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Demand coverage and energy savings by combined CO_2 refrigeration system and HVAC in supermarkets

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ABSTRACT

Transcritical CO_2 refrigeration systems for supermarkets are more and more integrated to DHW production, space cooling & heating. In CO_2 integrated solutions, the thermal loads of supermarket buildings are fully covered by the refrigeration system, but advantages in terms of total investment and running costs need to be carefully estimated. They are strongly dependent on several factors among which the quality of the building envelope, the climate, the ratio between HVAC demand and the refrigeration load. A model based on TRNSYS and in-house types, validated with field data from a fully instrumented plant in a supermarket located in Northern Italy, is used to investigate the integration as a function of the thermal loads. The size of the refrigeration unit is kept constant, four different sales area and six climate zones are considered. The full coverage of space cooling by the CRU is reached on the whole Italian territory, whereas the coverage of DHW production and space heating is guaranteed up to 90% and 83% respectively in the worst case and up to 99.3% and 100% in the most favourable case.

Keywords: Carbon dioxide, Transcritical booster system, Commercial refrigeration, Heat recovery, Modelling.

1. INTRODUCTION

In the last decades, CO_2 plants have spread in food retail stores even at warm climatic zones thanks to the higher performance reached by improved technical solutions, such as parallel compression, overfed evaporators and compression work recovery. Furthermore, the high discharge temperature of the transcritical carbon dioxide cycle has been turn into an advantage by recovering the waste heat in favour of space heating and DHW production in the supermarket building (Sawalha, 2013; D'Agaro et al., 2018; Karampour and Sawalha, 2018) and even to supply the district heating network (Giunta and Sawalha, 2021). Lately, in the view of reducing the investment cost and increasing the compactness of the system, the integration has been extended also to space cooling. "CO_2 only" or "all-in-one" vapor-compression units providing refrigeration and space cooling & heating to the supermarkets located in South Europe have been successfully proposed and monitored on the field within the European Multipack project (Azzolin et al. 2021, Tosato et al. 2020).

The coverage of HVAC and DHW demands reachable by an integrated system and the energetic suitability are difficult to assess. They depend on many factors, among which the energy performance quality of the building envelope; the ratio between the supermarket area dedicated to refrigerated products and the total selling area; the climatic conditions. In order to investigate this aspect, a booster commercial refrigeration unit sized to cover the peak refrigeration demand of a 1200 m^2 supermarket in Northern Italy has been taken as reference case. Several scenarios have been studied through the dynamic simulation of the interacting components (building thermal behaviour, display cabinets and cold rooms, commercial refrigeration unit), the coverage limits are established as a function of the climate zones and the load ratio; the annual electrical energy use is compared to the solution of separated HVAC and refrigeration systems.

2. INTEGRATED REFRIGERATION SYSTEM

The commercial refrigeration system considered supplies a nominal cooling capacity of 10.8 kW for 14 m of refrigerated display cabinets for the low temperature level (LT) and a nominal cooling capacity of 70.5 kW for 40 m of closed cabinets and 23 m of serve-over or open ones for the medium temperature level (MT). It is a transcritical CO_2 booster system, with a liquid receiver at the intermediate pressure (35 bar) and two evaporating temperatures: -35 °C and -10 °C for the LT and MT applications respectively. Each of the compressor racks, is composed of two semi-hermetic reciprocating compressors; one is a variable-speed type (Bitzer 2KSL-1K, displacement of 2.71 m^3h^{-1} at 1450 rpm, for LS and Bitzer 4JTC-15K, displacement of 9.2 m3h-1 at 1450 rpm, for HS) and the other is an ON/OFF type (Bitzer 2JSL-2K, displacement of 3.48 m3h-1 at 1450 rpm, for LS and Bitzer 4FTC-20K, displacement of 17.8 m3h-1 at 1450 rpm, for HS). The parallel compressor is another variable speed Bitzer 4JTC-15K.

Waste heat can be recovered at two temperature levels at the exit of HS compressors: heat exchanger HR1 recovers high temperature heat for DHW production (70 - 55 °C) while heat exchanger HR2 supplies lower temperature water for space heating purposes (45 - 35 °C). In order to increase the amount of heat recoverable in wintertime, the refrigeration system is forced to operate in transcritical regime, at a p_{HS} value higher than the subcritical one which would be driven by the outdoor temperature.

Furthermore, two additional evaporators are fed by the liquid from the receiver at intermediate pressure: an external evaporator, placed outside, that increases the amount of heat recovered during the heating season, and the evaporator named "AC supply", that provides cooling capacity for the air conditioning system during the cooling season. A parallel compressor removes the flash gas (and eventually the vapor from the AC supply) from the receiver to the high stage (HS) pressure p_{HS} in transcritical operation, which occurs both for outdoor temperature higher than 26°C and at heat recovery operation. At wintertime, the parallel compressor is used to operate the external evaporator and to control its pressure.

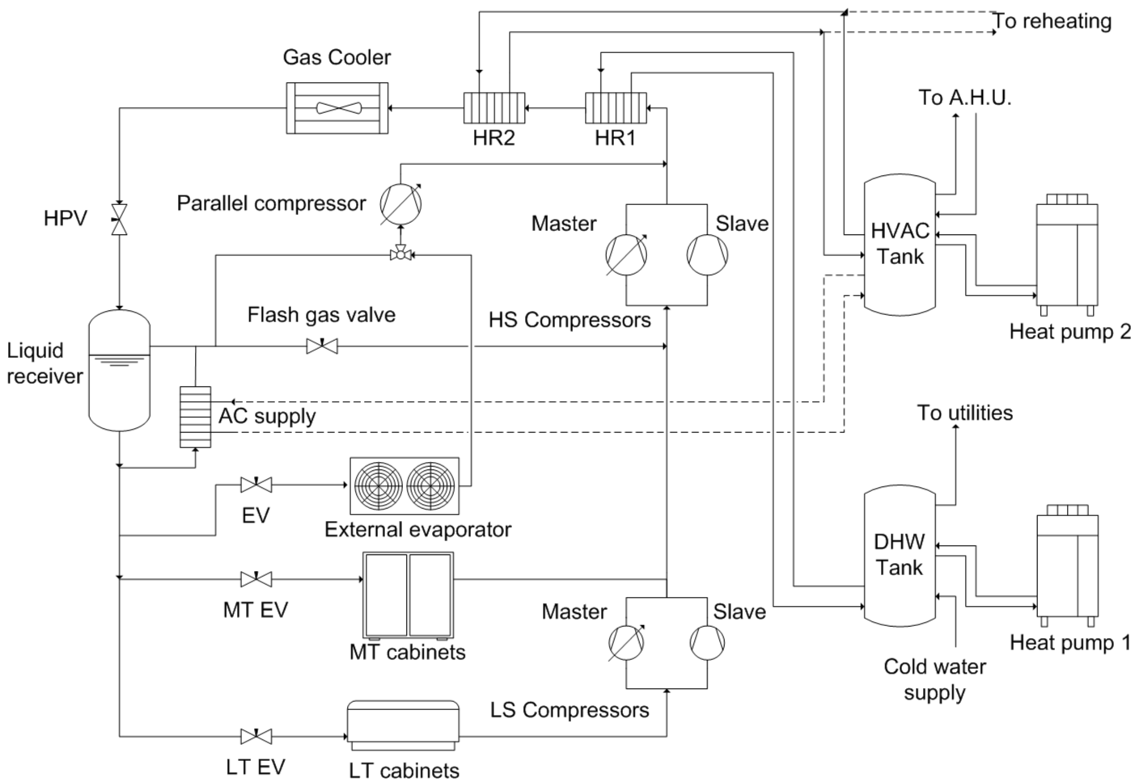

Figure 1: Schematic of the "all-in-one" CO_2 refrigeration system with heat recovery, external evaporator and evaporator for AC supply

2.1 Model

The comprehensive model of the entire refrigeration system, which includes the operation of the display cabinets, cold rooms and of the refrigerating unit, has been implemented in the TRNSYS environment, developing in-house routines for each component. In particular, the sub-hourly cooling load profiles at the MT and LT evaporating levels are predicted by adjusting the cooling capacity at rated condition of each display cabinet in accordance with time-dependent operating conditions in the supermarket as described in detail in Polzot (2017).

In-house routines in the TRNSYS environment allow modelling the refrigerating unit; the thermodynamic and thermophysical properties of the refrigerant are calculated by a link to the CoolProp libraries (Bell et al., 2014). Pressure losses in the cycle are neglected, lamination processes are assumed isenthalpic, the measured superheating values (30 K for LT and 20 K for MT) were assigned, following a conservative approach, entirely to the suction lines. The compressors are described through correlations provided by the manufacturer in accordance with the Standard EN 12900:2013. Specific control rules are implemented to satisfy the heating and cooling demands. In particular, when the heat recovery for space heating (HR2) is active, the high stage pressure p_{HS} is set to 78 bar, i.e. in transcritical operation, otherwise it is driven by the outdoor temperature t_e as follows:

$$\begin{cases} \text{HR2 active} \Rightarrow \; p_{HS} = \text{ 78 [bar] transcritical operation in heating season} \\ \text{no HR2 } \Rightarrow \begin{cases} \text{in transcritical operation: } p_{HS} = \max\,(75;\, 1.75\,(t_e + \Delta t_{ap}) + 22.13) \text{ [bar]} \\ \text{in subcritical operation: } p_{HS} = \max\,(p_{sat}(t_{cond,\min});\; p_{sat}(t_e + \Delta t_{ap} - \Delta t_{sc})) \text{ [bar]} \end{cases} \end{cases} \quad \text{Eq. (1)}$$

where the gas cooler/condenser approach temperature Δt_{ap} is set equal to 4K; the minimum condensing temperature $t_{cond,\min}$ is set equal to 6 °C and the degree of subcooling at subcritical operation Δt_{sc} is set to 3K. The control rule in summer transcritical operation has been derived through optimization for maximum COP in a previous work D'Agaro et al. (2019a).

The heat recovery is carried out whenever the temperature in the tank (1 m^3 capacity), is lower than the setpoint value (i.e. 70°C for DHW and 45°C for space heating) and the temperature of the water at the exit of the heat recovery exchangers HR1 and HR2 is higher than the temperature of the corresponding tank. In summer the HVAC tank stores refrigerated water from "AC supply" evaporator at 7°C, whereas the HVAC return one is set at 12°C.

In Figure 1, two heat pumps HPs are sketched: HP1 supports the DHW production and a reversible one (HP2) the HVAC demand. Whenever the temperature in the corresponding tanks drops below the limit value, the corresponding heat pump is activated to bring back the set point value. The performance of HPs are reported in D'Agaro et al. (2019b).

3. HVAC AND REFRIGERATION DEMANDS

3.1 Supermarket building envelope

The supermarket is rectangular in plan with a selling area of 1200 m^2 and it is 3.16 meters high. It is at the ground floor of the building sketched in Figure 2; the upper floor is entirely occupied by a gym and a warehouse area is located in the underground. The north-east façade (marked 1) is glazed and shaded by a porch; the external façade (marked 2) is mainly opaque, as just 2% of its surface is glazed. A gallery runs along the two other façades. The model for the building energy simulation, which was realized using multi-zone building Type 56, derives from the outcomes of the EU CommONEnergy project (Antolin et al., 2016). In this study, six locations are considered, each representative of a different Italian climate zone (from the warmest A to the coldest F). The heating degree days (HDD) and cooling degree days (CDD) for each location are reported in Table 1. The weather files used in the simulation of both the commercial refrigeration unit and the building are the Typical Meteorological Year (TMY) which derive from the Meteonorm database (Meteonorm, 2017).

In order to perform a fair comparison among the heating and cooling demand loads on the HVAC system at different climate conditions, the energy performance quality of the building has been kept constant by setting the thermal characteristics of the building envelope in compliance with the "reference building" defined in the Italian National Decree of June 26th, 2015. The prescribed U-values are reported in Table 1 for the main envelope components.

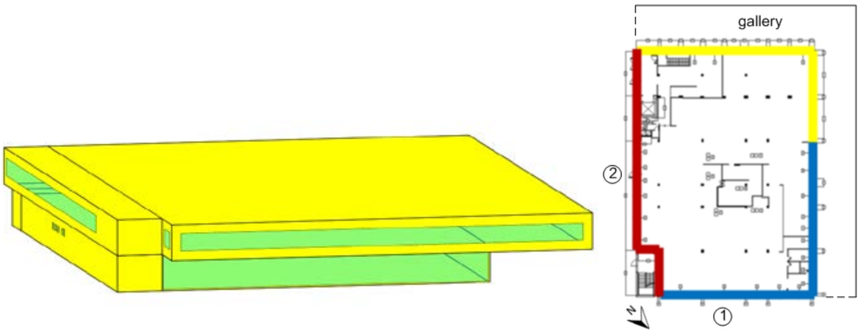

Figure 2: Case study supermarket: 3D sketch of the building (left) and the supermarket plan (right)

Table 1. Climatic region, heating and cooling degree days, transmittance *U* for external walls and glazed surfaces, for reference buildings

Climate	HDD*	CDD**	*U walls* [W/m^{2}K]	*U windows* [W/m^{2}K]	*U basement* [W/m^{2}K]
A	305	894	0.43	3.0	0.44
B	610	380	0.43	3.0	0.44
C	1112	236	0.34	2.2	0.38
D	1599	248	0.29	1.8	0.29
E	1837	395	0.26	1.4	0.26
F	3949	0	0.24	1.1	0.24

* estimated from weather data Meteonorm for 18 °C indoor temperature and outdoor temperature lower than 15 °C – [EUROSTAT]
** estimated from weather data Meteonorm for 21 °C indoor temperature and outdoor temperature higher than 24 °C – [JRC/MARS]

3.2 Refrigeration and HVAC loads estimation

There is a mutual interaction between the refrigerated display cabinets and the indoor environment: the heat and mass transfer occur through the contours of the refrigerated volumes, mainly because of the air and humidity entrainment which takes place in open-fronted display cabinets or during door openings. The sensible and latent contributions from each cold room and display cabinet typology (i.e. temperature class; vertical/horizontal; open/closed; etc.) to the HVAC are dynamically calculated as a function of the indoor air temperature and humidity (Polzot et al., 2016). As an example, in figure 3 the absolute value of the sensible contribution from the all the refrigerated food storage equipment is reported for a winter week (it increases the heating demand) and a summer week (it reduces the cooling demand) for two climates.

The other internal loads are: occupancy in the selling area 0.2 person/m^2; lighting power density 12 W/m^2 and equipment power density 10 W/m^2 ; all of them are modulated according to suitable schedules. As an example, the internal load from lighting is a two step function, whole value during the opening hours and null value during the closing period; the internal loads from occupancy are modulated to reach the maximum at noon, a relative minimum of 30% between 14 p.m. and 15 p.m. and a relative maximum of 60% between 19 p.m. and 20 p.m.

The simulations performed on the building at the various climate zones gave rise to different distributions of heating and cooling loads, which are summarized in terms of monthly energy values in Fig. 4. The load profile for DHW production has been assumed constant for all climates, given the commercial use of the building.

On the other way round the refrigerating cooling capacity of cabinets and cold rooms is influenced by indoor temperature. The indoor temperature is set to 20°C during the heating season and to 24°C during the cooling season, but it is influenced by outdoor conditions when the HVAC is switched off, i.e. in middle seasons and during the supermarket's closing hours. That's the reason why the refrigerating cooling load changes at different climates, as reported in the frequency value distribution plotted in Figure 5.

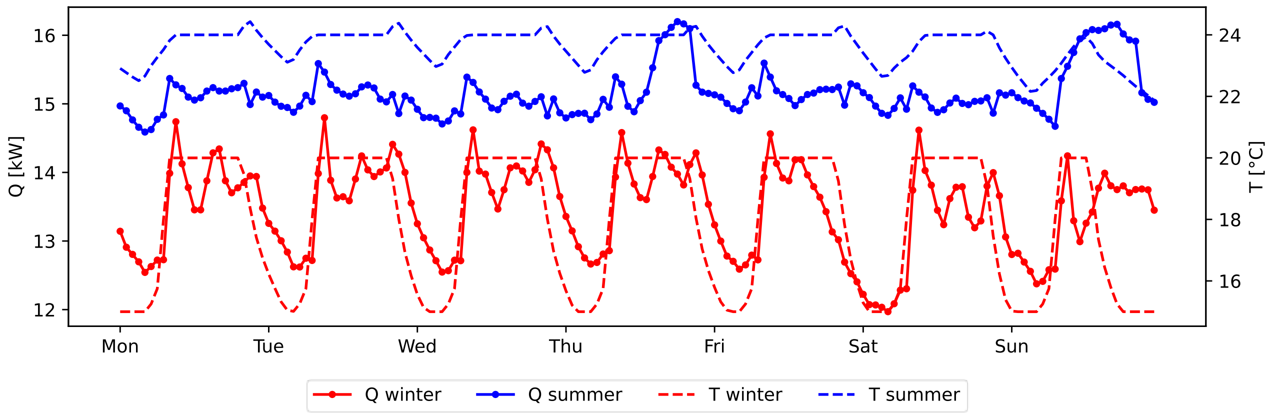

Figure 3: Sensible load (absolute value) from the refrigeration food storage equipment to HVAC and indoor temperature for a winter week and a summer week in climate E.

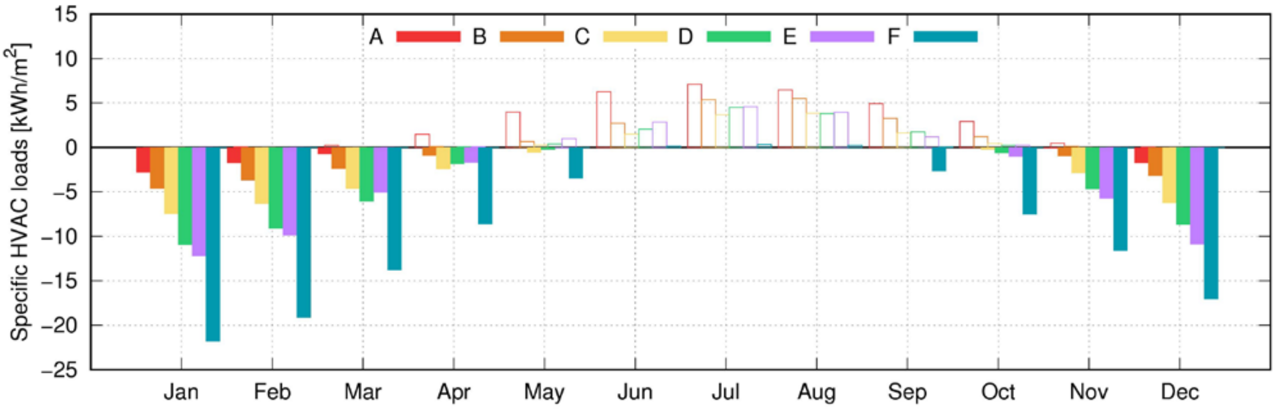

Figure 4: Monthly energy HVAC demand per unit area.

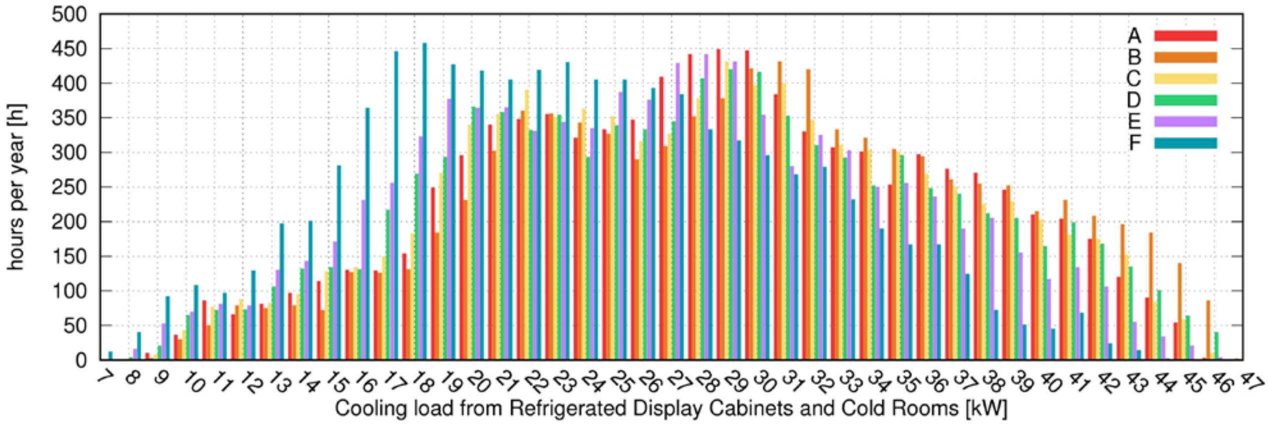

Figure 5: Distribution of the refrigerating cooling capacity required by cabinets and cold rooms for the six selected climatic regions

The energy demands on annual basis are reported in Table 2. The last two columns report the ratio between the space heating and refrigeration demands and between space cooling and refrigeration demands, which is a representative parameter for the all-in-one solution and it is strongly dependent on the climate despite the energy performance quality of the building has been kept constant.

Table 2. Annual energy demand for refrigeration Q_{REF}, for space heating Q''_{HEA} and for space cooling Q''_{AC} per selling area and the relative ratio α and β for the reference case 0.

Climate	Q_{REF} [kWh/y]	Q''_{HEA} [kWh/m²y]	Q''_{AC} [kWh/m²y]	$\alpha = Q_{HEA}/Q_{REF}$	$\beta = Q_{AC}/Q_{REF}$
A	250131	7.3	34.2	3.5 %	16.4 %
B	260636	16.0	19.1	7.4 %	8.8 %
C	247660	31.2	11.5	15.1 %	5.6 %
D	243874	42.8	12.9	21.1 %	6.4 %
E	230700	47.3	14.1	24.6 %	7.3 %
F	206890	109.3	0.8	62.0 %	0.46 %

4. ENERGY USE FOR THE INTEGRATED SOLUTION

4.1 Parameterization

A key question is the coverage of DHW and HVAC demands reachable at different climate conditions with a commercial refrigeration unit sized to cover the peak refrigeration demand. In order to investigate this aspect, a series of tests is done where the refrigeration demand Q_{REF} is kept constant and the HVAC demands are increased and decreased (thus the α and β parameters) to simulate different ratios between the supermarket area dedicated to refrigerated products and the total selling area, or a better/worse energy performance quality of the building. The parameter γ has been defined as follows:

$$\gamma = \alpha/\alpha_0 = Q_{HEA}/Q_{HEA,0} \equiv \beta/\beta_0 = Q_{AC}/Q_{AC,0} \qquad \text{Eq. (2)}$$

where it is assumed that, for a larger building, the increase in heating demand Q_{HEA} and cooling demand Q_{AC} is the same with respect to the reference building at the same climate conditions.

The case studies considered are:

- Case 0: γ = 100 % reference case described in the previous paragraph
- Case 1: γ = 116.7 %
- Case 2: γ = 83.3 %
- Case 3: γ = 66.7 %.

4.2 Load coverage and energy use

Load coverages are reported in Figure 6. Fully coverage of HVAC demands is reached only in case 3 for almost all climates, except for the heating load at the coldest one (F). Coverage of the heating demand by heat recovery is from 89 to 99% in case 0 depending on the climate; it is always higher than 98% in case 3 and lower than 96% in case 1. Cooling load is fully covered on all the Italian territory as the extension of climatic zone A is negligible. Thanks to the limited solar gains and the positive contribution of display cabinets, the coverage of space cooling by the CRU is not critical. The hot water production is more difficult: it is guaranteed up to 90% in the coldest climate of case 1 and to 99.3% in the warmest climate for case 3.

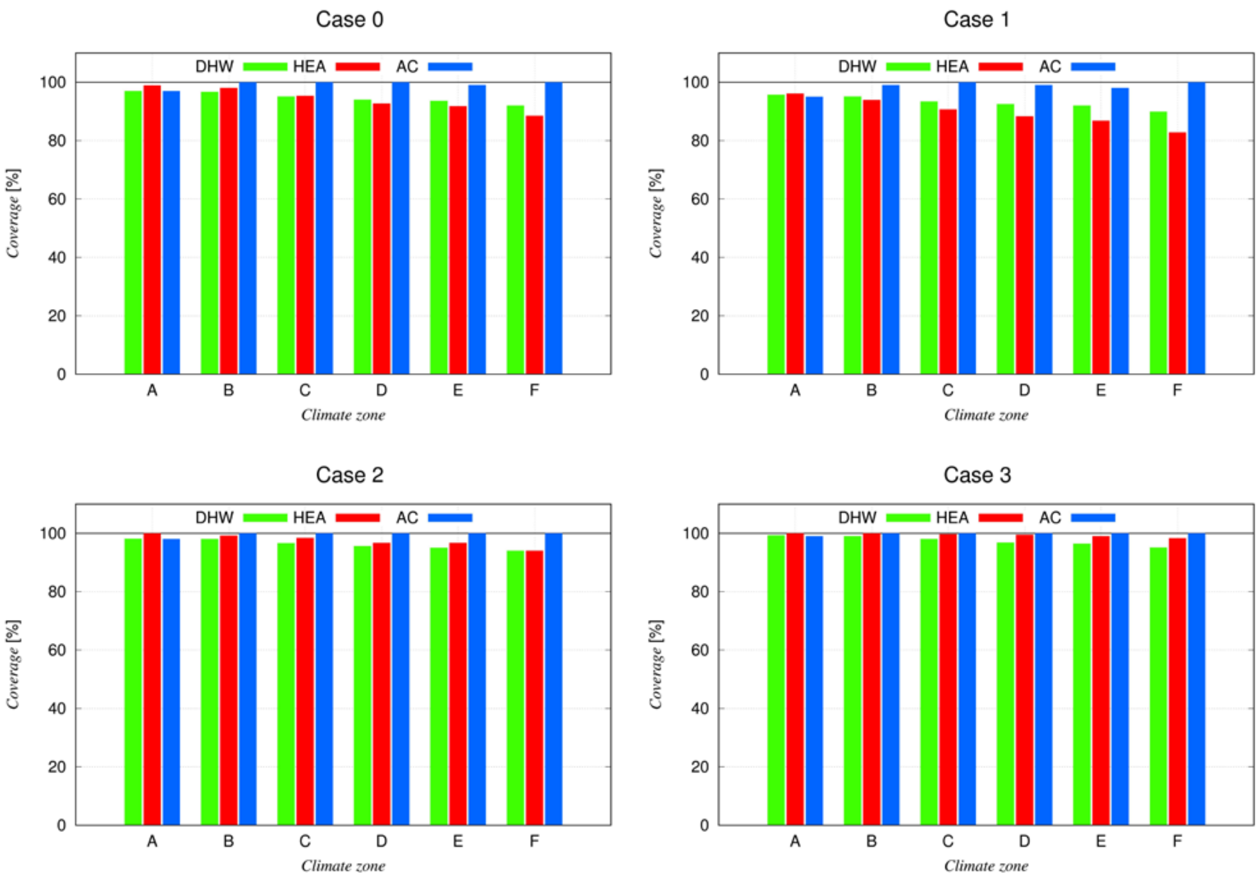

Figure 6: Load coverage of the energy need for space heating (HEA) and cooling (AC) and of the energy demand for DHW production at the six selected climatic regions for the four cases considered.

The electrical energy use is estimated for the base case with separated systems : the booster CRU supplying the refrigeration load and heat pumps supplying the DHW and HVAC ones. The HP energy use for each load is reported in Figure 7 for all cases and climates. Obviously, the annual energy for space cooling decreases and that for space eating increases from the hottest climate A to the coldest one F. In particular in climate F the low demand and the hight EER allowed by favourable summer external temperature keeps the electrical energy use negligible.

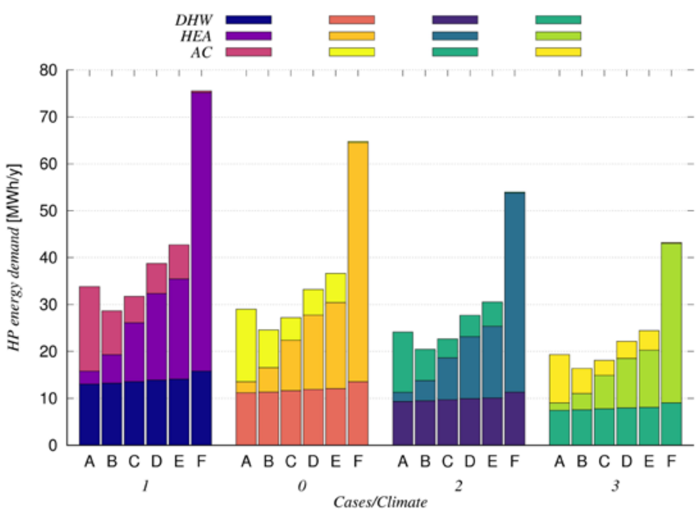

Figure 7: DHW, HEA and AC Heat Pump electrical energy use for separated systems.

In Table 3 the CRU energy use is reported (CRU separated), which changes with the climate but remaining constant for different values of the parameter γ. Table 3 shows also the increment in energy use of the integrated CRU, with respect to CRU separated, for the load coverages discussed above and the savings on total electrical energy use assuming that the residual request is covered by HPs. It can be noticed that the electrical energy use of the integrated CRU always increases with respect to the separated CRU when moving from the hottest to the coldest climate because of the increase in the heating demand and thus a longer transcritical operation, forced by heat recovery. Total energy savings do not necessarily decrease when the integrated refrigerating unit needs more energy: in case 0, case 1 and case 2 saving show a minimum in correspondence of intermediate climate D whereas in case 3 for climates colder than C the integrated system is not convenient as it gives a higher total energy request than the separated solution.

Table 3. Annual energy use for refrigeration CRU in the case of separated systems (base case) and percent variation in CRU energy use and in total electrical energy use for the integrated solutions.

		Case 0		Case 1		Case 2		Case 3	
Climate	CRU separated [kWh/y]	CRU integrated [%]	Total energy [%]	CRU integrated [%]	Total energy [%]	CRU integrated [%]	Total energy [%]	CRU integrated [%]	Total energy [%]
A	146275	8.3	-9.1	9.1	-10.5	7.2	-7.7	6.2	-6.1
B	138113	10.0	-6.3	10.5	-7.8	9.2	-4.7	8.5	-3.0
C	127748	15.3	-4.2	15.7	-6.0	14.7	-2.2	13.8	-0.2
D	123123	21.0	-3.4	21.6	-5.4	20.1	-1.3	19.2	1.3
E	120006	23.2	-4.0	24.0	-5.8	22.6	-1.5	21.4	1.2
F	90774	51.3	-6.8	52.4	-9.1	49.4	-3.8	47.9	1.0

5. CONCLUSIONS

This simulation aims to identify possible energy effective solutions for an all-in-one system allowing to supply refrigeration as well as space heating&cooling and DHW production to a benchmark commercial premise.

It has been found that a unit intended to supply the design refrigerating cooling load to display cabinets and cold rooms can effectively cover also space cooling needs for summer, while space heating and hot water can be partially covered. Almost full coverage is met just in case 3, i.e. when the selling area surface is the lowest of the ones investigated, but not for the coldest climate conditions.

When the heating demand increases (either for colder climate conditions or for larger selling areas) the integrated refrigerating unit needs more energy but not necessarily savings decrease. They show maximum values for the hottest and coldest climates, minimum values for the intermediate climate. When looking at the ratio between refrigeration and HVAC demands (influence of the selling area), the lowest selling area leads to the highest coverage but to worse energy performance.

The results are strongly influenced by a delicate balance between the performance of the refrigerating unit and heat pumps, as well as by the ratio between HVAC and refrigeration demands, in combination with the climate conditions. This leads to underline the need for a thorough prediction of the energy use of various system configurations, to allow the highest reduction in GHG emissions.

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NOMENCLATURE

AC	Air Conditioning	LT	Low Temperature
CDD	Cooling Degree Days	MT	Medium Temperature
COP	Coefficient Of Performance	Q	annual energy need [kWh y^{-1}]
CRU	Commercial Refrigeration Unit	Q”	annual energy need per vending area [kWh m^{-2} y^{-1}]
DHW	Domestic Hot Water	RDC	Refrigerated Display Cabinets
HDD	Heating Degree Days	REF	Refrigeration
HEA	Space Heating	t	Temperature [°C]
HP	Heat Pump	U	Reference U-value [W/m^2K]
HR	Heat recovery	α	ratio between heating and refrigeration load
HS	High Stage	β	ratio between cooling and refrigeration load
LS	Low Stage	γ	parameter defined in Eq.2

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