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Energy benefit assessment of a water loop heat pump system integrated with a CO_2 commercial refrigeration unit

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Abstract

The improvement of energy efficiency and the use of environmentally friendly working fluids are key elements of current European policies. Supermarkets are intensive energy consumers and approximately the 40% of their annual energy consumption is for refrigeration. Direct emissions of greenhouse gases associated with the use of high Global Warming Potential (GWP) refrigerants, and the indirect impact on the environment related to high electrical energy consumption, make shopping malls not sustainable buildings. This paper analyses the energy saving potential of integrated supermarket air conditioning and refrigeration systems using a Water Loop Heat Pump system (WLHP). A basic CO_2 booster commercial refrigeration system, applied to cold rooms and display cabinets, is considered. Heat recovery from the refrigeration circuit is performed in the heating season, while in the cooling season a dry cooler on the water loop allows heat rejection to outdoors. The building and all systems are modelled in the Trnsys environment taking into account the hourly weather data, the simulated daily profiles of the cooling and heating load demand and the request from refrigerated food storage equipment. Such a model allows a thorough understanding of the potential for energy savings with heat recovery solutions.

Keywords: CO_2 booster system; control strategies; HVAC; integration; supermarket.

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Nomenclature

AHD	baseline HFC system
t_c	condensing temperature, °C
t_e	evaporating temperature, °C
t_{AH}	auxiliary heater set-point temperature, °C
t_{DC}	dry-cooler set-point temperature, °C
t_{HR}	heat recovery set-point temperature, °C
WLB	operation mode with full cooling/condensation of CO_2 in the heat recovery exchanger
WLD	operation mode with partial cooling/de-superheating of CO_2 in the heat recovery exchanger
WLHP	Water Loop Heat Pump

1. Introduction

Sustainability of shopping malls relies on their energy consumption and related emission of CO_2, which are their main source of environmental impact. They are among the most energy intensive commercial buildings, and it is demonstrated that integration between commercial refrigeration and HVAC systems is one of the most effective ways to reduce their energy consumption. About 50 % of the global energy use in a middle size supermarket is due to refrigeration for display cabinets and cold rooms. Such plants generate a significant amount of heat which is usually rejected outdoors to air or water, sometimes partly recovered for water heating. Solutions where heat recovery is performed meet growing interest, and usually involve recovery of condenser heat for supplying hot water production and HVAC heat demands, such as space heating in winter and reheating in summer after dehumidification. Hot water production asks for high temperature levels, while a lower temperature level (around 40 °C) is sufficient for heating purposes. However, in wintertime, floating condensing pressure is often implemented to take advantage of low outdoor temperature in the view of energy efficiency, thus preventing effective heat recovery. [1, 2].

The Water Loop Heat Pump (WLHP) system has been introduced years ago to take advantage of both low condensing temperature and distributed heating/cooling generation with local control. The system consists of a water loop acting simultaneously as sink or source for a number of reversible water/air heat pumps each serving autonomously a confined space [3 - 5]. The temperature of the water loop is a consequence of the balance between the number of units that are operating in cooling and heating mode. The most effective operation occurs when the two operating modes are balanced, which could happen in the mid seasons or when some zones require heating or cooling all through the year. For this reason, this system is particularly effective where zones exist with high internal loads. Nevertheless, one operating mode is often prevalent and water in the loop has to be cooled or heated.

In recent years, commercial refrigeration has to face new limitations to the use of high GWP refrigerants, which are pushing towards new synthetic fluids or natural ones, like carbon dioxide. CO_2 systems offer significant potentiality for heat recovery in transcritical mode [6 - 8] especially when special control strategies can be adopted [9] following thorough investigations on their energy and environmental performance [10]. However in wintertime subcritical operation is desirable, for the sake of efficiency. For this reason, heat recovery at low temperature like the one achievable through a WLHP scheme could be of great advantage and should be investigated.

In this paper the energy saving potential of integrated supermarket air conditioning and refrigeration systems using a WLHP system is investigated. For the above mentioned reasons a basic CO_2 booster commercial refrigeration system with auxiliary compression and heat recovery is considered.

2. System description

The system sketched in Figure 1 is made up of a traditional WLHP system, where several electric reversible heat pumps provide climate control on the various thermal zones of the building, and a CO_2 transcritical booster system with auxiliary compression and an additional high pressure heat exchanger (water cooled gas cooler/condenser in Figure 1) for heat recovery purposes. Heat recovery from the refrigeration circuit is performed in the heating season by heating up the water loop through the high pressure side heat exchanger of the refrigeration system. The air cooled gas cooler/condenser can be by-passed and all the condensing process can take place in the water cooled condenser. This solution is named WLB in the following. Otherwise, in the water cooled heat exchanger only the de-superheating process takes place and condensation is completed in the air cooled condenser. This solution is named WLD in the following. When the amount of heat available from the refrigeration system is low, the water loop temperature can be increased through an auxiliary heater (air to water heat pump).

In the cooling season, when heat pumps are operating for air conditioning, a dry cooler on the water loop allows heat rejection to outdoors.

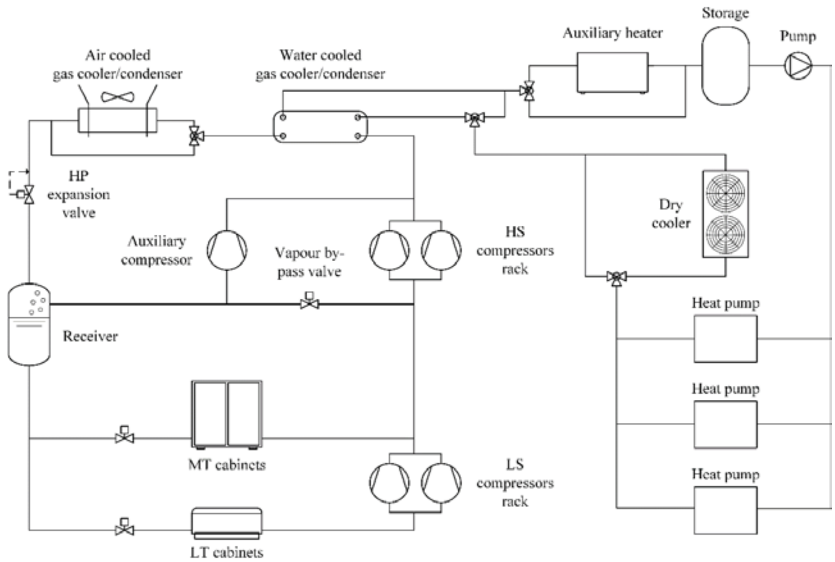


Fig. 1. Schematic of integration between WLHP system and CO_2 transcritical booster system with auxiliary compression.

At wintertime the water loop acts as source for the water to water or water to air heat pumps, and its temperature needs optimisation because it is a crucial factor in the operation of the whole system. High temperature values favour the heat pumps but are of detriment for the refrigeration plant which could take advantage of the low external temperature. Thus it is essential to balance the needs and adequately control the loop temperature. On the contrary, at summertime the water loop temperature should be as low as possible to favour the heat pumps.

Both large roof top heat pumps and medium/small units are employed. The hydrofluoroolefin R1234ze(E) is employed as low-GWP working fluid, in accordance with the use of CO_2 in the commercial refrigeration unit.

The mass flow rate in the loop is kept constant throughout the year, and its value is the sum of the mass flow rate values required by the various heat pumps at rated conditions. Finally, a water reservoir is provided for the sake of thermal storage, to shave peaks and reduce the intervention of the auxiliary heater or dry cooler.

A traditional direct expansion system is chosen as reference for the refrigeration system while several electric air-to-air or air-to-water reversible heat pumps provide the climate control of the zones and DHW production. In the reference solution, named AHD, hydrofluorocarbon refrigerants are employed: R404A in the refrigeration unit, R410A in the heat pumps for climate control and R134a in the heat pump for DHW production.

3. Loads on the HVAC and refrigeration systems

The system has been considered for a food store of 6352 m^2 vending area and the annexed warehouses, services and hallways for other eleven thermal zones and 5411 m^2 . The supermarket is located at the ground floor of a larger modern shopping mall.

The building has been simulated in Trnsys in order to assess the cooling and heating demands of the thermal zones.

It should be pointed out that the sensible and latent interaction between the refrigerated display cabinets and cold rooms with the ambient air in the food store have been taken into account. The total cooling capacity at rated conditions of each display cabinet is adjusted taking into account realistic and time-dependent working conditions in a supermarket [7]. The influence of indoor air temperature and humidity on the sensible and latent fractions of the cooling load are considered as well as the time schedule for auxiliary devices [11]. On the other hand the contributions due to the sensible and latent heat transfer by air infiltration from the open cabinets and to conductive heat transfer through both display cabinets and cold rooms contours are computed as credits to the HVAC system.

1.1. Heating, Cooling and DHW demands

In this study three different locations characterized by mild and warm climate conditions have been considered. The weather files are extrapolated by Meteonorm [12] from data collected from 2000 to 2009. In Table 1, synthetic values of daily and annual external temperature are reported for different climates.

Table 1. Climate conditions considered.

Climate conditions	Climate 1	Climate 2	Climate 3
Annual average temperature [°C]	18.6	16.4	11.6
Average annual temperature fluctuation [K]	13.4	15.8	22.2
Maximum daily temperature fluctuation [K]	10.6	10.4	23.7
Heating degree days [HDD/year]	751	1435	2404

Simulation are carried out with a 15 minutes time step. The monthly heating and cooling demands of the various thermal zones are detailed as an example in Figure 2(a) for climate 2. The heating demand is reported as a negative value, while cooling demand is depicted as a positive value.

The thermal zones show simultaneous heating or cooling demand for most of the year except in May and October, when the internal zones, like the food stores and the warehouses, have a very low request for heating, while some hallways, due to their orientation, have a request for cooling. The same behaviour has been detected for the other investigated climates.

In Figure 2(b) the total heating/cooling demands are compared for the different climates. The domestic hot water usage is estimated at a maximum value of 250 dm^3 per hour during the opening hours. The monthly domestic hot water demand is also depicted in Figure 2.b.

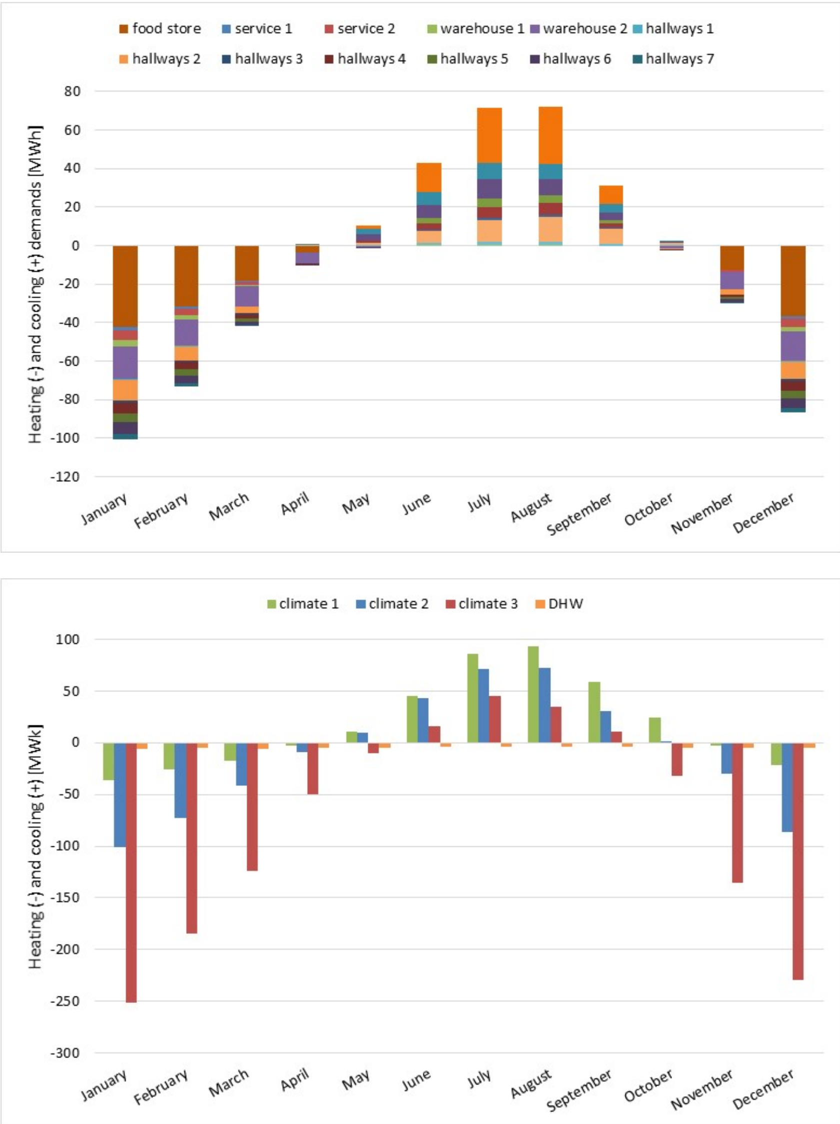
1.2. Refrigeration demand

In the food store there are refrigerated display cabinets for a total length of 208 m and 10 cold rooms from the Medium Temperature (MT) and frozen food display cases for a total length of 86 m and 2 cold rooms from the Low Temperature (LT). The refrigeration plant has a capacity of 140 kW at the Medium Temperature and 28 kW at the Low Temperature evaporative level fixed at -35°C and -8°C respectively. The cooling load profile is evaluated on the basis of a detailed simulation of display cabinets and their interaction with the indoor ambient [7]. The monthly values of cooling load are reported in Figure 3.

4. Model description

The mathematical models of the WLHP system and CO_2 transcritical booster system have been developed in the TRNSYS environment [13], as it allows dynamic simulations of a complex system by easily implementing controls and interconnections between components.

In the following the description of each component and control rule is given in detail.



(a)

(b)

Fig. 2. Heating and cooling demands of the WLHP (a) for different thermal zones in climate 2; (b) for different climates.

1.3. Commercial refrigeration unit

The models of the CO_2 transcritical booster system with auxiliary compression and of a traditional R404A direct expansion system are based on thermodynamic relations. BITZER Software [14] has been used to define the global efficiencies of the compressors as a function of the pressure ratio, whereas the thermodynamic properties have been obtained by using CoolProp libraries [15]. The CO_2 refrigeration system operates at subcritical, transition or transcritical conditions depending on outdoor temperature as shown in Table 2 for the climate conditions considered.

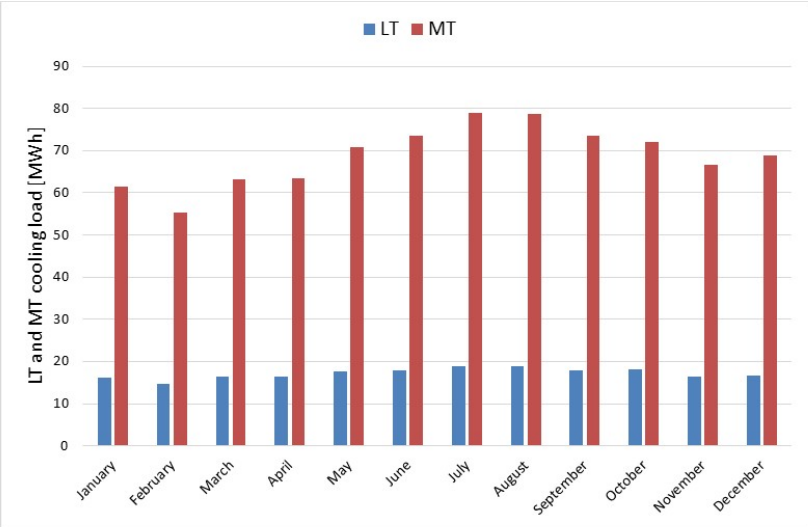


Fig. 3. Cooling load on LT and MT evaporators of the refrigeration system

Table 2. Percent time of operation modes of the commercial refrigeration unit at the climate conditions considered

Climate	Climate 1	Climate 2	Climate 3
Subcritical [%]	60	68	81
Transition [%]	28	25	14
Transcritical [%]	12	7	5

As for the booster system in transcritical mode, the gas cooler discharge pressure and the intermediate pressure at the liquid receiver are optimized to maximize the COP. The correlations developed by the authors for this specific design cycle and the details on the head pressure control in transition operation can be found in [7]. The values of the main design parameters considered for the simulations of the commercial refrigeration unit are listed in Table 3.

Table 3. Main design parameters for the commercial refrigeration unit

Parameter	**Value**
LT Evaporating temperature	-35 °C
MT Evaporating temperature	-8 °C
Superheating at evaporators	5 K
Superheating in the suction lines	5 K
ΔT approach of the condense/gas cooler	3 K
Minimum condensing temperature	8 °C
Liquid receiver pressure (in subcritical operation)	3.8 MPa
ΔT approach of heat recovery	5 K

1.4. Heat pumps

In-house models for the heat pumps involved in the system and in the reference solution have been developed. Vapor compression cycles for different working fluids have been implemented in Trnsys linked to CoolProp libraries. The global efficiencies of the compressors are obtained as a function of the pressure ratio by using BITZER Software [14] for the HFC heat pumps and Frascold Software [16] for the HFO heat pumps. More details on compressors selected in all the investigated solution and the correlations of the global efficiency can be found in [17].

The effect of defrosting has been taken into account through a COP reduction for the air/air heat pumps in the reference solution. The values of the main design parameters considered for the simulations of the heat pumps are listed in Table 4.

Table 4. Main design parameters for the heat pumps

	WLB	Reference (AHD)	
Parameter	R1234ze(E)	R410A	R134a (DHW)
Useful superheating [K]	4.0	4.0	4.0
Superheating in the suction lines [K]	4.0	4.0	4.0
Subcooling in heating mode [K]	3.5	3.5	3.5
Subcooling in cooling mode [K]	2.0	2.0	-
Approach temperature of the source heat exchanger [K]	5.0	10.0	10.0
Approach temperature of the load heat exchanger [K]	5.0	5.0	5.0
Minimum condensing temperature in cooling mode [°C]	25.0	25.0	-

Simulations have been carried out and correlations for estimation of COP and EER have been derived as a function of the evaporating and condensing temperatures respectively. As an example the correlation for the R1234ze(E) heat pumps are:

$$COP_{\mathrm{R1234ze(E)}} = 0.12 \cdot t_e + 3.10 \tag{1}$$

$$EER_{\mathrm{R1234ze(E)}} = -0.18 \cdot t_c + 9.44 \tag{2}$$

For each heat pump in the water loop, the electric consumption is estimated as well as the heat exchanged with the water loop in cooling or heating mode. Once the water return temperature is calculated, the heat recovery from the refrigeration system and the auxiliary devices of the water loop are activated by the following control strategy:

- when the water loop temperature drops below a heating set-point temperature (t_{HR}), the heat recovery from refrigeration is activated to heat the water loop;
- when the water loop temperature drops below a second heating set-point temperature (t_{AH}), lower than t_{HR}, the auxiliary heater is activated;
- when the water loop temperature rises to a cooling set-point temperature (t_{DC}) the dry-cooler is activated to cool the water loop in accordance with the outdoor temperature and the dry-cooler approach temperature.

5. Results

1.5. Set-point temperatures

A preliminary parametric analysis of the set-point temperatures has been performed for WLB and WLD systems. The heat recovery set-point temperature is fixed at 20 °C while the auxiliary heater set-point temperature t_{AH} is varied from 3 °C to 15 °C. The dry-cooler set-point temperature t_{DC} ranges from 20 °C to 25 °C. As an example, for the climate 2, figure 4 makes a comparison in terms of annual energy consumption of the overall system among the solutions WLB and WLD with the different set-point temperatures above mentioned. It can be noticed that the annual energy consumption is only slightly dependent on t_{AH} while the influence of the dry-cooler set-point temperature t_{DC} is more significant especially with the WLB solution. Same qualitative considerations can be extended to all climates considered with slight differences on the optimal set.

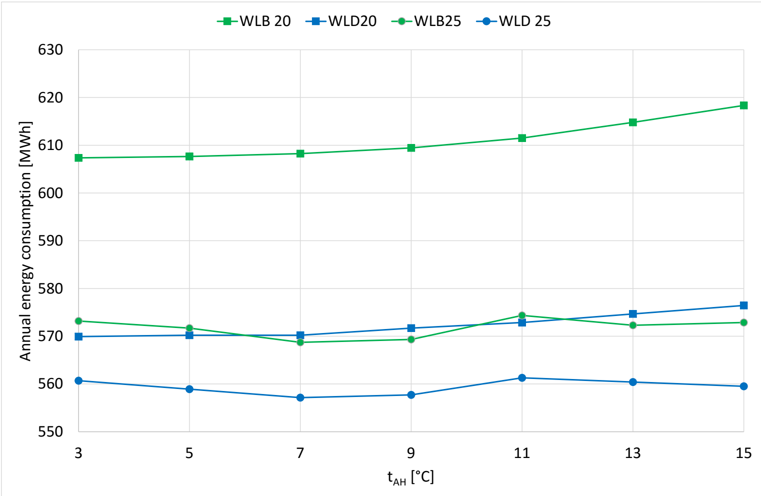


Fig. 4. Parametric analysis in terms of annual energy consumption of the overall system for climate 2 Influence of t_{AH} and t_{DC}

At all the investigated locations, the optimal value of the dry-cooler setpoint temperature is 25 °C. This means that at summertime reducing the temperature of the loop by the dry-cooler requires more energy than what could be saved at the heat pumps.

1.6. Global energy consumption

The global energy consumption for the investigated cases is compared in Figure 5. HVAC indicates the electric energy consumption of heat pumps which provide climate control on the thermal zones, DHW indicates the electric energy consumption of HFC heat pump for DHW production in the baseline solution AHD. In WLB and WLD systems the DHW demand is satisfied directly by heat recovery from the refrigeration unit through a high pressure heat exchanger. Auxiliary devices in the water loop solutions include the auxiliary heater, the dry cooler and the circulator pumps.

The water loop heat pump solution with a CO_2 booster as refrigeration system shows to be an effective solution.

In general it leads to a lower energy consumption for the HVAC, which is much more evident at cold climate conditions. At all climates the refrigeration plant with the WLD scheme uses less energy than the baseline, while the WLB scheme requires higher values, even higher (climates 1 and 2) than the baseline. This is due to the fact that refrigerant temperature at the high pressure side has to be kept high in favour of heat recovery.

At climate 3, the coldest one, the energy savings which can be achieved with the WLB solution is equal to 9.4 % over the baseline, 4.6 % in climate 2 with the WLD system and 2.9 % in climate 1 with the WLD system. It appears that at cold climate conditions it is effective to perform full heat recovery from the refrigeration plant. At such climate, the solution with partial heat recovery is less effective due to the high consumption of the auxiliary heater.

1.7. Water loop temperature

The temperatures profiles of the water loop at the return from the heat pumps are depicted in Figure 5, for the two different solutions WLB and WLD. The solutions with the lowest annual energy consumption (WLD for climates 1 and 2, and WLB for climate 3) are in thick lines. At mild climates it is preferable to allow heat recovery only from de-superheating, with some support from the auxiliary heater if the case. On the contrary, at colder climate full heat recovery is needed, because the water loop temperature remains at low levels.

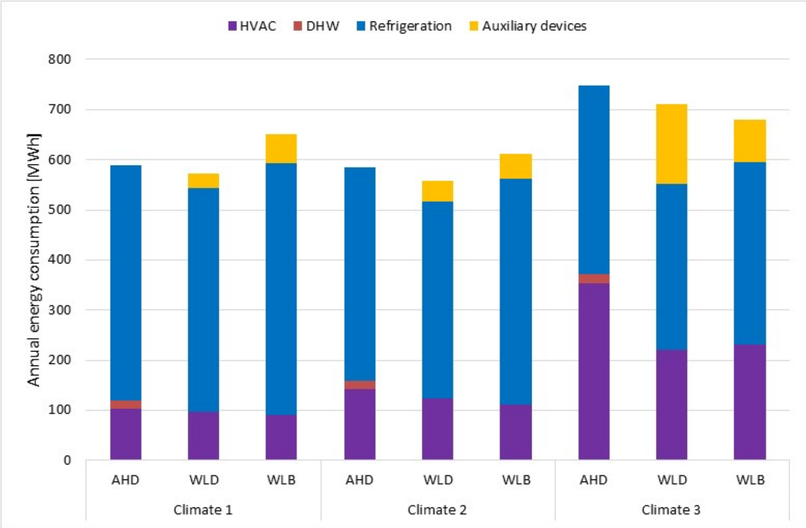


Fig. 5. Global energy consumption of the investigated systems.

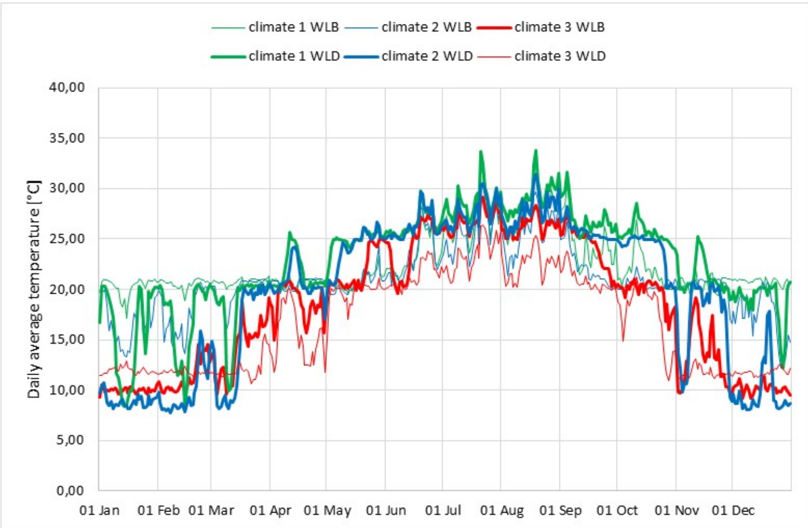


Fig. 6. Daily average temperature of the water loop for configurations WLB and WLD. In thick lines the solutions with the lowest annual energy consumption at each climate.

6. Conclusions

A model has been set up for a Water Loop Heat Pump system integrated with a CO_2 commercial refrigeration plant. This model allows a thorough understanding of the performance of the whole system at various operating conditions, and can provide directions for choosing the most suitable configuration of the refrigeration and HVAC plants in supermarkets and shopping malls. Low-GWP refrigerants have been considered (CO_2 for refrigeration, HFOs for heat pumps). The building considered, with its thermal loads, does not lead to the best exploitation of WLHP advantages occurring when both cooling and heating loads are required. However it appears that the solution with WLHP and heat recovery is more effective than a classical HFC solution with separate plants and heat pumps for heating purposes. It has to be expected that with more favourable heating/cooling load distributions the system is even much more effective, thus confirming that low-GWP solutions are already available provided heat recovery is performed and the comparison made on a global system level. For this reason a comprehensive investigation through modelling is necessary.

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