SEASONAL ENERGY EFFICIENCY RATIO FOR REMOTE CONDENSING UNITS IN COMMERCIAL REFRIGERATION SYSTEMS

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Abstract

Different technological solutions are nowadays available to improve energy efficiency in centralised commercial refrigeration packs. Energy assessment of possible solutions is nowadays performed with different computational tools and comparison between alternative solutions is often difficult. In this paper, a methodology following the European Standards EN 14825 and EN 13215 is proposed for discussion.

Sensitivity analysis with respect to climatic categorisation of European and Mediterranean Countries is presented. The accuracy of the Seasonal Energy Performance Ratio (SEPR) estimations in comparison to traditional hourly based calculation is discussed using both absolute and relative energy figures. The reliability and usefulness of the presented method as a tool for a fair and accurate comparison between different systems is presented. Finally, next steps needed for the method full development are identified and possible further steps meeting the technology trend in commercial refrigeration are highlighted.

Keywords: Carbon Dioxide, Supermarket, Energy Efficiency, Seasonal Performance, Ecolabel

Highlights

- Energy labelling is a useful and simple tool to compare between different solutions
- Extension of the existing standards to remote condensing units is discussed
- The accuracy vs complexity of this simplified approach to SEPR is discussed
- Good capability to compare different systems annual performance is demonstrated

1 Introduction

Since 1992, European Commission has been constantly working towards environment safety, with special emphasis on energy efficiency, providing consumers with information that allows them to choose more efficient products while ensuring the free movement of energy-related products in the European Union.

As recently reported by the EU Commission to the European Parliament and the Council (COM(2015) 345), these objectives remain as relevant as they were more than 20 years ago.

The Eco-design Directive (2009/125/EC) addresses the environmental impact of energy related products 'pushing' the market towards greener products by banning the worst performing ones. The Energy Labelling Directive (2010/30/EU) 'pulls' the market towards more energy efficient products by informing consumers about the energy efficiency and other resources use of products through an energy label, thereby encouraging them to buy more energy efficient product.

Nowadays, refrigeration systems are openly involved in this process limitedly to domestic and professional refrigerated cabinets, which are addressed by specific delegated regulations, i.e. (EU) No 1060/2010 and (EU) 2015/1094 respectively, explicitly excluding those system operating with remote condensing units. However, the interest for remote refrigeration units is increasing amongst commercial refrigeration stakeholders under an energy labelling perspective, as they are more often required to rank their units or installations in terms of energy consumption or CO_2 emissions or to forecast the annual energy consumption of specific sites. As far as standard HFCs based solutions are declining, mainly as a result of the F-gas Regulation (EU No 517/2014), manufacturers are proposing different alternatives, mostly based on the natural fluid Carbon Dioxide. Those alternatives perform differently and must be weighted following the installation boundary conditions, i.e. cooling loads, indoor and outdoor conditions, which often results in a challenging task.

The development of a simple figure to compare different solutions available on the market is becoming a need for contractors and end users and might be a powerful tool for market competition. To this extent, manufacturers will be given the chance to clearly qualify their products by quantifying their energy consumption at rated conditions.

In this paper, a standard methodology is presented, extending the base structure of the European Standards EN 14825, intended for HVAC systems, and EN 13215, conceived for refrigeration condensing units, to centralised commercial refrigeration packs, to be connected to remote condensers and to serve multiple evaporators for display cabinets and cold rooms. In this paper the procedure is applied to medium temperature cabinets and cold rooms, typically representing the major load for the refrigeration system. The main parameter of the simplified procedure are optimized to best fit the result of a reference annual simulation. These parameters include the simplified load and the reference climate profile to be used. Results of the proposed procedure are compared to results obtained from seasonal simulations, critically discussed and settled as basis for further considerations.

2 Simplified SEER calculation

The proposed methodology follows the European Standards EN 14825 and EN 13215, that apply to HVAC systems and condensing units respectively.

The reference standards name to the seasonal performance indicator as Seasonal Coefficient of Performance (SCOP) in the case of heating units (EN 14825), Seasonal Energy Efficiency Ratio (SEER) for conditioning units (EN 14825) and Seasonal Energy Performance Ratio (SEPR) for condensing units (EN 13215:2017). The last definition will be adopted in this work.

When off duty periods are neglected, all these parameters (SCOP, SEER and SEPR) are based on the same calculation procedure:

$$SEPR_s = \frac{Q_s}{W_s} = \frac{\sum_{j=1}^{71} h_j \dot{Q}_J}{\sum_{j=1}^{71} h_j \frac{\dot{Q}_J}{COP_J}} \qquad 1$$

where h_j is the number of hours at the j^{th} temperature T_j during the reference year; $\dot{Q}_j$ is the cooling (for SEER and SEPR) or heating (for SCOP) demand corresponding to an external temperature $T_{ext} = T_j$ and COP_j

is the coefficient of performance corresponding to external temperature equal to T_j. As this paper concerns refrigerating units; $\dot{Q}_j$ will be referred to cooling demand. External temperature is tabulated using 1 °C step, so that $T_1 = -30, T_2 = -29, \ldots, T_{71} = 40$ °C.

The subscript s is used throughout the manuscript to distinguish simplified procedure results from the one obtained by means of a detailed annual simulation, which will be used later on in this work for validation purposes.

The $SEPR_s$ value is then affected by three main elements. The first two are defined by the procedure itself (the reference year and the cooling demand profile, being a function of the external temperature), while only the last one (COP_j) depends on the system.

The following sessions will briefly introduce the approach the existing standards use toward these parameters.

2.1 Temperature bin distribution

EN 14825 is based on four temperature bin profiles. Three of them refer to the winter period and are used for the labelling of heat pumps in three different climatic conditions. The first one is defined as the average European winter condition, which is derived from the standard weather in Strasbourg. The second and the third ones correspond to a warmer and a colder condition, derived from the actual standard temperature profiles in Athene and Helsinki. The fourth temperature bin is used for the air conditioning units working during the summer time and it is unique throughout Europe. No reference to an actual city climatic condition is attributed to this profile.

Standard EN 13215 uses only one temperature profile. While no reference to an existing location is given, the profile seems coherent with a smoothed version of the Strasburg reference year as visible in Figure 1. (U.S. Department of Energy, 2016).

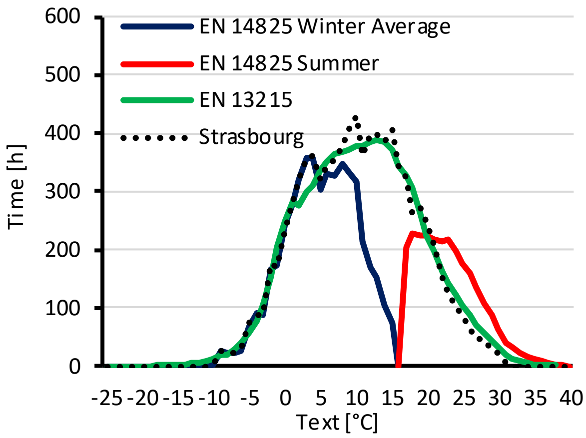

Figure 1: bins distribution in the existing regulation.

2.2 COP

Coefficient of performance should be provided in a tabular form, at 4 prescribed external temperatures (labelled from A for the highest temperature down to D to the lower one). The curve is then approximated by means of interpolation: for values of T_{ext} between T_A and T_D linear interpolation is used, while constant extreme values are used for extrapolation outside this interval.

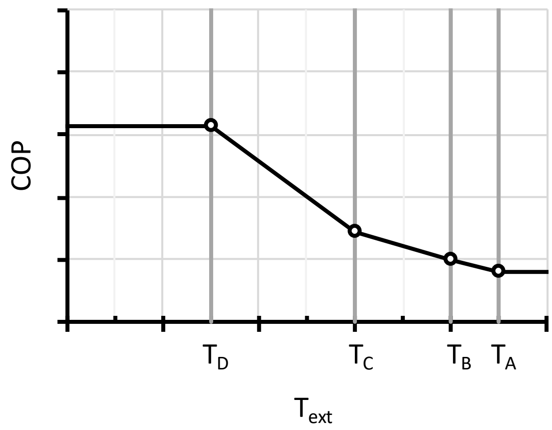

Figure 2: COP curve based on prescribed points T_A... T_D

2.3 Load Profile

Load is modelled as a piecewise linear function of the external temperature. When refrigeration (EN 13215) and air conditioning (EN 14825) are accounted for, load assumes the trend reported in Figure 3.

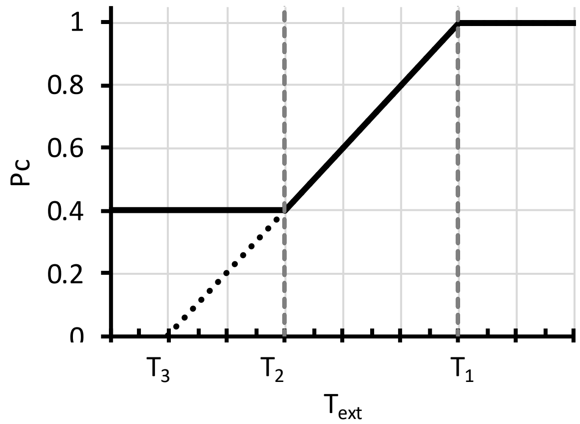

Figure 3: normalized load distribution as function of the external temperature.

The required non dimensional cooling power (Pc) is modelled as a linear function, defined by the full load temperature T_1 and the zero load temperature T_3. As in refrigeration applications, a minimum load is present even under very low external temperature, this is supposed to affect the cooling requirement only in the range between T_1 and T_2, with $T_2>T_3$. As it was done for the COP, outside the interval [T_1, T_2] the cooling load is extrapolated using the nearest extreme value.

The non dimensional cooling power is used to compute the cooling load of each bin, given the nominal power of the system (Pn), rated at T_1:

$$\dot{Q}_j = Pn \cdot Pc(T_j) \quad (2)$$

3 Reference simulations

When dealing with commercial refrigeration systems, seasonal performance evaluation and comparison amongst different layouts, require quasi stationary or transient simulation of the refrigeration system, accounting for the effects of the air conditioning system, the building and the external condition based on the reference climatic year for the considered location. This approach is then adopted in this paper to assess the proposed simplified method accuracy.

The internal temperature of the building is assumed to be controlled by the air conditioning system. The ambient temperature inside the shop, T_{in}, is based on the standard design conditions for summer and winter reported in Riviere (2009), assuming the air conditioning system to be capable of maintaining the actual temperature constantly equal to the set temperature. When the external temperature is higher than $T_{in\,max} = 23$°C, air conditioning system enters the cooling mode, preventing the ambient temperature to further increase. In the same way, the air conditioning system is assumed to maintain the minimum temperature inside the building above $T_{in\,min}^{day} = 21$°C from 8:00 to 19:00, and above $T_{in\,min}^{night} = 12$°C during the remaining time of the day. The internal temperature is assumed to be equal to the external one when $T_{in\,min} < T_{ext} < T_{in\,max}$.

The absolute air humidity inside the store is assumed to be equal to the external one when this results in a relative humidity inside the building lower than 60%. When this value is reached, the air conditioning system starts dehumidifying the ambient air, maintaining the maximum relative humidity to this limit.

Given the ambient temperature, the thermal load of the cabinets is computed. The first step requires the calculation of thermo-hygrometric conditions near the cabinets, in order to account for the so-called cold aisle effect, based on the internal temperature, humidity :

$$\begin{cases} T_{in,c} = 6.5\ 10^{-3} T_{in}^{1.473} RH_{in}^{0.6067} \\ RH_{in,c} = RH_{in,c}(T_{in,c}, r_{in}) \end{cases} \tag{2}$$

This equation sets was derived in [Cecchinato et al. 2010], as a best fit of original data from Orphelin et al. [Orphelin et al. 1999]. These local conditions were then used in order to assess the actual thermal loads of the cabinets:

$$HER = f_{HER}(T_{in,c}, RH_{in,c})\ HER_N \tag{3}$$

Where the correction factor f_{HER} was computed according to the polynomial expression:

$$f_{HER}(T_{in,c}, RH_{in,c}) = a_1 T_{in,c} RH_{in,c} + a_2 T_{in,c} + a_3 RH_{in,c} + a_4 \tag{4}$$

where the values of a_1.. a_4 coefficients depends on the type of the equipment considered [Cecchinato et al. 2010].

During the closing hours, nights blinds use was assumed, reducing then the HER of vertical display cabinets by 45%. The overall cooling power demand is then the sum of the actual extraction rate of each medium temperature equipment:

$$\dot{Q} = HER_{Vertical\ Multi\ Deck} + HER_{Horizontal\ Open\ Wall\ site} + HER_{Horizontal\ Open\ Wall\ island} + HER_{Cold\ Room} \tag{5}$$

The seasonal performance was then obtained as the ratio between the cooling energy over the whole year divided by the overall energy consumption:

$$SEPR_h = \frac{Q_h}{W_h} = \frac{\sum_{i=1}^{8760} \dot{Q}_i\ 1[h]}{\sum_{i=1}^{8760} \frac{\dot{Q}_i\ 1[h]}{COP_i}} \tag{6}$$

Both the cooling energy and the energy consumption are obtained by the summing the corresponding power values over the whole year assuming 1h integration step.

Table 1 reports the supermarket MT display cabinets and cold rooms dimensions and nominal heat extraction rate (HER_N) used. It is worth reminding that, although market trend is going the opposite direction, open cabinets were selected to try the simulation under the most thermo-hygrometrical condition sensitiveness.

Table 1: Medium temperature.

Display Cabinets	**Dimension**	***HER_N***
Vertical Open Multi Deck	40.0 [m]	41.1 [kW]
Horizontal Open, wall site	22.5 [m]	10.2 [kW]
Horizontal Open, island	7.5 [m]	3.0 [kW]
Cold Room		
MT	224 [m^3]	33.0 [kW]
TOTAL		**87.3 [kW]**

4 Optimization of the model parameter

In order to obtain the best set up for simplified SEPR calculation model, an optimization procedure was solved. The main goal of the optimization was to best fit the results obtainable with the annual simulation but using a consistently lighter and simple model. The procedure use as objective the average behaviour over a wide range of COP curves (representing different systems) and locations all over the European and Mediterranean area.

The design vector, the design function and the optimization problem constraints will be presented in the following sections.

4.1 Design vector

The design vector define the dimensionality of the design space. As described in section 2.1 and 2.3, the simplified procedure is defined by the selection of the reference temperature bins, four Temperatures for the COP description (T_A, T_B, T_C, T_D) and three for the Load Profile (T_1, T_2, T_3). The assumption that the maximum load temperature T_1 is equal to highest temperature for the COP curve description T_A was in accordance with existing regulations. The reference locations for the bin profiles were considered part of the design vector, while the number of the profiles z was set as a parameter for the problem and varied from 1 to 5.

The design vector can therefore be expressed by the following:

$$X = \{bin_1 \dots bin_z \; T_A \; T_B \; T_C \; T_D \; T_2 \; T_3\} \quad 7$$

each bin distribution contains the hours at each temperature in the range between -30 and 40 °C:

$$bin_i = [h_{i,j}] \in \mathbb{N}^{71} \quad 8$$

under the constraint $\sum_{j=1}^{71}[h_{i,j}] = 8760$. The design vector results in $70\,z + 6$ degrees of freedom.

Using this approach, the value of every bin $h_{i,j}$ (number of hours at T_j in the i-th distribution) is independent on the neighbours $h_{i,j+1}$ and $h_{i,j-1}$, corresponding to the temperatures T_{j+1} and T_{j-1}, which are 1 °C higher and lower than T_j. Sharp variation between contiguous bins are then numerically possible, but clearly unphysical. To enforce a certain degree of continuity between contiguous bins, reference profile are computed as a linear combination of the actual profiles in 65 European locations (see test location in section 4.2), instead of having a free variable for every bin:

$$bin_i = \sum_{u=1}^{65} x_{i,u}\, bin_u \text{ where } x_i \in \mathbb{R}^{65} \text{ under the constraint } |x_i| = 1 \qquad 9$$

The bias of this linear combination x_i are used in the design vector, that is then rewritten as:

$$X = \{x_1 \dots x_z\ T_A\ T_B\ T_C\ T_D\ T_2\ T_3\} \qquad 10$$

resulting in $65\,z + 6$ degrees of freedom.

4.2 Objective function

The objective function implies the calculation of the seasonal performance using both the simplified figure and the seasonal simulation presented in section 3, for given systems (COP curves) and location sets.

The annual simulation presented in section 3 can be applied straightforward, as it depends only on the COP curve and the provided location climatic profile: $SEPR_H = SEPR_H(loc, COP_v)$.

The assessment of the simplified figure is instead independent of the actual location, but based on the input vector and the COP: $SEPR_S = SEPR_S(X, COP_v)$.

When $z = 1$ just one reference climate is used for the simplified procedure, resulting in an input vector of 65+6 elements: $X = \{x_1\ T_A\ T_B\ T_C\ T_D\ T_2\ T_3\}$. The objective function is designed as the average normalized difference between the results of the simplified procedure and the seasonal simulation, in terms of both $SEPR$ and annual power consumption:

$$f(X) = \frac{|\overline{\Delta SEPR(X)}| + |\overline{\Delta W(X)}|}{2} \qquad 11$$

The average error is evaluated for all the European climatic areas and for different systems according to:

$$\overline{\Delta \mathrm{SEPR}(X)} = \frac{\sum_{u=1}^{U} w_u \sum_{v=1}^{V} \left|1 - \frac{SEPR_S}{SEPR_H}\right|}{V \sum_{u=1}^{U} w_u} \qquad 12$$

$$\overline{\Delta \mathrm{W}(X)} = \frac{\sum_{u=1}^{U} w_u \sum_{v=1}^{V} \left|1 - \frac{W_S}{W_H}\right|}{V \sum_{u=1}^{U} w_u} \qquad 13$$

where $v = 1..V$ is used to take cycle between different COP curves, $u = 1..U$ allows to cycle between different test locations, and w_u is a weight assigned to every location. Weight was introduced to prevent extended and unpopulated areas to have an excessive weight on the overall results. The weight is defined as the product of the area assigned to the u^{th} location and the local population density. Areas are assigned to the test locations applying the Voronoi algorithm to calculate the patches and subtracting the areas covered by seas and oceans. Population densities are assumed coherently to the average population density of the corresponding Country.

In the present work U = 65 location were used as test location, to represent different European climatic areas. Figure 4 reports the locations used to test the procedure, the corresponding patches and population density.

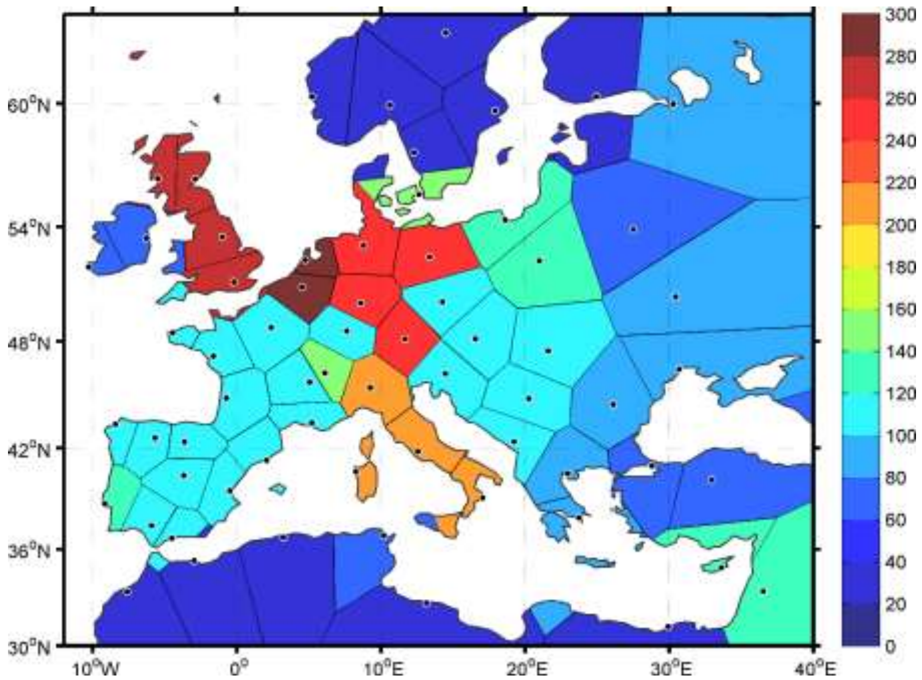

Figure 4: test location and corresponding patches and population density.

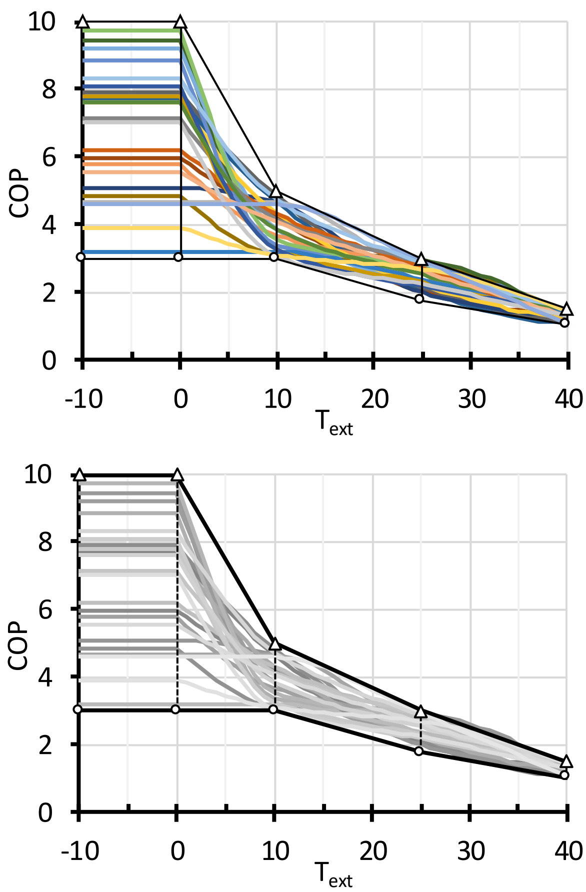

Figure 5: Synthetic random COP curves (only 25 of the 100 curves displayed).

To test the robustness of the method over a wide variety of systems, 100 COP curves are randomly generated. A Latin Hypercube sampling of dimension 4 was used to obtain a set of 100 random four-control-point for the COP curve scaled between the minimum and maximum values of Table2. The four control point curves are then interpolate to smoothed curves using polynomial interpolation (order 3). Curves are thereafter forced to be monotonic as a function of the external temperature, and to be constant below 0°C, to take care of limitation in minimum condensation temperature. A random selection of 25 curves of the validation set is reported in Figure 5 as an example. Despite none of these curve represents the coefficient of performance of a specific real system, credibly every real commercial refrigeration system can stay within these values, therefore the considered curves can efficiently train and tune the procedure.

Table 2: Minimum and maximum COP values used to generate random curves.

Text	COP min	COP max
[°C]	[ ]	[ ]
0	3.00	10.00
10	3.00	5.00
25	1.75	3.00
40	1.00	1.50

For $z > 1$ more than one climate reference profile is provided in the input vector. The above procedure is then repeated for each reference profile, obtaining than more than one estimation for the $SEPR_S$, noted as $SEPR_{S,i}$, with $i = 1..z$. In this case, equations 12-13 is rewritten as follow, in order to use for every test location $loc_u$the most fitting reference bin set:

$$\overline{\Delta\mathrm{SEPR}(X)} = \frac{\sum_{u=1}^{U} w_u \, min\left(\sum_{v=1}^{V}\left|1-\frac{SEPR_{S,i}}{SEPR_H}\right|\right)}{V\sum_{u=1}^{U} w_u} \quad , i = 1..z \qquad 14$$

$$\overline{\Delta\mathrm{W}(X)} = \frac{\sum_{u=1}^{U} w_u \, min\left(\sum_{v=1}^{V}\left|1-\frac{W_{S,i}}{W_H}\right|\right)}{V\sum_{u=1}^{U} w_u} \quad , i = 1..z \qquad 15$$

4.3 Optimization problem statement and solution

The optimization problem is then set as a minimization problem according to the following:

find $X = \{x_1 \dots x_z \; T_A \; T_B \; T_C \; T_D \; T_2 \; T_3\}$

which minimizes $f(X)$ for a given z

under the constraints:

$$T_A > T_B > T_C > T_D$$
$$T_A > T_2 > T_3$$
$$|x_i| = 1, i = 1..z$$

The flow chart of the optimization procedure (neglecting the optimum search algorithm) and the evaluation of the objective function is reported in Figure 6.

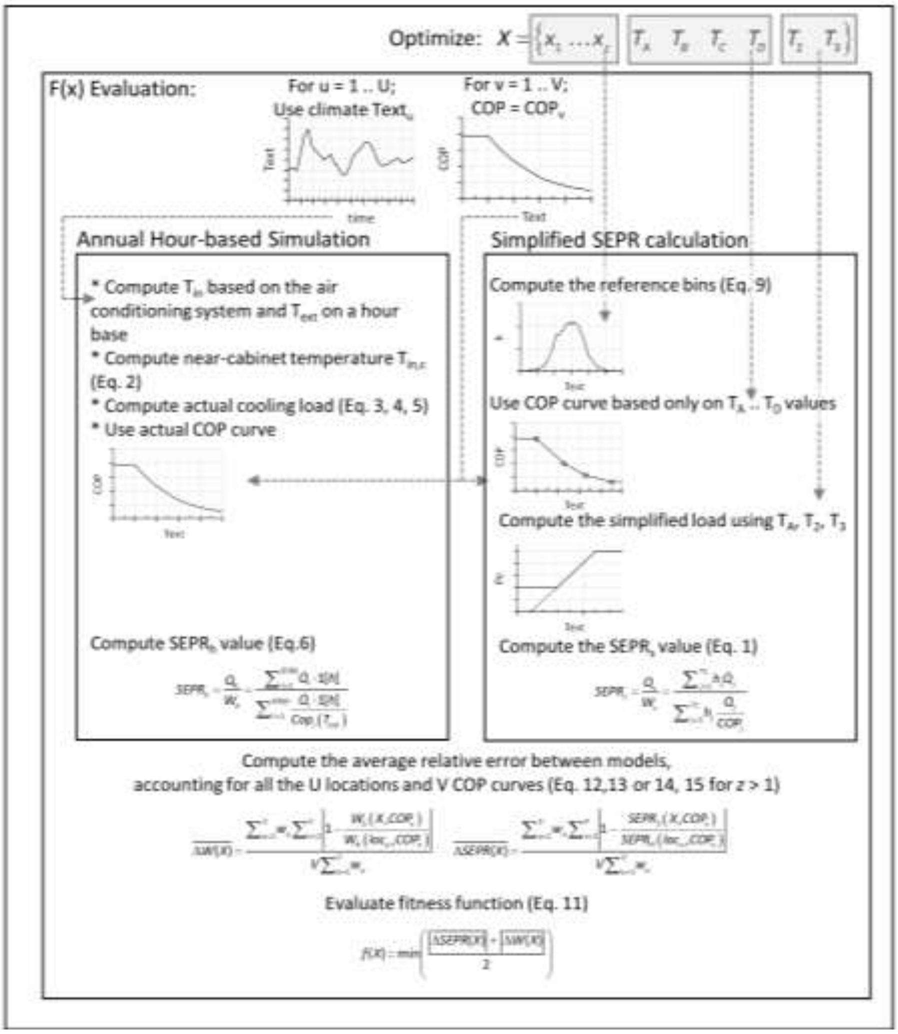

Figure 6: Flow chart of the optimization problem and the fitness function evaluation.

The optimization problem is solved by applying the Particle Swarm Optimization algorithm, modified as proposed in [Shu-Kai et al. 2006].

5 Results

5.1 Sensitivity to reference climatic regions number z

Figure 7 reports the optimum value of the objective function for $z = 1..5$. In Figures 8a-d the climates segmentation of the European validation regions based on different reference climate are illustrated ($z = 1$ is omitted, as in this case all the validation areas belong to the only available reference climate).

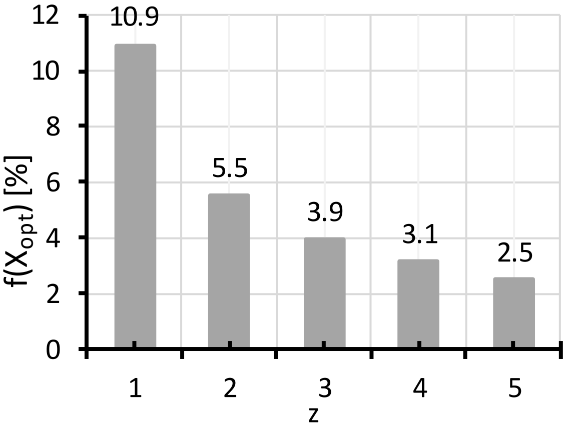

Figure 7: average error as computed by the objective function for different number of climates areas.

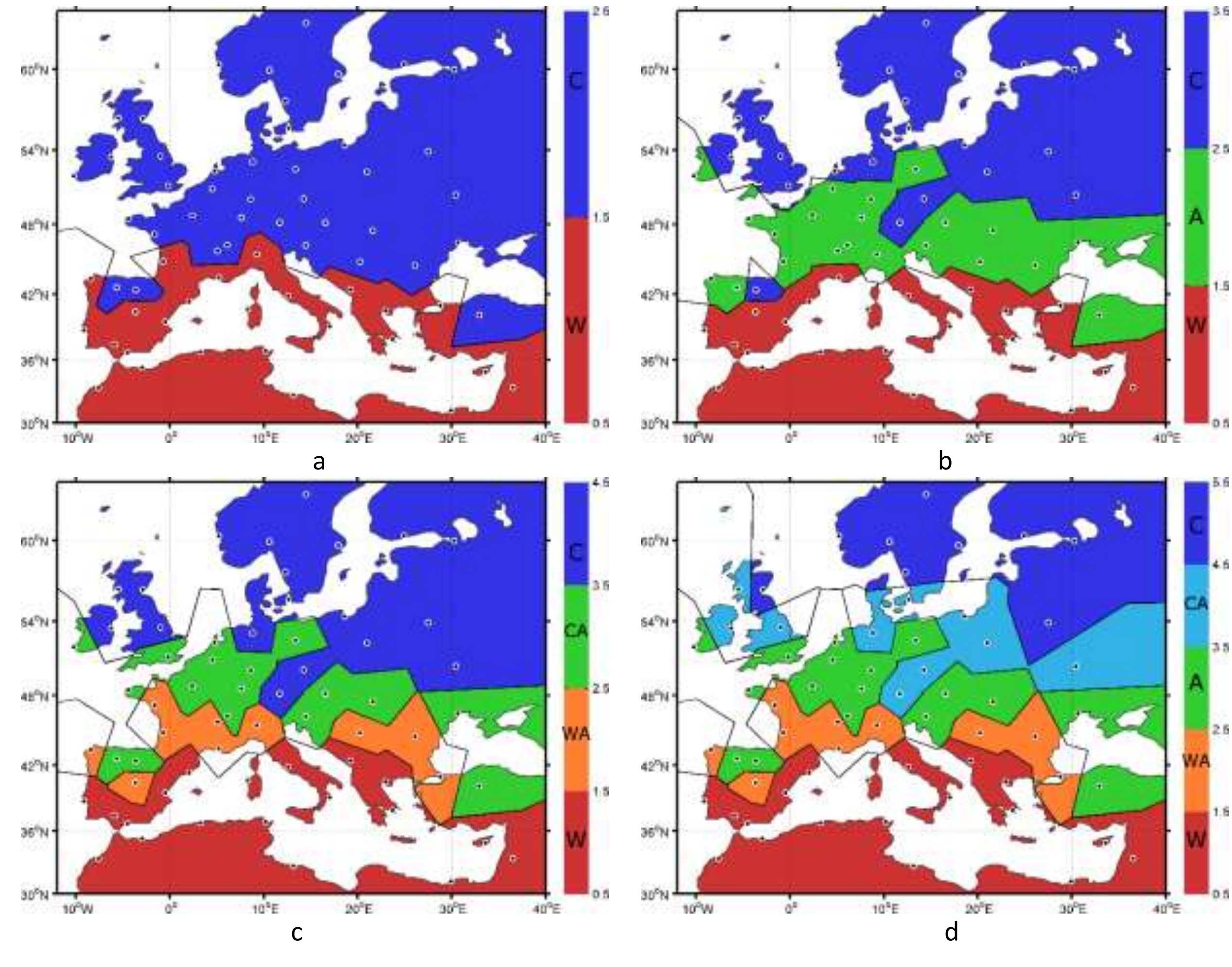

Figure 8: climate segmentation: a) 2 climatic regions model: W – warmer; C – colder; b) 3 climatic regions model: W – warmer; A – average; C – colder; c) 4 climatic regions model: W – warmer; WA – warmer average; CA – colder average; C – colder; d) 5 climatic regions model W – warmer; WA – warmer average; CA – colder average; MC – mildly colder; C – colder.

Resulting areas of influences are quite intuitive, especially when two or three regions are considered. The four climatic zones model basically divides the average climatic area in two sub-areas with only slight modification of the already identified colder and warmer areas. In details, the average climatic region is divided into a warner zone centred on the 46°N, and a colder one, including large part of the Atlantic coast and central Europe. The five climatic regions model divides then the colder area in two zones, separating the continental northern regions from the Scandinavian countries and East Scotland.

The 3 and the 4 regions models can be considered a good compromise between accuracy and complexity. The average error between the simplified and the hour based simulation on the seasonal performance $\overline{\Delta\mathrm{SEPR}(X)}$ is reported in Figures 9 and 10 for the 3 and 4 climatic regions models.

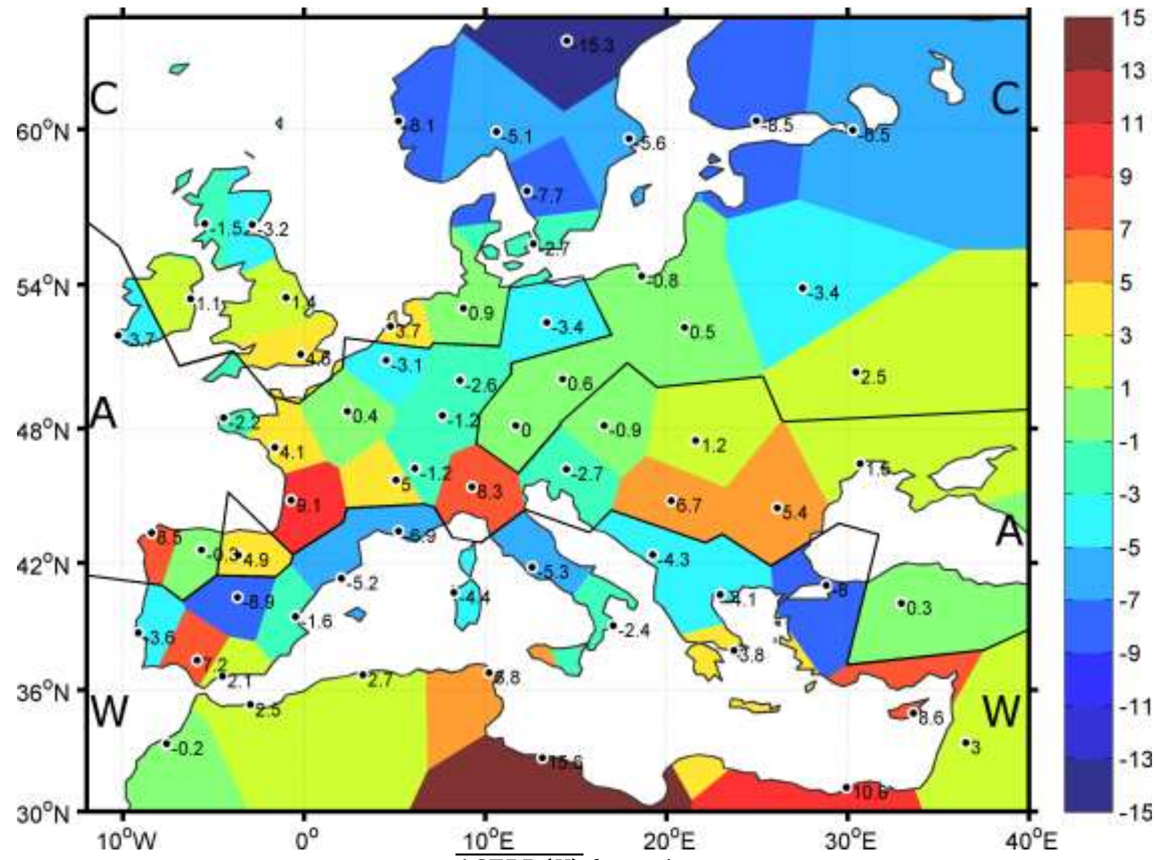

Figure 9: $\overline{\Delta\mathrm{SEPR}(X)}$ for 3 climatic areas.

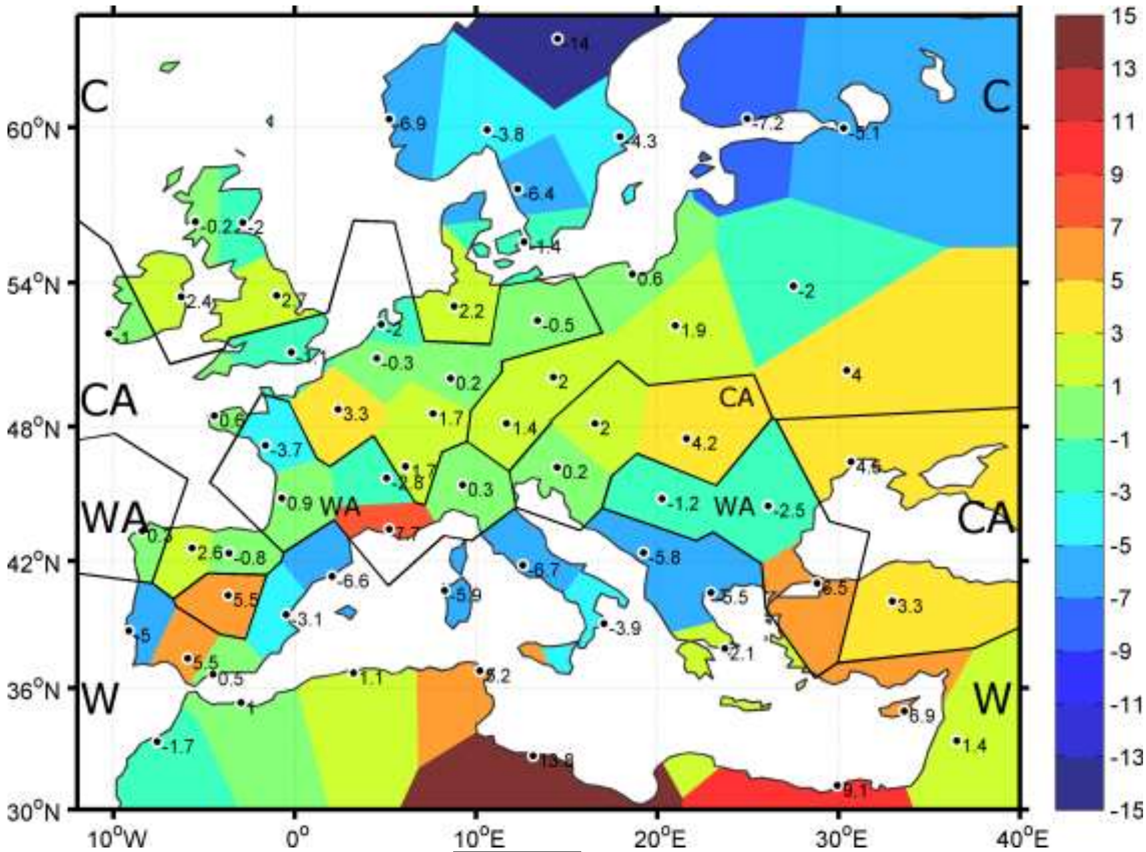

Figure 10: $\overline{\Delta\mathrm{SEPR}(X)}$ for 4 climatic areas.

Both models present large areas on the ±3% band. While the average impact of the fourth climate do not change drastically the overall accuracy of the model (just 0.7% improvements on the average, Figure 7), the presence of two moderate climate profiles instead of one allows a significant local improvement on the regions between 42 and 46°N. In the 3 climates model, none of the reference climates approximate the local weather which is in between the average and the warmer profiles, leading to high positive or negative errors respectively if the average or the warmer profiles are used. Due to this significant local impact, the subsequent sessions will be focused on four climates model.

The temperature bins profiles identified by the 3 and the 4 templates are reported in Figures 11 and 12, while Figure 13 shows the optimum normalized load profile, as defined in Section 2.3. The normalized load appears to be marginally affected by the number of the climatic regions. This is consistent with the study, as the load is mainly affected by the internal temperature and the cabinet types, and not by the climatic area.

The optimal values obtained from the optimization procedure for the four control temperatures T_A … T_D, used for the interpolation of the COP curve, are reported in figure 14. Optimum values result almost evenly distributed in the range 0 – 35°C. As for the normalized load, the optimum values appears to be marginally affected by the number of climate regions. This is reasonable, as the main object of these temperature is the reliable reconstruction of COP curve, which is not directly related to the accuracy in the description of the external conditions.

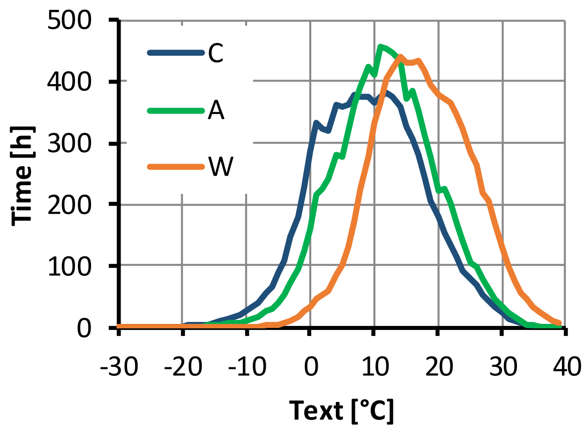

Figure 11: reference 3 climatic region bins profiles.

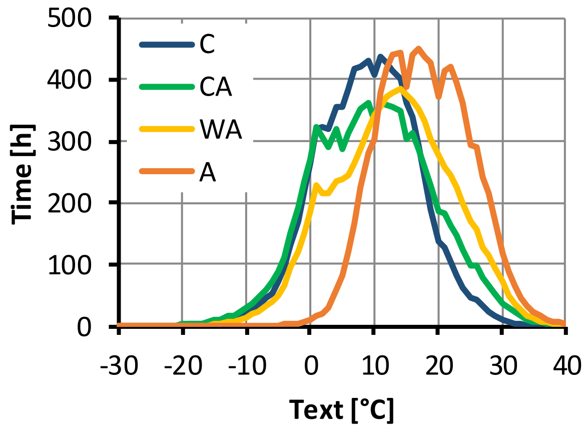

Figure 12: reference 4 climatic region bins profiles.

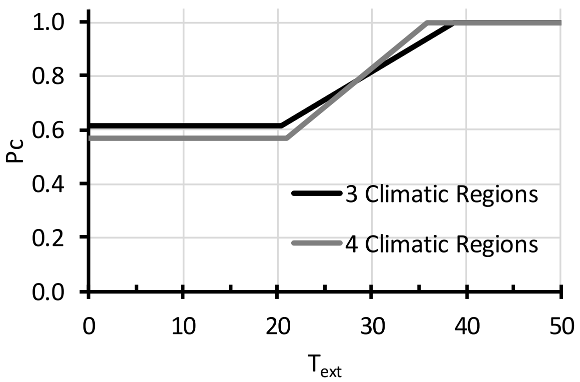

Figure 13: optimum normalized load profile for 3 and 4 climatic regions.

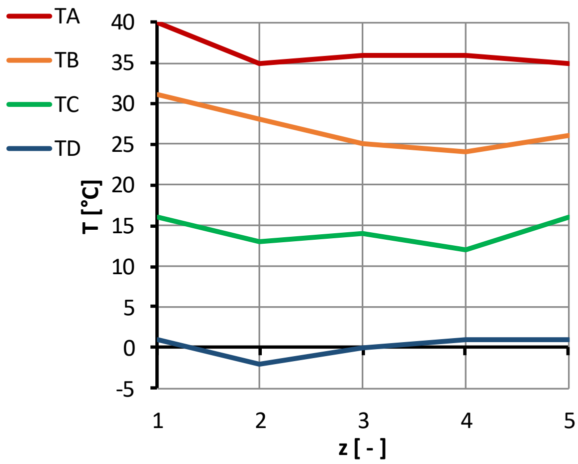

Figure 14: Optimum values of T_A... T_D as a function of the number of the climatic regions z.

5.2 Case study application: quantitative comparison between two systems.

To allow quantitative comparison between systems simulation were carried out for two systems presented in [Sawalha et al. 2015 and 2017] for three location belonging to three different climate areas. The first system [Sawalha et al. 2015] is part of a traditional booster system, integrating medium and low temperature evaporation level into a unique machine: the medium temperature application features a single stage compression, while the low temperature one is served by a low stage compressor whose discharge line enters the suction line of the high stage, medium temperature application, compressor, thus resulting in a double stage compression. A two stage expansion with intermediate vessel allows liquid feeding for both low temperature and medium temperature expansion valves. Pressure at intermediate vessel is controlled by flash vapour venting to high stage compressor; flash vapour expansion provides subcooling, thiugh limited, to the liquid out of the vessel. The HFC system [Sawalha et al. 2017] consists of independent medium temperature and low temperature units. The medium temperature system cools down a heat transfer fluid in indirect loop arrangement, while direct expansion (DX) is applied in low temperature. The low temperature unit is sub-cooled by the medium temperature brine. An indirect water loop rejects heat outdoor via a dry-cooler. In both cases, only MT was considered and related COP values accounted for.

The COP curves for different ambient temperatures are reported in Figure 15, while the resulting SEPR from the hour-simulation and the optimized simplified model is reported for the two systems in Figure 16.

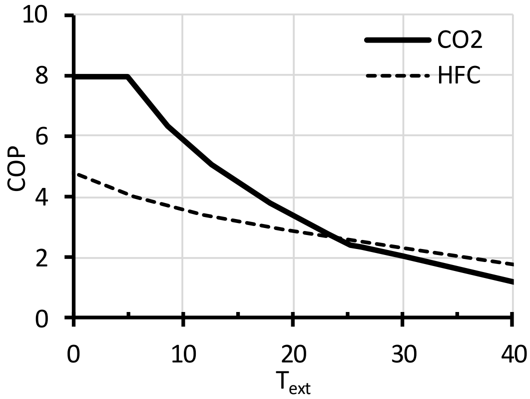

Figure 15: COP curves of the system under comparison

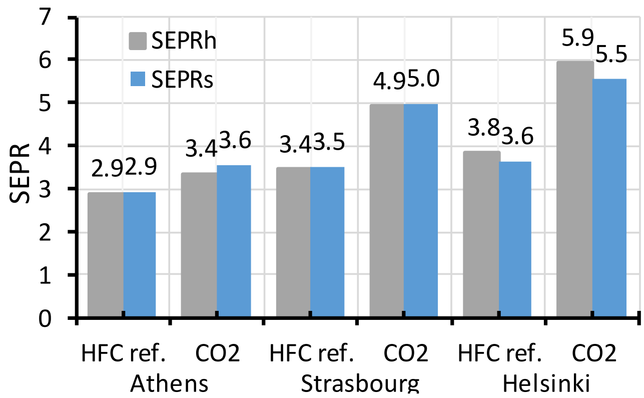

Figure 16: SEPR comparison.

The simplified approach leads to a good absolute estimation of the seasonal performance for Strasbourg and Athens, while differences are greater in the case of Helsinki, in agreement to the average accuracy reported in Figure 10. As predictable from the COP trend, both approaches identify the CO2 transcritical system to be the best performing amongst the two.

Despite the absolute SERP values are interesting to qualify the systems, the comparison amongst them is usually required in early system planning. The normalized energy consumption difference between two systems can be evaluated assuming the most performing as reference, and calculating the relative increase in the annual consumption. In the present case, being CO2 the best solution, this figure can be evaluated as:

$$E = \frac{SEPR_{CO_2}}{SEPR_{HFC}} - 1 \qquad 16$$

The difference estimation obtained from the simplified and the annual approaches are reported in Figure 17. The two approaches lead to very similar results for both Helsinki and Strasburg, while a 5% error is obtained for Athens.

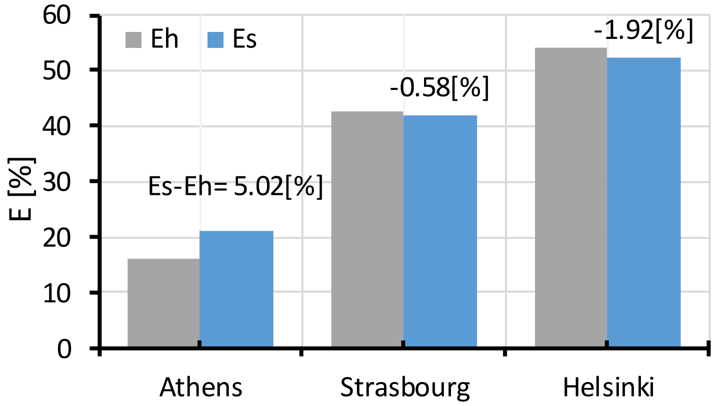

Figure 17: normalized energy consumption difference between HFC and CO2 systems using detailed (h) and simplified (s) simulation.

This comparison was then extended to all the 65 test locations; results are reported in Figure 18. For most of the European locations, the simplified approach proves to be a reliable comparison tool, as most of the areas results in an error on the normalize consumption increase below 5%.

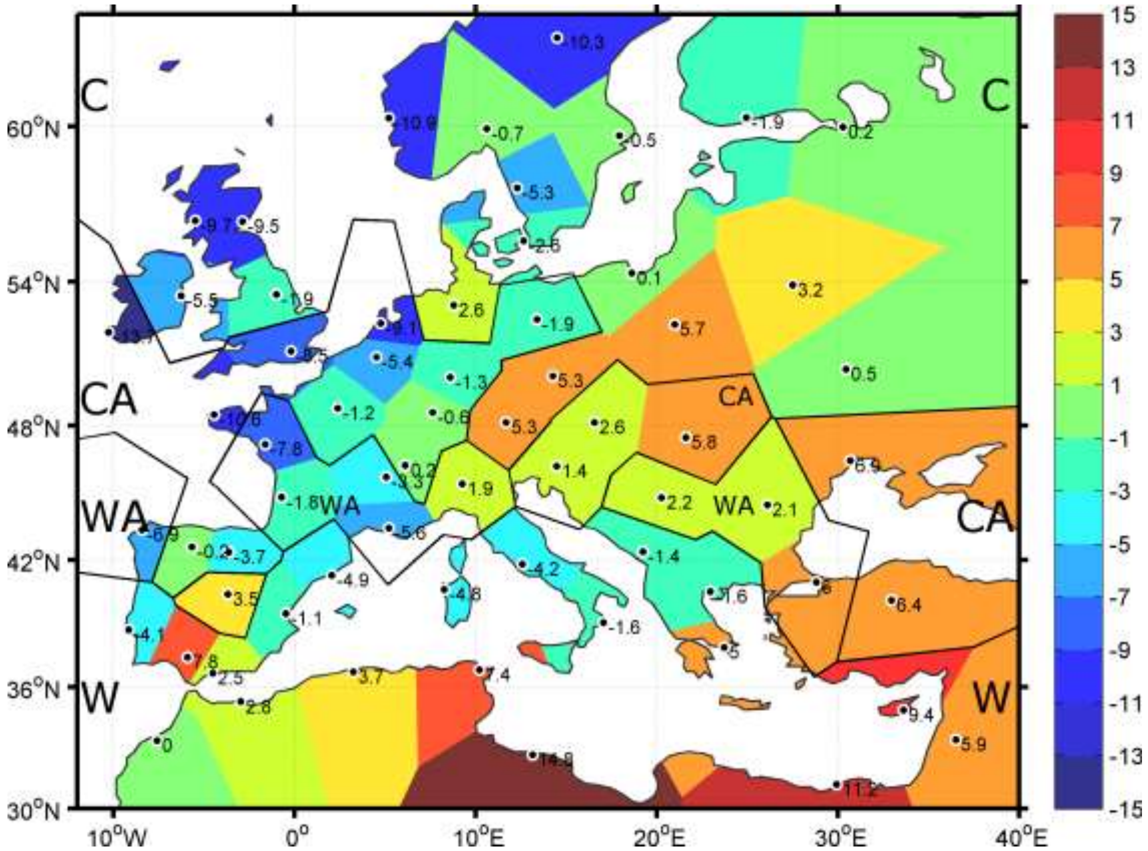

Figure 18: normalized energy consumption increase estimation error $\Delta E = E_S - E_H$.

6 Conclusions and future development

The application of the simplified method proposed by European Standards EN 14825 and EN 13215, that apply to HVAC systems and condensing units respectively, to commercial refrigeration packs has shown to be a powerful tool to easily compare system that have different COP at rated boundary conditions. When two systems are compared, the simplified method is able to maintain the relative difference in energy consumption provided by a detailed hourly based simulation.

A sensitivity analysis has demonstrated that considering 3 or 4 climatic regions in Europe can lead to a good compromise between accuracy and complexity, when referring to European and Mediterranean Countries.

Further development would include part load performances, following the path of EN 14825 and EN 13215.

When relating to parameters for the presentation of performance data, following the existing standards, particular care needs to be dedicated to the evaluation of new trends in technology, such as improved evaporator feeding technologies which allow significantly higher evaporation temperature (Minetto et al., 2014; Banasiak et al., 2015). For example, traditional declaration of data at -8°C for MT systems may not suitably represent the reference working condition of state of the art systems based on CO_2.

The presented procedure was demonstrated to be capable of comparing in a fair and reasonably precise way MT systems. The presented results can be then valuable as a starting point to extend the approach to more complex systems, such as those integrating medium temperature and low temperature cabinets and storage rooms like CO_2 transcritical boosters (Sawalha, 2008; Ge and Tassou 2011), heat recovery and the new-born integrated solutions including air conditioning (Hafner et al., 2015)

To accomplish this goal the definition of COP for complex or integrated systems should be addressed and the procedure accordingly developed.

Nomenclature

r	specific humidity ratio	[-]
z	number of reference climat/bins distribution	
Pc	non dimensional cooling demand	[-]
Pn	nominal cooling power	[-]
RH	relative humidity ratio	[%]
T	Temperature [K]	
U	number of test locations	
V	number of test COP curves	
W	Energy input [J]	
Q	Cooling Energy [J]	
$\dot{Q}$	Cooling Power [W]	

Subscripts

A, B, C, D	referring to the four control point for the interpolation of the COP curve
c	cold isle
ext	external
h	hour-based simulation
i	i-th hour of the year
j	j-th temperature or bin of the bins distribution accounted
in	internal
loc	reference location
s	simplified
u	u-th test location
v	v-th test COP curve

Acronyms

COP	coefficient of performance
LT	low temperature
MT	medium temperature

SCOP seasonal coefficient of performance

SEER seasonal energy efficiency ratio

SEPR seasonal energy performance ratio

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