmed the feasibility of these systems and highlighted relatively short investment payback times, proving that the studied configurations are not only more energy-efficient but also economically sustainable and advantageous compared to conventional solutions, such as gas boilers.

In conclusion, the study demonstrates that the implementation of high-efficiency multipurpose heat pumps, combined with strategies for the recovery of waste heat, is a winning choice from an energy, economic, and environmental perspective. The methodological approach developed in this article could be further explored by extending the analysis to different climate scenarios and building types. Another possibility is to evaluate integration with other renewable technologies, such as photovoltaics and energy storage, and to develop a model to account for the impact of defrost cycles on the overall efficiency, particularly significant in locations with harsh, humid climates, such as London. This would further refine optimization strategies and actively contribute to the decarbonization of the sector.

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References

[1] Dongellini M, Natale C, Naldi C, Rossi di Schio E, Valdiserri P, Morini GL. Energy and environmental performance comparison of heat pump systems working with alternative refrigerants. *Appl Sci* 2023;13:7238.
[2] European Commission. European Green Deal – Research & innovation call. Publications Office of the European Union; 2021.
[3] European Commission. Fit for 55: Delivering the EU's 2030 climate target on the way to climate neutrality. Brussels: European Commission; 2021.
[4] European Union. Directive (EU) 2018/2001 of the European Parliament and of the Council on the promotion of the use of energy from renewable sources. *Official Journal of the European Union*; 2018.
[5] European Union. Regulation (EU) No 517/2014 on fluorinated greenhouse gases (F-Gas). *Official Journal of the European Union*; 2014.
[6] Buonomano A, Calise F, Palombo A. Buildings dynamic simulation: Water loop heat pump systems analysis for European climates. *Applied Energy* 2012;**91**:222–234.
[7] Polzot A, Dipasquale C, D'Agaro P, Cortella G. Energy benefit assessment of a water loop heat pump system integrated with a CO2 commercial refrigeration unit. *Energy Procedia* 2017;**123**:36-45.
[8] Coccia G, D'Agaro P, Cortella G, Polonara F, Arteconi A. Demand side management analysis of a supermarket integrated HVAC, refrigeration and water loop heat pump system. *Applied Thermal Engineering* 2019;**152**:543–550.
[9] Huang C, Zong Y, You S, Træholt C, Thorsen JE, Larsen LFS. Economical heat recovery dynamic control and business model for supermarket refrigeration system coupled with district heating system. *Sustainable Energy, Grids and Networks* 2022;**32**:100800.
[10] Everbyte s.r.l. REF.BOX. Available at: https://everbyte.it/it/ref-box/ [Accessed January 2026].
[11] Galletti S.p.A. Available at: www.galletti.com [Accessed January 2026].
[12] European Committee for Standardization (CEN). EN 14825:2016 – Air conditioners, liquid chilling packages and heat pumps, with electrically driven compressors, for space heating and cooling – Testing and rating at part load conditions and calculation of seasonal performance. Brussels: CEN; 2016.
[13] Thermal Energy System Specialists LLC. TRNSYS: A Transient System Simulation Program. Version 18. Madison, WI: TESS; 2018. Available at: https://www.trnsys.com/ [Accessed January 2026].
[14] Department for Energy Security and Net Zero. Greenhouse gas reporting: Conversion factors 2024. London: UK Government; 2024.
[15] Viessmann s.r.l. Available at: www.viessmann.it [Accessed January 2026].
[16] Office for National Statistics. UK electricity prices. Available at: www.ons.gov.uk [Accessed January 2026].
[17] Office of Gas and Electricity Markets (Ofgem). Gas quality. Available at: www.ofgem.gov.uk [Accessed January 2026].