Review of Regulation 206/2012 and 626/2011

Air conditioners and comfort fans

Task 3 report

Users

Final version

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Abbreviations

COP	Coefficient of Performance for air conditioners in heating mode
BLc	Annual cooling load per square meter of room area (kWh/m2/year/)
CDD	Cooling Degree Day
EER	Energy Efficiency Ratio for air conditioners in cooling mode
GWP	Global warming potential
HCK	The number of hours the unit is considered to work in crankcase heater mode for air conditioners
HOFF	The number of hours the unit is considered to work in off mode for air conditioners
HSB	The number of hours the unit is considered to work in standby mode for air conditioners
HTO	The number of hours the unit is considered to work in thermostat off mode for air conditioners
INF	Infiltration in kW (cooling capacity lost - negative capacity value - because of infiltration)
PCK	The electricity consumption during crankcase heater mode for air conditioners
POFF	The electricity consumption during off mode.
PSB	The electricity consumption during standby mode for air conditioners
PTO	The electricity consumption during thermostat off mode for air conditioners
QCE	The reference annual cooling demand for air conditioners in cooling mode
QHE	The reference annual heating demand for air conditioners in heating mode
Sc	Sizing coefficient for cooling
SCOP	Seasonal Coefficient of Performance for air conditioners, heating mode
SEER	Seasonal Energy Efficiency Ratio for air conditioners, cooling mode
SHR	Sensible Heat Ratio for air conditioners
VRF	Variable Refrigerant Flow

Introduction to the task reports

This is the introduction to the interim report of the preparatory study on the Review of Regulation 206/2012 and 626/2011 for air conditioners and comfort fans. The interim report has been split into five tasks, following the structure of the MEErP methodology. Each task report has been uploaded individually in the project's website. These task reports present the technical basis to define future ecodesign and energy labelling requirements based on the existing Regulation (EU) 206/2012 and 626/2011.

The task reports start with the definition of the scope for this review study (i.e. task 1), which assesses the current scope of the existing regulation in light of recent developments with relevant legislation, standardisation and voluntary agreements in the EU and abroad. Furthermore, assessing the possibility of merging implementing measures that cover the similar groups of products or extend the scope to include new product groups. The assessment results in a refined scope for this review study.

Following it is task 2, which updates the annual sales and stock of the products in scope according to recent and future market trends and estimates future stocks. Furthermore, it provides an update on the current development of low-GWP alternatives and sound pressure level.

Next task is task 3, which presents a detailed overview of use patterns of products in scope according to consumer use and technological developments. It also provides an analysis of other aspects that affect the energy consumption during the use of these products, such as component technologies. Furthermore, it also touches on aspects that are important for material and resource efficiency such as repair and maintenance, and it gives an overview of what happens to these products at their end of life.

Task 4 presents an analysis of current average technologies at product and component level, and it identifies the Best Available Technologies both at product and component level. An overview of the technical specifications as well as their overall energy consumption is provided when data is available. Finally, the chapter discusses possible design options to improve the resource efficiency.

Simplified tasks 5 & 6 report presents the base cases, which will be later used to define the current and future impact of the current air condition regulation if no action is taken. The report shows the base cases energy consumption at product category level and their life cycle costs. It also provides a high-level overview of the life cycle global warming potential of air conditioners and comfort fans giving an idea of the contribution of each life cycle stage to the overall environmental impact. Finally, it presents some identified design options which will be used to define reviewed ecodesign and energy labelling requirements.

Task 7 report presents the policy options for an amended ecodesign regulation on air conditioners and comfort fans. The options have been developed based on the work throughout this review study, dialogue with stakeholders and with the European Commission. The report presents an overview of the barriers and opportunities for the reviewed energy efficiency policy options, and the rationale for the new material/refrigerant efficiency policy options. This report will be the basis to calculate the estimated energy and material savings potentials by implementing these policy options, in comparison to no action (i.e. Business as Usual – BAU).

The task reports follow the MEErP methodology, with some adaptations which suit the study goals.

3 Task 3

Task 3 follows the MEErP methodology and aims to identify consumer behaviour likely to influence the assessment of the environmental impact and the life cycle cost of the product, and it should also identify barriers and restrictions to possible ecodesign measures, due to social or infrastructural factors. At last, it should quantify relevant user-parameters that influence the environmental impact during product-life and that are different from the standard test conditions. It includes the following sections:

1. Systems aspect: Use phase
2. End-of-life behaviour: Discussion of the current state of play in terms of end of life options and practices.
3. Local infrastructure: Description and identification of barriers and opportunities relating to the local infrastructure.

3.1 System aspects use phase

Use phase often accounts for the largest energy consumption in a product's lifetime. It is therefore important to investigate what influences the use phase consumption.

At the time of the preparatory study, a seasonal performance standard has been developed to account for main variables influencing air conditioner energy consumption, which are the variations of the cooling energy supplied and of outdoor temperature along the year. This is in line with the "extended product approach" defined by MEErP, that as the energy-related products (ErP) are subject to various load/user demands, the product scope should extend to controllability i.e. the flexibility and efficiency to react to different load situations. In the following subsections, the foundations of this extended product approach are briefly described, and main parameters are reviewed to see whether they have to be adapted or not.

Regarding portable air conditioners, a seasonal performance standard is not yet available and an additional parameter is to be considered under the extended scope approach, which is the infiltration of outdoor air due to the operation of the product. The impact of infiltration is studied in heating and in cooling mode. Possible seasonal performance metrics are discussed.

These points are discussed in the following sections:

- **The basics of energy consumption**: understanding in general what the consumption consists of, and in particular the implications of modifying the main parameters of the present seasonal performance metrics;
- **Cooling loads and SEER metrics parameters:** investigating if it is necessary to update the parameters of the metrics. Important factors are the cooling loads, sizing, standard equivalent full load hours, SEER temperature profile, hours of operation per mode.
- **Heating loads and SCOP metrics parameters:** investigating if it is necessary to update the parameters of the metrics in heating mode. Important factors are the use of heating (reversible units) and the heating load.
- **Crankcase heater in SEER and SCOP metrics:** investigating the need to update the measurements.

- **Case of portable air conditioners:** investigating the infiltration of air and whether seasonal performance cooling metrics should be adopted.
- **Real life versus standard performance** and the impact of proper maintenance.
- In the case of comfort fans, the **use patterns of comfort fans**.

3.1.1 Basics of energy consumption of air conditioners

The energy consumption used for cooling/heating by air conditioners (reversible air conditioner) depends on the cooling load (respectively cooling and heating load), product efficiency for hours with cooling SEERon (or heating SCOPon) and consumption of the unit in auxiliary modes (thermostat-off, standby, crankcase heater).

Cooling energy consumption is calculated via the following equation:

$$Q_{CE}\ (kWh) = \frac{Q_C}{SEER_{on}} + H_{TO} \times P_{TO} + H_{SB} \times P_{SB} + H_{CK} \times P_{CK} + H_{OFF} \times P_{OFF}$$

Where:

- *Q_{CE} = Annual electricity consumption for cooling, expressed in kWh/a*
- *Q_C = The reference annual cooling demand, expressed in kWh/a*
- *H_{TO}, H_{SB}, H_{CK}, H_{OFF} = the number of hours the unit is considered to work in respectively thermostat off mode, standby mode, crankcase heater mode and off mode.*
- *P_{TO}, P_{SB}, P_{CK}, P_{OFF} = the electricity consumption during respectively thermostat off mode, standby mode, crankcase heater mode and off mode, expressed in kW*

The cooling load is the sum of the transmission heat losses, air change due to ventilation and infiltration, internal loads (heat loads due to people and equipment releasing heat inside the building) and solar heat gains to maintain a desired room temperature. Transmission and air change losses mainly depend on temperature difference indoor/outdoor, hence on climate and indoor set point temperature and the building envelope. In addition, there may be dehumidification loads due to moisture condensation on cold coils of cooling equipment.

The cooling load of a cooled area/building is in general given in annual kWh of cooling load per square meter of room area (noted as *BLc*). This is the useful unit for the building but has no meaning for the product itself. What matters for a product consumption is the cooling energy it supplies, or kWh cooling load per product noted below as *Q_{ce} (product)*. In Regulation 206/2012, it is supposed what is called a "perfect sizing" condition, which means the capacity of the unit at design conditions is exactly matching the maximum cooling load in these same conditions.

The link between the product and building loads is then derived as follows:

$$Q_C(product) = P_{designc}\ (in\ kW) \times\ H_{CE}\ (in\ hours)$$

$$\text{and } H_{CE} = \frac{BLc\ (kWh/m^2/y)}{Sc\ (W/m^2)}$$

With:

- Sc: the sizing coefficient for cooling (Sc), which is the ratio linking the maximal cooling load at design conditions and the cooled area,

- H_{CE}: the equivalent number of full load cooling hours,
- Pdesignc: the maximum building cooling needs in kW, which is equal to the maximum cooling capacity of the unit.

It is important to notice that the higher the sizing coefficient Sc, the lower the number of equivalent hours H_{CE}, the lower the reference annual cooling demand Q_C and the lower is the energy consumption

Regarding the seasonal cooling efficiency of a given unit over a season, it depends on:

- the indoor and outdoor temperatures of operation,
- the indoor humidity,
- the load ratio of the unit (ratio of building cooling load to unit capacity), which in turn depends on the sizing of the unit and on the unit's capability to modulate its capacity,
- the indoor air flow set by the end-user (in general the outdoor air flow is fixed by the unit controller),
- its power consumption when it is not cooling,
- and the maintenance of the unit.

Oversizing here may lead to decrease in efficiency and then to an increase in energy consumption; this depends on efficiency variation with loads and also whether consumption of the unit when compressor is not working is proportional or not to the size of the unit. It is also important to notice that reducing the equivalent number of full load cooling hours (Hce) tends to increase the weight of low power mode consumption and then to reduce efficiency (increasing the gap between SEERon and SEER).

Regarding heating, what has been discussed above for cooling load is also valid for heating load, with two main differences:

- there is no latent load for heating,
- in heating mode, products may not be designed to supply by themselves the whole heating capacity of the building at design conditions due to e.g. the sizing was done according to the cooling load (for reversible units), therefore part of the heat may be supplied by a back-up heater[1] (result of an economic optimization) so that Pdesignh is a declaration of the manufacturer.

Using the same notations but replacing c for cooling by h for heating:

$$Q_H(building) = P_{designh}\ (in\ kW) \times\ H_{HE}\ (in\ hours)$$

$$and\ H_{HE} = \frac{BLh\ (kWh/m^2/y)}{Sh\ (W/m^2)}$$

Still it is supposed that the declared maximum capacity of the unit plus the back-up heater capacity at Tdesignh equals the maximum building load in the same condition to derive the heating load of the unit. In present Regulation (EU) 206/2012, the SCOPon calculation includes the impact of an electric back-up heater with a coefficient of performance of 1.

[1] An electric back-up heater is defined in Regulation (EU) N° 206/2012, but other types might be used.

3.1.2 Cooling loads and SEER metrics parameters

In this part, the hypothesis of the SEER metrics in Regulation (EU) 206/2012 and 626/2011 are assessed if they need to be updated. This regards the following parameters:

- the specific cooling loads BLc in kWh/m^2/year,
- the sizing coefficient Sc in W/m^2/year,
- the equivalent full load hours,
- The outdoor temperature profile.

3.1.2.1 Specific cooling loads

As it cannot be measured easily, there is no statistics available on real life building cooling loads or the cooling capacity supplied by air conditioners (note that building cooling load equals the air conditioner cooling capacity, except in case of under-sizing); the same is true for cooling efficiency (which requires the measurement of cooling load).
Information on energy consumption for space cooling in Europe is also scarce and information on measured values generally insufficient to identify separately product efficiency and load (consumption being the ratio of load to efficiency). The preparatory study used simulations to derive cooling loads of air-conditioned buildings. Three different simple buildings were simulated, one house, one office and one shop (in Figure 1, respectively noted as "Res", "Off" and "Shop"), varying envelope insulation characteristics and air change properties for existing buildings and for new buildings built after 2006. The main characteristics of these buildings are given in the ANNEX.

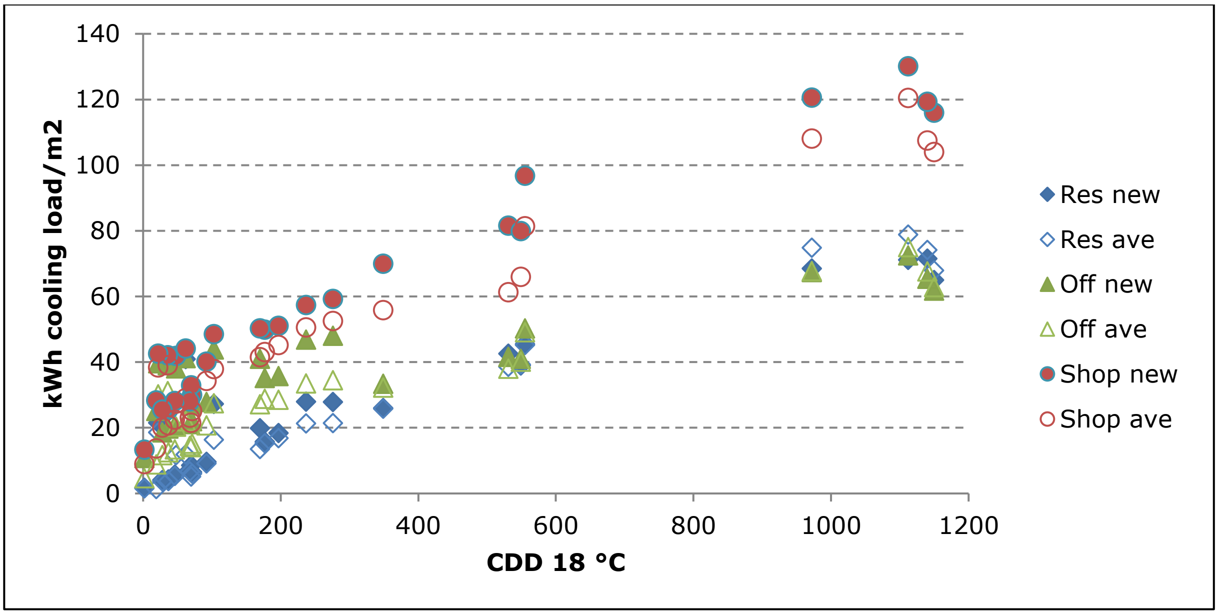

Figure 1: Yearly cooling load versus cooling degree days (CDD)[2] with base temperature 18 °C, source preparatory study.

As it can be seen on Figure 1, the total cooling load is in general proportional to the cooling degree day indicator of the specific climate simulated. Offices and shops, with larger internal loads and with larger operating hours have higher requirements in total.

One of the specifics of these simulations is that, as it can be seen in table below in red, cooling loads appear very high as compared to the cooling degree day (CDD) indicator in

[2] CDD: cooling degree days have been defined in Task 2 paragraph 3.2.2.4

Nordic countries and other central Europe countries with simulated high U values. This is supposedly a heat trapping effect of internal loads because of high insulation levels, but in some buildings where people can open doors and windows (for instance residential buildings in quiet environment), this most likely overestimates the real air conditioner cooling loads. The insulation levels were based on an Ecofys report of 2006 and represented indicative insulation values for average existing and new buildings (respectively noted average and new in Figure 1 and Table 1). Note this is still the source used in (Kemna, 2014)[3] to estimate average EU heating load in Europe.

More insulated new buildings also have lower cooling loads for higher cooling degree day CDD values.

*Table 1: Cooling load versus cooling degree days, source preparatory study. * indicates CDD values not given in Preparatory study, these are issued from ASHRAE IWEC data. Below in red shows that cooling loads appear very high as compared to the cooling degree day (CDD) indicator in Nordic countries and other central Europe countries with simulated low U values.*

	CDD18	Cooling load in kWh/m2/year					
		Res new	Res ave	Off new	Off ave	Shop new	Shop ave
Ireland	2	1.7	1.5	10.8	4.6	13.4	8.9
Lithuania	19	28.2	1.3	25.1	8.9	28.3	13.7
Sweden	22	21.6	18.6	39.6	30.2	42.5	38.3
UK	28	4.1	3.6	18.6	11.7	25.5	20.1
Finland	36	24.7	22.3	39.0	31.2	42.0	39.1
Denmark	37	4.2	3.8	19.7	12.5	25.6	20.7
Netherlands	46	5.7	5.3	20.8	13.4	28.1	22.5
Estonia	48	39.7	11.7	38.1	20.3	41.9	27.4
Latvia	62	41.0	11.8	41.2	22.0	44.1	29.0
Germany	68	7.2	6.7	21.1	14.5	28.1	23.1
Czech Republic	70	8.5	5.2	27.2	14.1	32.9	21.5
Belgium	71	6.5	6.0	22.0	15.2	29.9	25.2
Luxembourg	92	9.6	9.1	27.9	20.8	40.1	34.3
Poland	103	27.2	16.4	43.9	27.5	48.5	37.9
Slovenia	170	19.8	13.5	41.0	27.2	50.2	41.5
Austria	177	15.7	15.3	35.1	28.8	49.8	43.1
France (Mâcon)	197*	18.4	16.9	35.8	28.5	51.1	45.2
Slovakia	237	27.9	21.3	46.9	33.5	57.3	50.5
Hungary	276	27.8	21.4	48.0	34.5	59.3	52.5
Italy (Milano)	349	26.0	25.8	33.4	32.2	69.9	55.8
Portugal	531	42.6	38.5	41.6	38.0	81.6	61.3
Spain (Madrid)	549	38.9	39.3	41.0	40.2	79.9	66.0
Italy (Roma)	555*	45.5	45.2	50.2	49.1	96.7	81.3
Malta	972	68.5	74.8	67.4	67.7	120.5	108.0
Cyprus	1112	71.2	78.8	72.5	75.0	130.1	120.4
Spain (Seville)	1140*	71.5	74.2	65.4	67.6	119.3	107.5
Greece	1150	65.0	67.9	61.8	62.9	116.0	104.0

[3] René Kemna (VHK), 2014. Final report on Average EU building heat load for HVAC equipment. Specific contract No. ENER/C3/412-2010/15/FV2014-558/SI2.680138 with reference to Framework Contract ENER/C3/412-2010. Prepared for the European Commission DG ENER C.3. Delft, 2014.

As mentioned, the above presented cooling loads were simulated. Although annual average real cooling loads of air conditioners < 12 kW are not known in Europe, we can compare the simulations of the Preparatory study with other sources.

Values of average real cooling loads monitored at substations of district cooling in different cities in Europe are given in the literature[4]; cooling loads are shown against cooling degree days of the cities in Figure 2; CDD values used are ASHRAE IWEC data[5]. This gives relatively high values, in line with buildings connected to district cooling (e.g. tall office buildings and data centres), which are likely to have higher cooling loads than residences or small shops and offices.

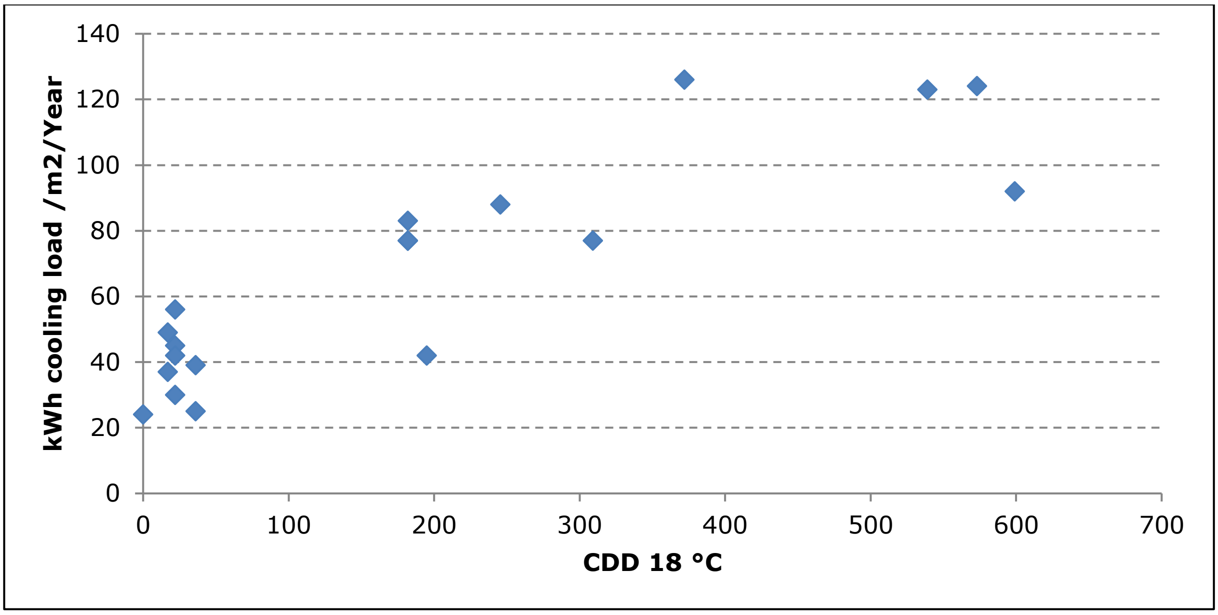

Figure 2: Average building cooling loads of some district cooling networks in Europe[6].

The JRC used US EIA estimate of cooling electricity consumption in the USA to derive an estimate of residential cooling load for air-conditioned buildings by dividing it by the SEER of the specific unit in house (using minimum mandatory US MEPS of the installation year in the house) and correlated this against CDD; both cooling electricity consumption and load are shown in Figure 3. There is a clear dependency of consumption and loads with cooling degree days (base 18 °C). Note that the JRC estimate has to be corrected to get the cooling load per m^2 of cooled area, as m^2 in JRC regression are total house m^2 and that in mild climate only a portion of the dwelling might be cooled. This is done in Figure 4 using input from Task 2 (Ratio of surface to penetration as a function of CDD computed on the same base data).

[4] Sven Werner, European space cooling demands, In Energy, Volume 110, 2016, Pages 148-156, https://doi.org/10.1016/j.energy.2015.11.028.
[5] https://energyplus.net/weather
[6] Sven Werner, European space cooling demands, In Energy, Volume 110, 2016, Pages 148-156, https://doi.org/10.1016/j.energy.2015.11.028.

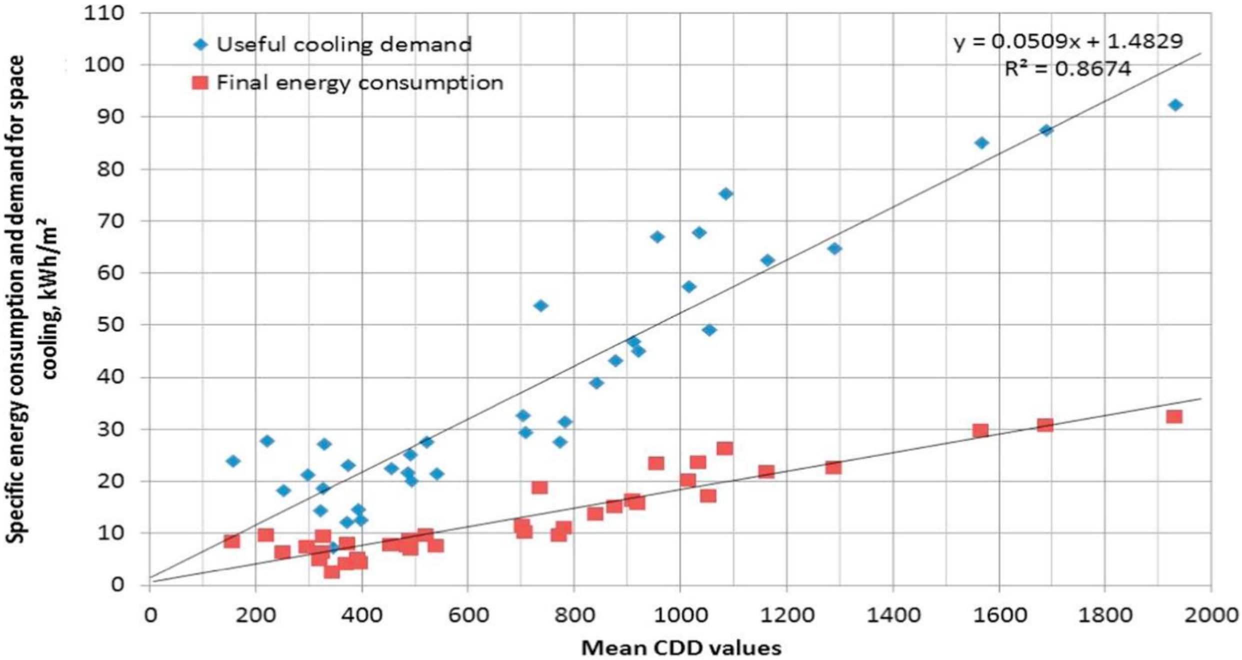

Figure 3: Dependency of final specific energy consumption for cooling and useful cooling demand in residencies of USA on climatic conditions in kWh/m² (of total dwelling surface)/year[7].

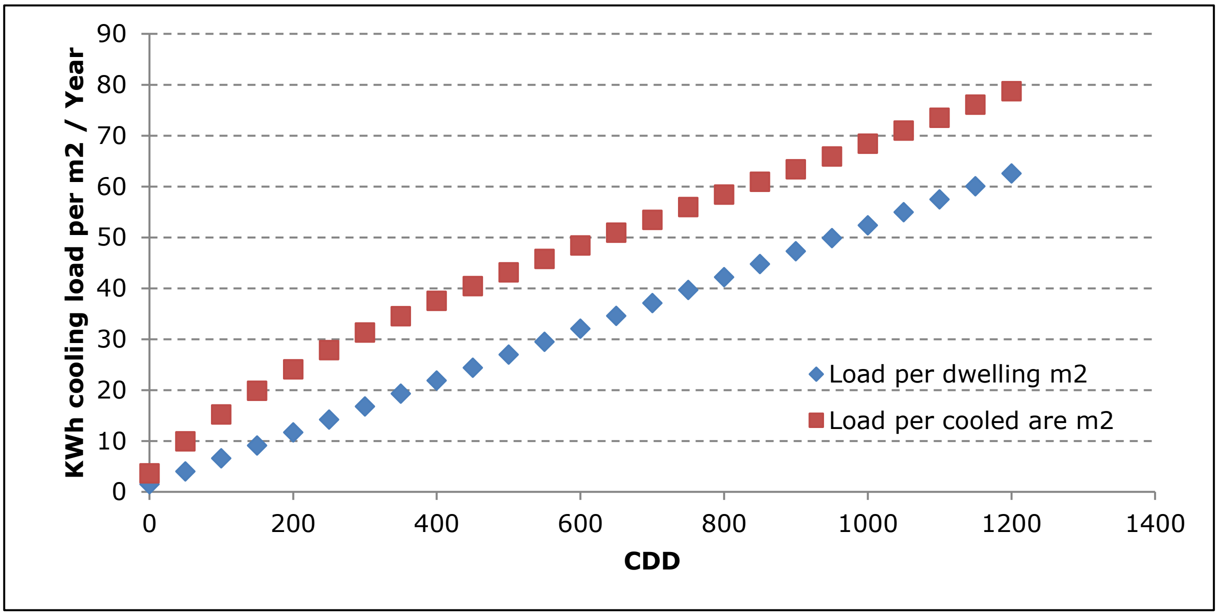

Figure 4: Dependency of final specific energy consumption for cooling and useful cooling demand in residencies of USA on climatic conditions reported to total dwelling are and to cooled area only, adapted from (Jakubcionis and Carlsson, 2017).

The ENTRANZE[8] project simulated cooling loads of typical old buildings (representative of very low insulation levels, typically built in years 1960 to 70's, i.e. before any building regulation) across Europe. We show here the results for houses in Figure 5. Cooling loads are also correlated to CDD (base temperature 18 °C). For offices, there is no marked difference with houses, while for apartments, the cooling loads are lower than in houses.

[7] Mindaugas Jakubcionis, Johan Carlsson, Estimation of European Union residential sector space cooling potential, In Energy Policy, Volume 101, 2017, Pages 225-235, https://doi.org/10.1016/j.enpol.2016.11.047
[8] Paolo Zangheri, Roberto Armani, Marco Pietrobon and Lorenzo Pagliano (eERG), Heating and cooling energy demand and loads for building types in different countries of the EU, D2.3. of WP2 of the Entranze, March 2014. http://www.entranze.eu

The comparison of all information sources for residential sectors are shown in Figure 5 and service sector in Figure 6.

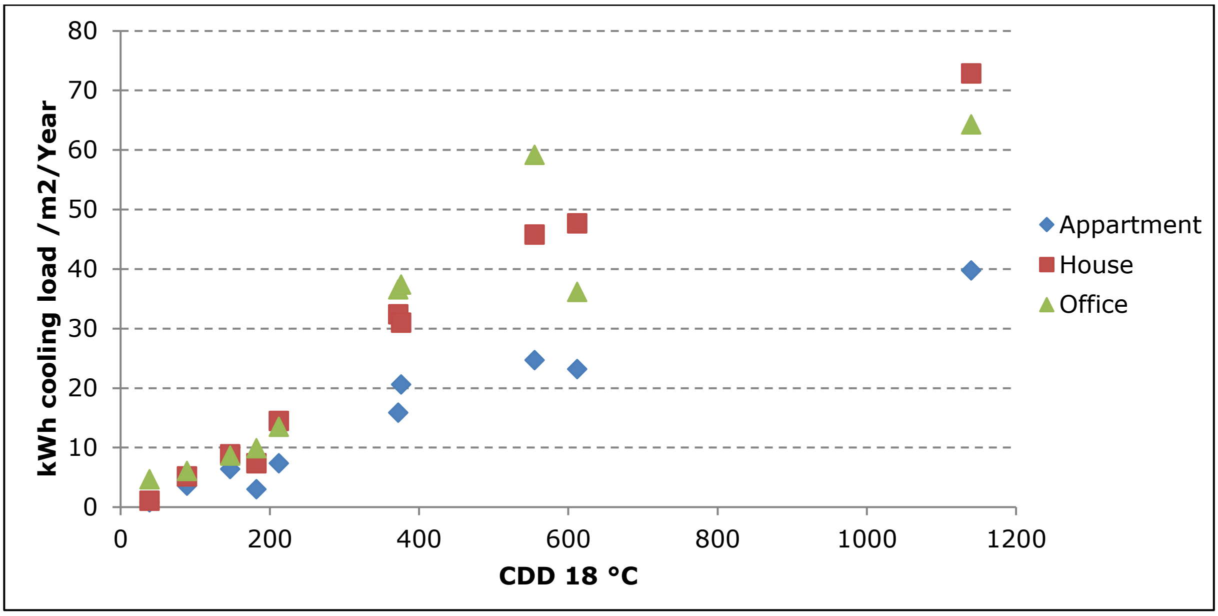

Figure 5: Simulation of cooling loads of typical old buildings, source ENTRANZE project

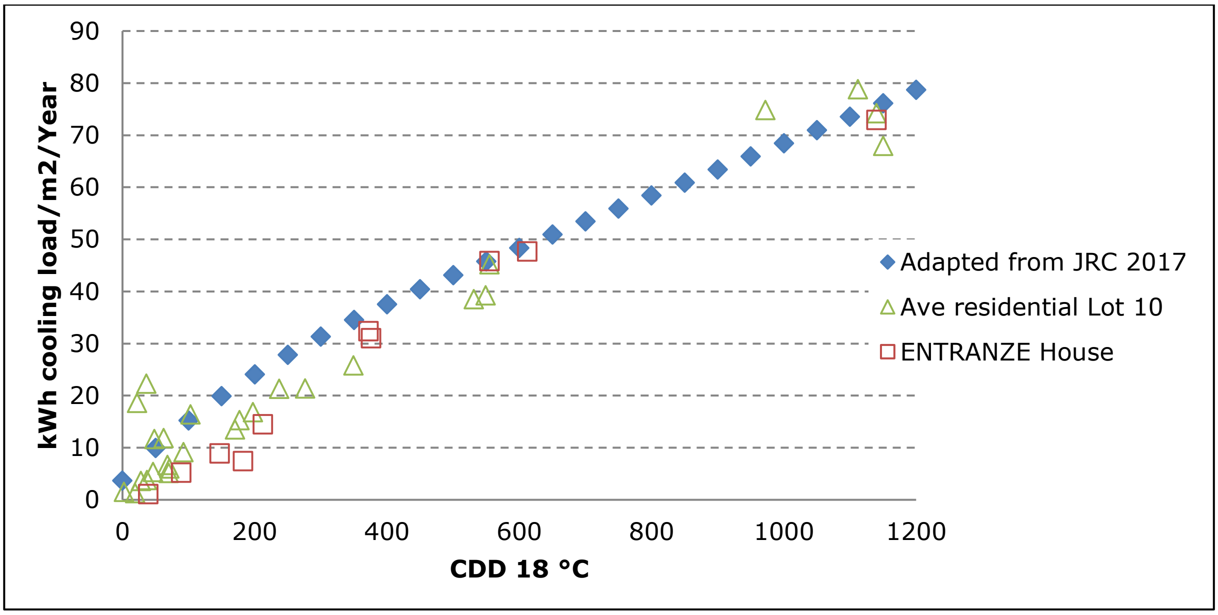

Figure 6: Comparison of residential cooling loads from different sources

For residences, all 3 sources are comparable at high loads; estimates from JRC are higher at low loads than in Lot 10 preparatory study simulation except for countries with very high insulation levels. The ENTRANZE simulations do not exhibit the heat trapping effect shown by the Preparatory study simulation, but this is in line with heat insulation properties selected for the simulation. Overall, there is a good agreement between the 3 sources, and values in Preparatory study still can be used. The only doubt is about apartment whose load in average seems lower than for houses according ENTRANZE simulations and which are not accounted for in the cooling load estimates for residences in Lot 10 preparatory study. The respective share of cooling system < 12 kW installed in apartments and houses in Europe is not available, therefore cannot be assessed.

For the service sector, real loads as seen at district cooling levels are much higher than simulated values. This is possibly linked with the building types connected to district cooling networks. Building Research Establishment (BRE) realized estimates of cooling loads in offices in the UK[9]. It was based upon field monitoring of electricity consumption and assessment of system efficiency from technical data. Average estimated cooling load in real buildings is about 30 kWh/m2/year (with about 30 % of the 99 office buildings considered having values of 40 kWh/m2 or more). So, it is possible to find buildings with high loads (up to 60 kWh/m^2/year for very low CDD values) as shown by cooling networks but it is probably not representative of typical buildings. Another striking point is the steeper slope for cooling load increase plotted against Cooling Degree Days (CDD) for district cooling buildings. It is difficult to explain the steeper slope without more knowledge of the loads being connected to the district cooling system. If this is due to the system, this can be linked to the decrease in the share of free cooling with increasing CDD and/or to the non-linear increase in heat released by pumps and fans with higher loads.

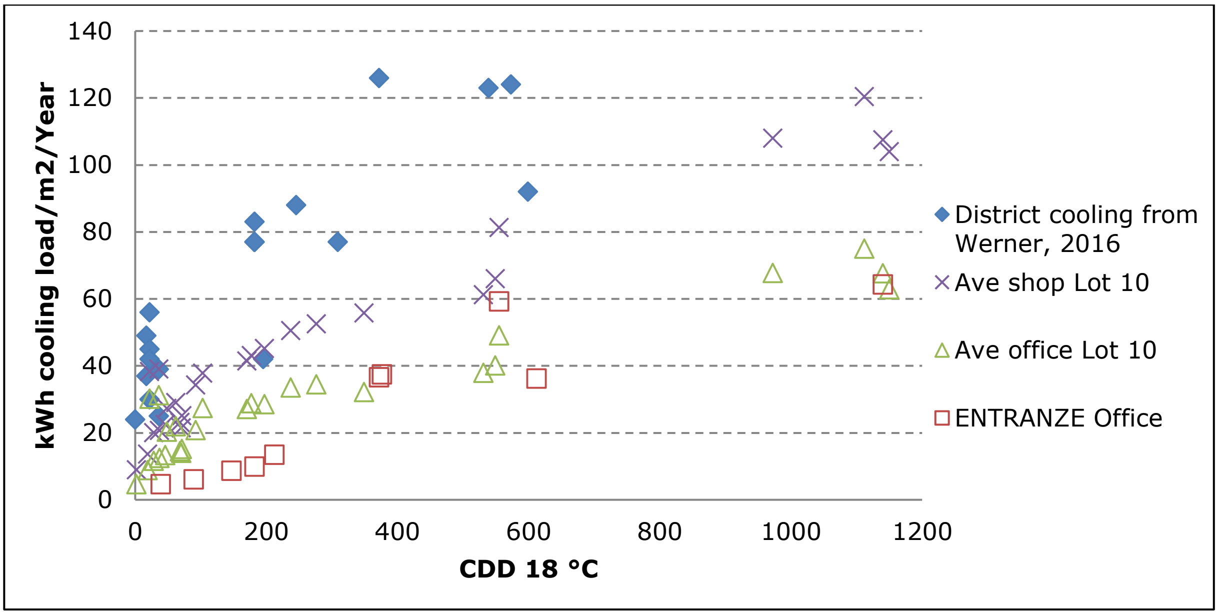

Figure 7: Comparison of service cooling loads from different sources

An audit campaign on more than 100 offices in Sweden[10] give also loads similar to the ones of Lot 10 simulation results for average office buildings: specific electricity consumption of 10.3 kWh/m^2/year for cooling, which for this climate, probably gives more than 30 kWh/m^2 cooling loads when multiplying by the average SEER of the cooling machines.

ENTRANZE simulation for offices give similar results at higher loads than Preparatory study simulation but much smaller loads at low CDD values: thanks to lower insulation level and higher infiltration air change per hour (ENTRANZE typical buildings are older buildings), internal loads are more easily mitigated, decreasing the cooling loads. Both sets of simulation are thus consistent. In fine, we will then continue using the Preparatory simulations.

Preparatory study loads include both latent and sensible load. Latent load is uncontrolled in general; it arises from the condensation of the water vapor of the air on the cooling coil

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[9] https://www.bre.co.uk/filelibrary/pdf/projects/aircon-energy-use/StudyOnEnergyUseByAirConditioningFinalReport.pdf

[10] https://www.sintef.no/globalassets/project/eldek/publisering/tr-a6999-state-of-the-art-projects-for-estimating-the-electricity-end-use-demand.pdf

of the indoor unit when the coil temperature is below the air temperature dew point. However, in average, the share of the latent load is supposed to be low. In the Task 4 of the preparatory study, the sensible heat ratio (SHR, share of sensible to total cooling capacity) of the average unit was supposed to be about 0.9 (so 10 % latent load) for an average outdoor temperature of 23 °C and 50 % humidity (23 °C is close to the average outdoor temperature in cooling season[11]). With present split air conditioners, the air flow is constant at reduced capacity, which increases the evaporating temperature and increase the SHR. At 50 % load, the SHR is close to 1 so that there is no latent load.

So, the fact that the unit capability to dehumidify is decreasing because of higher evaporating temperatures and inverter operation at part load will probably tend to curb the specific cooling load and energy consumption of air conditioners in Europe by a maximum of 10 % in average in the future.

Conclusions regarding cooling loads

There is no or limited statistics on cooling load in Europe and on cooling energy consumption. Through reviewing other potential sources of load estimates, it appears appropriate to keep the Lot 10 Preparatory study simulations to estimate cooling loads. Available sources give comparable values.

In order to properly model the impact of climate change and to include in this review study countries not modelled in the Preparatory study (Bulgaria, Croatia, Romania), correlations between specific loads and CDD based on Lot 10 preparatory study are used.

Task 2 stock model does not link building age and air conditioning installations nor the type of building in the service sector. Hence, specific CDD - average loads correlations by sector (services / residential) are computed on the following basis (same for all EU countries):

- 15 % of installation in new buildings and 85 % in existing buildings,
- regarding the service sector, about 50 % in offices and 50 % in retail buildings following 2015 data purchased from BSRIA (note this was about 25 % and 75 % resp. in the preparatory study using the same BSRIA source for 2005)

[11] René Kemna (VHK), 2014. Final report on Average EU building heat load for HVAC equipment. Specific contract No. ENER/C3/412-2010/15/FV2014-558/SI2.680138 with reference to Framework Contract ENER/C3/412-2010. Prepared for the European Commission DG ENER C.3. Delft, 2014.

Correlations of these specific cooling loads are drawn versus CDD as shown in Figure 8 and Figure 9.

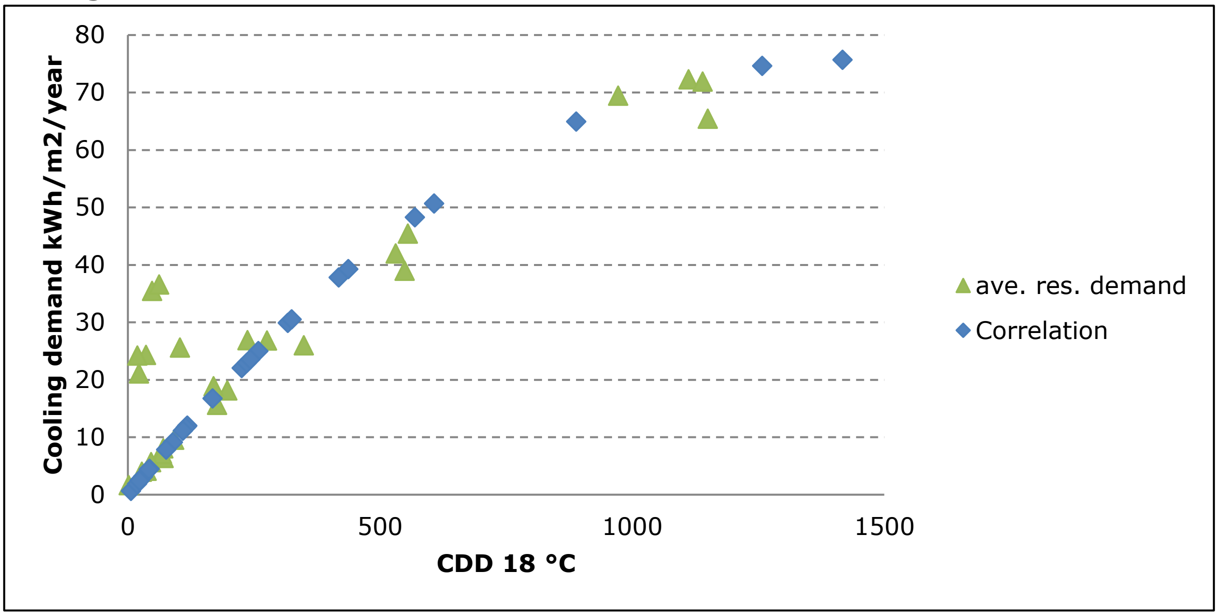

Figure 8: Specific demand per country in residences and correlation, adapted from preparatory study

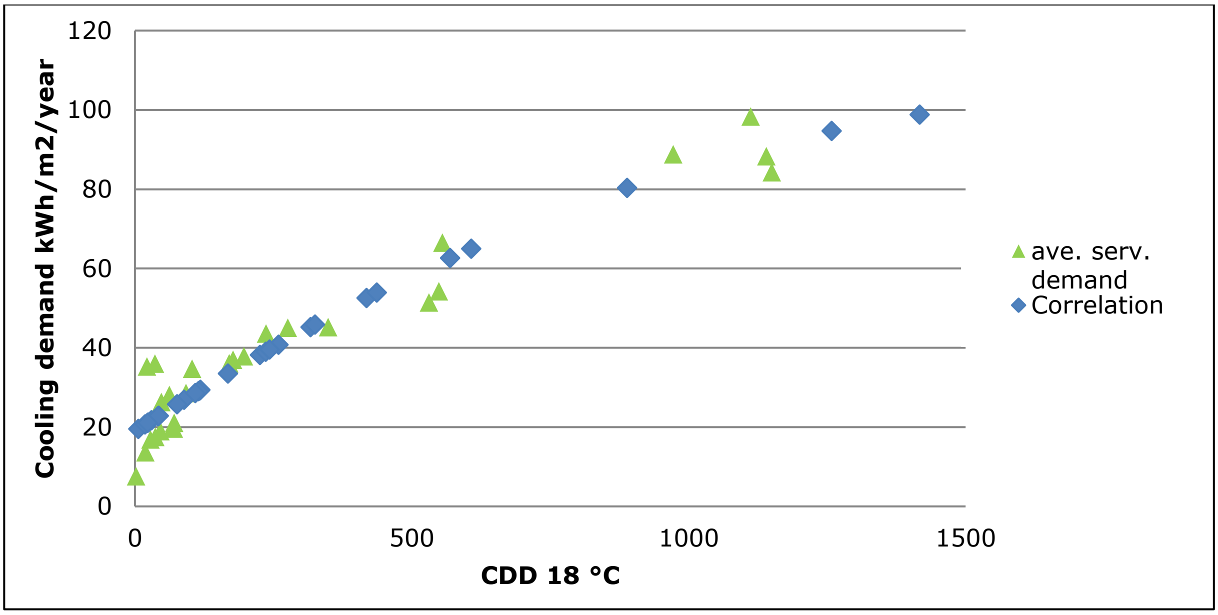

Figure 9: Specific demand per country in services and correlation, adapted from preparatory study

These correlations give specific cooling loads per country using CDD values per country as indicated in Task 2. When combining country specific loads with the stock model of Task 2, the EU-28 average specific cooling load can be calculated (Table 2). Weighted average values (per m^2 of cooled area) is of about 45 kWh/m^2/year, with limited planned evolution despite accounting for climate change impact, because of the faster increase of sales in countries with milder countries. This value is similar to the weighted average in Lot 10 study (it was 33 for cooling only and single duct units in 2005 and 39 for reversible split[12]). The weighted average CDD value is 474.

[12] Preparatory study, Task 4 report, table 4-30, page 89

Conclusively, the average EU28 load estimate in 2015 is slightly higher than in the Preparatory study. The EU average specific load of 45 kWh/m^2/year is to be used to compute the equivalent full load cooling hours below in section 3.1.2.3 and in Task 5 to compute the global environmental impacts of the use phase.

Table 2: Estimated weighted average (by cooled floor area) CDD and specific demand for EU28

Year	2015	2020	2025	2030	2035	2040	2045	2050
CDD residential	537	527	537	539	543	548	553	559
CDD services	311	294	304	318	296	335	342	350
CDD weighted average	474	456	464	471	469	487	495	504
Cooling demand residential kWh/m2/y	44	42	43	43	43	44	44	45
Cooling demand services kWh/m2/y	44	44	42	43	44	45	46	46
Average cooling demand kWh/m2/y	44	43	43	43	44	44	45	45

3.1.2.2 Sizing

The very few studies available (though not statistically significant) show that large oversizing is the rule in cooling mode in Europe. In the USA, even though cooling load calculation is mandatory and standardized[13], oversizing is also a common practice. It is thus not surprising that oversizing is common in Europe, where there is in most countries no agreed method to size air conditioners (liability of the installer in most cases for fix installations and following retailer advise for portable ones). In addition, part of the units is also sized to satisfy the whole building heating needs, which may also lead to oversizing in cooling mode in central and northern Europe.

Recent findings[14] using on-board performance measurement method (these methods are discussed in section 3.1.7.1 below) confirm this large oversizing although this regards larger systems (Variable Refrigerant Flow systems used in service buildings). Figure 10 shows that for 95 % of the time, the average load in cooling mode (or in heating mode) was lower than 80 % (respectively 85 % in heating mode) for all systems, and lower than 50 % (respectively 55% in heating mode) for most of them. Figure 10 shows although the efficiency axis is not given, that there is an impact on performances figured by the dot line: the lower the load and the lower the performance. This impact is particularly striking in cooling mode for the UK with a fast degradation of performances with lower maximal load. Despite much milder climate than Spain, seasonal efficiency appears lower in the UK than in Spain at lowest loads.

[13] ACCA Air Conditioning Contractor's Association of America Manual J available at: http://www.acca.org/
[14] Stefan Vandaele, Hiroshi Aihara, Optimisation of VRF systems in buildings by monitoring, in Heiselberg, P. K. (Ed.) (2016). CLIMA 2016 - proceedings of the 12th REHVA World Congress: volume 9. Aalborg: Aalborg University, Department of Civil Engineering.

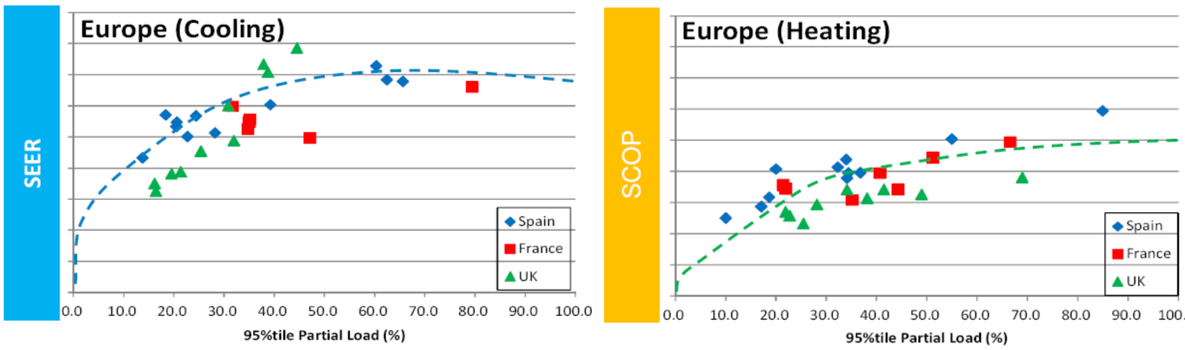

Figure 10: Link between field measured seasonal efficiency in cooling and in heating mode, climate and unit sizing (A 95%tile load of 50% means that during 95% of the time the part load ratio of the unit was 50% or lower - 100% partial load, means full capacity of the unit)

VRF air conditioners with less than 12 kW all use DC inverter compressors (in Task 2 it was mentioned that close to 100% of all split air conditioners sold in 2015 in Europe have DC inverter compressor motors). This does not seem enough to guarantee good performance when large oversizing occurs. These first field monitoring sizing results ever (for VRF) most likely show the impact of cycling below minimum compressor speed, thermostat-off, crankcase and standby consumptions and so the necessity to properly include these phenomena in Regulation 206/2012 (and in Regulation 2281/2016 for larger systems).

It also shows that the seasonal metrics hypothesis of "perfect sizing", in which the 100 % load of the building matches the declared rated capacity of the unit, does not correspond to real life situation. One possibility could be to change the Regulation 206/2012 seasonal performance metrics to account for this proven oversizing. However, it is not sure that such a modified metrics would lead to lower SEER/SCOP values than with "perfect sizing" because the cycling performance is accounted for with a simplified Cd factor, that would probably not translate in high enough performance degradation. In addition, the real-life sizing values and seasonal performance degradation due to oversizing are not known for split air conditioners, which makes it difficult to make design choices for a modified metrics.

So, it is advised to improve the cycling representation in Regulation 206/2012, including the test of cyclic behaviour at low loads for inverter driven compressors and to encourage the development of methods enabling to measure real-life efficiency measurement and real-life sizing (see section 3.1.7.1 below). Thus, if in the future, enough material an oversizing is gathered, an oversizing coefficient could be included in regulation to properly account for the impacts of oversizing[15]. Note also that the EU Energy Performance of Building Directive 2010/31, in articles 14 and 15, requires Member States to evaluate units larger than 12 kW oversizing and to propose solutions to improve real life efficiency. For air conditioners (smaller or larger than 12 kW), measurement on field can only be done by these internal measurement methods presented in section 3.1.7.1.

[15] The perfect sizing hypothesis, 100 % building load corresponds to rated capacity of the air conditioner, could be replaced by, e.g. 100 % + X %. SEER value would decrease because of lower loads and higher impact of auxiliary modes, and Hce as well. But at the moment, there is not enough information to calibrate the oversizing coefficient X %.

Note that portable air conditioners, a LBNL[16] field study led on 19 portable air conditioners gives average load ratio between 50% and 100 % suggesting that undersizing is more common than oversizing.

As information is missing on real life sizing and on the impact on performances, we will keep for now the sizing values used in the Preparatory study.

Sizing in W/m² for energy calculation is presently based on the same Lot 10 preparatory study simulations as used to compute loads. The values are maximal load values of annual hourly dynamic building simulation. They are presented for the different building types in Figure 11. In Figure 12 these sizing values are weighted for residential and service sectors as it was done for specific loads before (15 % new buildings 85 % existing, 50 % offices and 50 % retail for the service sector).

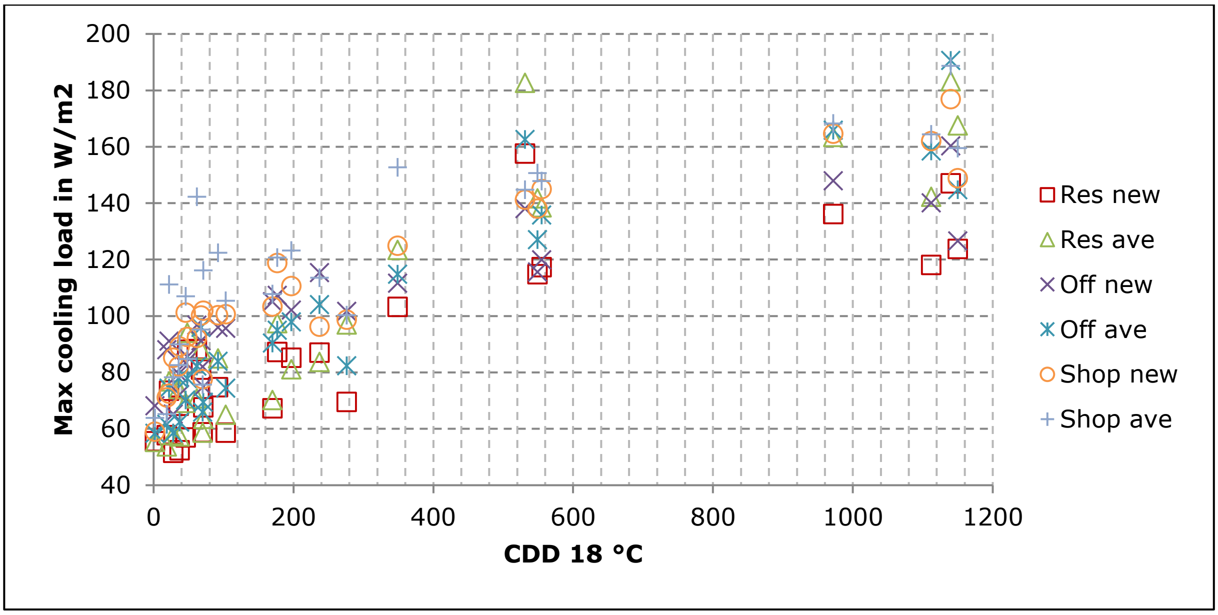

Figure 11: Maximum cooling load in W/m2 and cooling degree days, source preparatory study; CDD values not given in Preparatory study[17]

[16] T. Burke, et al., Using Field-Metered Data to Quantify Annual Energy Use of Portable Air Conditioners, Lawrence Berkeley National Laboratory, Report No. LBNL-6868E (December 2014). Available at: www.osti.gov/scitech/servlets/purl/1166989

[17] ASHRAE IWEC data available on https://energyplus.net/weather

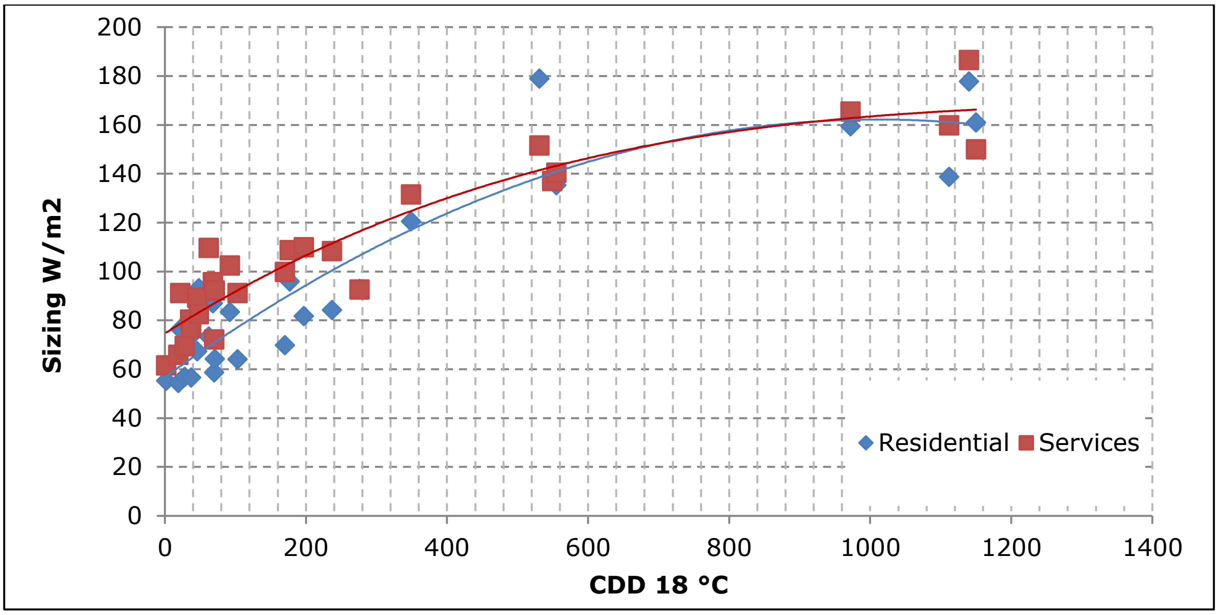

Figure 12: Residential and services average maximum cooling load in W/m2 and cooling degree days, source preparatory study

These values in figures above are in relatively good agreement with good practice indicated by the industry presented in Table 3.

Table 3: typical practices indicated by the industry

Sector / Climate	Warmer	Average	Colder
Residential	120-180 W/m2	100-120 W/m2	70-100 W/m2
Retail	-	100 W/m2	-

In conclusion, the sizing values from the Preparatory study are used in this review study under the form of two CDD correlations, respectively for residential and service sectors.

3.1.2.3 Standard equivalent full load hours

As mentioned in section 3.1.1, equivalent full load hours are essential to evaluate the energy consumption of products as the energy consumption of the unit is proportional to the equivalent full load hours.

In the preparatory study, the EU average cooling load was established at about 40 kWh/m2 by weighting the simulated cooling load by country (in kWh/m^2) by the share of the total installed cooling capacity (in % of kW) per sector and per country (stock weighted average). The weighted average CDD was 472, which was close to the CDD value estimate for Italy in the preparatory study (average of Milano and Rome climates). The specific demand of 40 kWh/m^2 was also the weighted average demand considering specific loads and share of installations in different building types for Italy. For this climate, the average value of sizing was 120 W/m^2 resulting in 333 full load hours (40 kWh/m^2 x 1000 W/kW / 120 W/m^2) was rounded to 350 equivalent full load hours.

Present review study average load estimate is a bit higher at 45 kWh/m^2/y. The increase is due to:

- the changes in the model used to assess the EU air conditioner stock of appliances, in which the weight of Eastern and Northern countries have been decreased according to available information for these countries (information not available for these countries at the time of the Preparatory study),
- the increasing weight and CDD (as compared to Preparatory study estimate) for Italy (estimated to represent about 30 % of the stock in 2015), which in turn increases significantly the specific cooling demand for this country and the EU.

The 45 kWh/m^2/year corresponds to CDD 474 (Table 2) and to sizing values in the range 130 to 140 W/m^2. This leads to similar values as in present regulation: between 320 and 345 hours, it is then suggested to keep the present full load hour number of 350.

In conclusion, the number of equivalent full load cooling hours H_{CE} is kept constant at 350 hours for this review study.

3.1.2.4 SEER temperature profile

Temperature profile has been established by weighting the temperature distribution of cooling season hours of capital cities[18] by sales of split air conditioners according to JRAIA sales publications of 2006 supplied by EPEE, as explained in the Preparatory study Task 4 report[19]. Weighting ASHRAE IWEC data by the % sales distribution of split air conditioners by country in numbers. Weighted average implied CDD value for air conditioner installations in Europe was about 455.

Looking at the resulting distribution of hours by temperature bin in Regulation 206/2012, it is supposed that cooling starts at 16.5 °C (0 % load at 16 °C bin). This is based on the Preparatory study simulation results for residences and service buildings (the latter supposedly represented about 60 % of the stock in GW of installed units in 2006 according to the Preparatory study). The present input gives a 50/50 repartition. So, the situation is not so different.

On this temperature profile, it was suggested by stakeholders to study whether it was justified to rate cooling units with temperatures below 26 °C. Here are a few references on that point. In Australia, Hart and de Dear[20] data over 136 dwellings suggest that in the residential sector, 18 °C is a good candidate for zero loads in average. More recently, in France, a study on 10 dwellings in the South of France indicated first loads occurring between 15 (but with centralized water based system) and about 25 °C average outdoor air temperature, with most systems zero loads around 19 to 21 °C. (Kemna, 2014)[11] using average residential building characteristics and internal load suggests 20.5 °C could be a good minimum value (20.5 °C outdoor leads to an average of 25 °C indoors because of electric internal loads and solar gains equivalent to about 3.5 °C overheat inside the building). 20.5 °C is also the reference used in the ISO/FDIS 16358-1:2013 standard[21], which is also targeting mainly residential units. In the USA, the AHRI standard uses 18.3 °C.

Hence, available information suggests values between 18 °C and 21 °C for the zero load in residences.

[18] Spain climate selected was different here from before in Preparatory study: half Madrid and Barcelone (not half Madrid and Seville, which was considered to give too high average CDD values for Spain).
[19] Annex B.3 p 165 and below
[20] Melissa Hart, Richard de Dear, Weather sensitivity in household appliance energy end-use, In Energy and Buildings, Volume 36, Issue 2, 2004, Pages 161-174, https://doi.org/10.1016/j.enbuild.2003.10.009.
[21] See Task 1 for more information

However, for the service[22] sector, this temperature can drop to much lower levels. It is not uncommon to have cooling in large office buildings connected to district cooling at temperatures below 10 °C. The preparatory study simulations indicated balance points about 15 °C. So that the 17 °C (bin start 16.5 °C) was chosen as an average situation for buildings cooled by air conditioners. For residential air conditioners only as an average situation it may be too low. But being still at a 50/50 situation in installation proportions of residential products and service products, the 16 °C still seems adapted.

Regarding the number of hours per temperature bin, (Kemna, 2014)[11] studied a 3 month cooling season for Europe using population weighted average temperature bin for EU capital cities. The outdoor temperature for this weighted average is 23.2 °C, against 23 °C for the average temperature profile used in (EU) Regulation 206/2012. The evaluated monthly cooling degree days based on a simplified average model of a residential building has been calculated and is shown in Figure 12.

To build the temperature profile, cooling seasons of 3 months (Nordic countries) to 6 months (Southern countries) and 5 months for central Europe, have been used. This explains that the temperature profile, despite targeting southern Europe, gives more weight to lower temperature bins (in addition to the fact that base temperature is lower). If a residential metrics only were looked for, in addition to increasing the base outdoor temperature, the length of seasons in central European countries should probably be reduced to 3 to 4 months. This would tend to increase the average outdoor temperature of operation.

Other residential seasonal performance standard has indeed temperature profile corresponding to higher average outdoor temperature and load. What matters for SEER determination is the energy weighted average temperature and load of the profile (energy weights are obtained by multiplying load and number of hours):

- Regulation 206/2012 (50 residential / 50 services): energy weighted average load ratio 51 %, temperature 25.6 °C (eq. full load hours 350)
- AHRI 210/240 (residential): 27.8 °C, 57 % (equivalent full load hours 730)
- ISO 16348 (residential): 27.5 °C, 50 % (equivalent full load hours 800)

Weighted average energy values for temperature is lower in Europe as compared to USA and ISO standards, in agreement with milder average climate (lower equivalent full load hours) and to the fact that residential only is targeted.

[22] Service sector is sometimes also named tertiary or commercial sector and encompasses all non residential buildings except industrial ones.

Table 4: Evaluation of monthly residential cooling degree days base 20.5 °C[23]

	City, weather station	2013					2014							Total	Population %
		Aug	Sep	Oct	Nov	Des	Jan	Feb	Mar	Apr	May	June	July		
AT	Wien, Schwechat	85	7	0	0	0	0	0	0	1	11	44	65	213	1.6%
BE	Brussels, Zaventem	27	9	0	0	0	0	0	0	0	3	11	34	84	2.1%
BG	Sofia, Observ.	123	29	8	0	0	0	0	1	1	13	37	63	275	1.4%
CY	Larnaca, Cyprus	256	161	71	31	0	0	0	7	25	53	151	217	972	0.1%
CZ	Prague, Ruzyne	37	3	0	0	0	0	0	0	0	5	23	48	116	1.9%
DE	Berlin, Tempelhof	45	4	0	0	0	0	0	0	0	13	27	79	168	18.4%
DK	Copenhagen	10	0	0	0	0	0	0	0	0	1	2	44	57	1.2%
EE	Talinn, Talinn	12	0	0	0	0	0	0	0	0	14	4	48	78	0.3%
EI	Dublin, Dublin	0	1	0	0	0	0	0	0	0	0	1	3	5	0.7%
ES	Madrid, Barajas	208	96	16	1	0	0	0	2	22	40	103	164	652	7.0%
FI	Helsinki, Vantaa	15	1	0	0	0	0	0	0	0	15	6	60	97	1.2%
FR	Paris, Orly	48	15	1	0	0	0	0	0	0	2	20	52	138	13.3%
GR	Athens, Elefsis	265	135	33	5	0	0	0	1	9	57	156	232	893	1.9%
HR	Zagreb, Pleso	107	17	5	0	0	0	0	2	2	20	57	67	277	0.7%
HU	Budapest / Ferihegy	113	8	2	0	0	0	0	0	1	16	57	85	282	2.0%
IT	Roma, Ciampino	165	58	20	1	0	0	0	0	1	11	78	86	420	12.4%
LI	Vilnius, Vilnius	24	1	0	0	0	0	0	0	1	15	5	59	105	0.6%
LT	Riga, Airport	21	1	0	0	0	0	0	0	0	16	6	58	102	0.4%
LU	Luxembourg, LU	27	10	0	0	0	0	0	0	0	2	24	40	103	0.1%
MT	Valetta, Luqa	204	128	86	10	0	0	0	0	2	22	107	155	714	0.1%
NL	Amsterdam Schiphol	22	6	0	0	0	0	0	0	0	3	6	38	75	3.6%
PL	Warsaw, Mazowieckie	66	2	0	0	0	0	0	0	3	22	25	98	216	6.9%
PO	Lisbon, Rudela	106	74	16	0	0	0	0	1	4	23	42	58	324	1.8%
RO	Bucuresti / Imh	150	32	3	1	0	0	0	1	3	26	54	109	379	3.4%
SE	Stockholm (Arlanda)	15	2	0	0	0	0	0	0	0	8	2	54	81	2.1%
SI	Liubljana, Brnik	85	10	0	0	0	0	0	0	0	9	41	42	187	0.3%
SK	Bratislava, Ivanka	96	8	1	0	0	0	0	0	2	16	56	81	260	1.3%
UK	London, Heathrow	25	9	0	0	0	0	0	0	0	3	13	44	94	13.2%
EU	Total CDD	81	25	5	0	0	0	0	0	2	13	38	77	242	100.0%
EU	Average cooling season outdoor temperature at base 20.5 °C and 90 day cooling season, in °C													23.2	
EU	Average cooling season indoor temperature (including 4.5 K internal and solar gains), in °C													27.7	
EU	Average cooling season temperature difference ΔT with reference 25 °C, in °C													2.7	

Conclusively, the outdoor temperature profile for SEER computation corresponds to a mix of residential and service units. This leads to relatively low base temperature, high cooling season duration and relatively low average outdoor air temperatures and thus increases the rated SEER value. In the future, it is believed the mix of products will be more in favour to residential units and the metrics could then be updated to take this into account. If such a change was to happen however, it is highly recommended to plan a transition period during which both older and newer SEER values are published before to use the new profile to set label and MEPS requirements, given the sensitivity of the SEER values to the climate profile.

3.1.3 Heating loads and SCOP metrics parameters

The heating loads and SCOP metrics parameters from the preparatory study are discussed below and compared to other sources. This is done to check whether the assumptions should be updated.

[23] https://ec.europa.eu/energy/sites/ener/files/documents/2014_final_report_eu_building_heat_demand.pdf

3.1.3.1 Real use of reversibility

In the impact assessment study of (EU) Regulations 206/2012 and 626/2011, it was supposed based on stakeholder input that only 33 % of reversible air conditioners were used for heating[24].

In Commission decision 2013/114/EU[25], the percentage of reversible air conditioners supposed to be used for heating are: 10 % for Southern Europe, 40 % for Central Europe and 100 % for northern Europe (zones correspond to reference heating climates in (EU) Regulation 206/2012). The following explanation is given (footnote 10):

"*An Italian study (referred to on page 48 of 'Outlook 2011 — European Heat Pump Statistics') finds that in less than 10 % of the cases, heat pumps were the only installed heat generator. As reversible air-air heat pumps are the single most installed heat pump technology type (60 % of all installed units — mostly installed in Italy, Spain and France, as well as Sweden, Finland and Norway), it is important to adjust the figures appropriately. The Impact Assessment of Commission Regulation (EU) No 206/2012 of 6 March 2012 implementing Directive 2009/125/EC of the European Parliament and of the Council with regard to ecodesign requirements for air conditioners and comfort fans (OJ L 72, 10.3.2012, p. 7) assumes that EU wide, 33 % of reversible heat pumps are not used for heating. In addition, one can assume that a large number of the 67 % of reversible heat pumps are only used partly for heating, as the heat pump is installed in parallel to another heating system. The proposed values are therefore appropriate to reduce the risk of over-estimation*.".

Complementary information was received from stakeholders, which suggests that present rates of reversible units used for heating are higher, with residential rates between 20 % (only heating means) and 60 % (used in addition, partial replacement or replacement of an existing central systems) and commercial rates around 85 %. However, climate zone indications are not specified.

In this study, we will use the most recent available information in the Commission decision 2013/114/EU.

3.1.3.2 Heating load

In heating mode, it was decided at the time of the preparatory study to use the heating needs estimated in the preparatory study on space heating. The rationale is that reversible air conditioners installed for heating do compete with other heating means and should then be compared with the same heating loads, and that the heating needs identified from Lot 10 preparatory study are lower.

The equivalent full load hours have been defined based upon heating load and sizing estimates for the 3 climates.

[24] Impact Assessment [SWD(2012) 35]
[25] http://eur-lex.europa.eu/legal-content/EN/TXT/?uri=CELEX%3A32013D0114

Table 5: Equivalent full load hours calculation in Regulation 206/2012

Climate	Warm (Athens)	Average (Strasbourg)	Cold (Helsinki)
Heating load (from Lot 1) kWh/m2/y	71	87	148
Sizing at Tdesign in W/m2	71	87	96
Equivalent full load hours	1000	1000	1542
Correction of oversizing in Lot 1 (setback 1.2 and multizone 1.15)	1.38	1.38	1.38
Calculated equivalent full load hours	1380	1380	2128
Regulation 206/2012 equivalent full load hours (rounded)	1400	1400	2100
Implicit equivalent sizing at Tdesign in W/m2	51	62	70

Note that some stakeholders have proposed to modify the hours to the weighted average equivalent hours implied by the temperature distribution and load per bin. These values were already adopted for air-to-water heat pumps in EN14825:2016 standard (harmonised standard for (EU) regulation No 813/2013 which includes air-to-water and water-to-water heat pumps) and in Commission Decision 2013/114/EU[26] establishing the guidelines for Member States on calculating renewable energy from heat pumps from different heat pump technologies. However, these hours lead to very low equivalent sizing in W/m2 at design conditions, or conversely to very high heating loads on units:

- warm climate: 1335 hours versus 1400 hours, no correction done,
- average climate: 2065 hours versus 1400 hours, implicit sizing comes down to 40 W/m^2,
- cold climate: 2465 hours versus 2100, implicit sizing comes down to 60 W/m^2.

Sizing practice according to stakeholders in heating mode for air-to-air heat pumps are shown in Table 6.

Table 6: Stakeholder view of good sizing practice in heating mode for air-to-air heat pumps (capacities indicated at + 7°C standard rated capacities according to EN14511-2 standard)

Climate	Warmer	Average	Colder
Residential	80-100 W/m2	100-120 W/m2	120-180 W/m2
Retail	-	100 W/m2	-

To compare these values with design values, it is necessary to correct the heating capacity (Ph) at +7 °C given by the industry with the ratio Pdesignh / Ph, which according findings in Task 4 of this study is close to 0.85 or 3.4 kW at -10 °C for 4 kW rated heating capacity at +7 °C (3 kW at -7 °C declared in average as Pdesignh). Hence for average climate, implied stakeholder sizing condition is 85 W/m^2 at -10 °C, already much higher than the 62 W/m^2 in the regulation. Hence, if equivalent full load hours had to be modified, it would rather be to be lowered; for the average climate, equivalent full load hours would come closer to 1000 hours (close to the 954 weighted average hours identified from simulations in Lot 10 preparatory study, Task 4). Better insulation should also lead to lower heating equivalent hours in the future, but with lower sizing demand, resulting in a small decrease of equivalent full load hours; the impact of climate change should also lead to decrease the equivalent hours.

[26] http://eur-lex.europa.eu/legal-content/EN/TXT/?uri=CELEX%3A32013D0114

Hence, it does not seem justified to increase the number of equivalent full load heating hours. And if a change had to be made in the coming years, it would rather be to decrease the number of equivalent full load hours.

3.1.4 Crankcase heater in SEER and SCOP metrics

In the Regulation 206/2012, outdoor temperature for measurement of the crankcase heater power value was not specified (nor for other modes, but there should be no dependency to outdoor air temperature for other modes), although crankcase heater operation may be controlled as a function of outdoor air temperature (amongst other variables).

The reason for introducing the operation mode in the regulation was because it appeared frequently that the crankcase heater start operation as soon as the compressor stop, independently of the outdoor temperature, while operation is only required below a certain outdoor temperature point.

In standard EN14825:2016, crankcase heater consumption is measured in test condition D in cooling mode, i.e. at 20 °C outdoors (for heating only units this is 12 °C). This implies that for units with temperature control, crankcase heater is very low or zero. Only for units without a temperature control, crankcase value measured this way is high. Some stakeholders have indicated some units with temperature control of crankcase heaters between 12 and 20 °C, while others have more sophisticated control based on both indoor and outdoor temperature.

In the cooling season, crankcase heater hours are per design hours with temperature below 16 °C (for the thermostat-off period) and others when the unit is in standby mode with about 23 °C average (10/24 of the 2602 hours of cooling). In average, using 20 °C for crankcase hours of reversible units in cooling mode is acceptable.

In the heating season, for standard inverter controlled reversible air conditioners, as per standard temperature profile, the compressor is on most of the heating season. Nevertheless, for some part load hours above 7 °C most units still cycle on and off and crankcase consumption appears. Thermostat-off hours are hours with temperature above 16 °C. In average, using 12 °C for crankcase hours of reversible units in heating mode is acceptable.

It is then proposed as a first improvement to require two distinct crankcase power values, respectively for cooling mode - measured at 20 °C - and for heating mode - measured at 12 °C. Cooling (Heating) only unit crankcase power measurement should be done at 20 °C (12 °C), as already written in standard EN14825:2016. This modification is already included in draft standard prEN14825:2018.

3.1.4.1 Hours of operation per mode

The hours of operation adopted to the current study is presented in Table 7 and Table 8 for cooling and heating, respectively.

In Table 7 the hours per cooling mode is presented in the current regulation and the suggested changes. This table also includes other power modes such as standby.

Table 7: Hours per mode in cooling mode in Regulation 206/2012 and proposed changes

	Regulation 206/2012		Proposed changes	
Hours	**Mode**	**Value**	**Changes**	**Value**
Hours per year		8760		
Cooling season 5 months		3672		
Hours in standby mode (14/24*3672)	H_{SB}	2142		
Hours in other modes (active cooling or thermostat off 3672 - 2142)		1530		
Number of hours with outdoor temperature above 16 °C		2602		
Number of hours with outdoor temperature above 16 °C during operation period (10/24*2602)		1084		
Number of hours with outdoor temperature below 16 °C during operation period (=1530-1084) during the season		446		
Weighted average number of hours with outdoor temperature below 16 °C during operation period during the season	H_{TO}	221		
Crankcase hours	H_{CK}	2672	Change to H_{SB} + H_{TO}	2363 (=2142+221)

From the table it is visible that the only proposed changes are the crankcase heater hours (H_{CK}) that should be adjusted to the sum of hours in standby (H_{SB}) and weighted average number of hours with outdoor temperature below 16 °C during operation period during the season (H_{TO}), which gives a value of 2363.

For network connected products, standby is in fact network standby as it is in "networked standby", which according to Regulation (EU) No 801/2013 "means a condition in which the equipment is able to resume a function by way of a remotely initiated trigger from a network connection". It is therefore assumed that network standby hours are the same as standby hours in Table 7.

For the hour of heating no remarks was received regarding the low power mode hours for reversible units; that might be because the present impact of accounting for these modes on the SCOP is low (between 0.1 and 0.5 % according to available data in Task 4).

An issue in Regulation 206/2012 regarding heating mode hours of heating only units has been raised. Correction is proposed in Table 8 below.

Table 8: Correction of power mode hours of heating only heat pumps (values to be corrected in red)

	Climate	Warm	Average	Cold
Heating period	Heating season hours	4344	5088	6576
	H_{TO}	755	179	131
	H_{SB}	0	0	0
	H_{OFF}	0	0	0
	H_{CK}	755	179	131
Cooling season hours	Cooling season hours	4416	3672	2184
	H_{TO}	0	0	0
	H_{SB}	0	0	0
	H_{OFF}	4416	3672	2184
	H_{CK}	4416	3672	2184
Heating mode, if appliance offers heating only	H_{TO}	755	179	131
	H_{SB}	0	0	0
	H_{OFF}	4416	3672	2184
	H_{CK}	5171	3851	2315

In conclusion:

- Minor change (regarding crankcase hours) can be incorporated into a revised regulation as its impact on the SCOP value is supposedly low.
- There may be a need in the future to update the temperature profile for cooling to account for the larger proportion of residential units in the installations. However, in that case, it is strongly advised to have a transitional period with the two values published by all manufacturers so that proper labelling / MEPS scheme can be established.

3.1.5 Bivalent temperature in SCOP metrics

In Regulation (EU) no 206/2012, manufacturers choose a bivalent temperature; this is the outdoor temperature for which heating capacity of the unit matches building load. In general, this bivalent temperature is higher than the design heating outdoor temperature (Tdesignh) of the specific climate. It means that a backup heater needs to be used in addition to the unit at low ambient to reach the building load Pdesignh.

To design a real installation, the optimal bivalent point is the result of a life cycle cost minimization. For low bivalent point temperature, the share of the backup heater is lower. This means the system efficiency is high because backup heater energy consumption is low (few hours or days of use and so low consumption despite low efficiency). However, the system cost increases as the heat pump cost is proportional to its size and the heat pump price is higher than the backup price. Conversely, for high bivalent point temperature, the system efficiency decreases because of the increasing share of the backup heater and the cost decreases. So, purchase price is lower, but energy consumption is higher.

The optimal bivalent point is dependent on total heat load, load curve, heat pump performance curve, capacity and cost, backup fuel type and efficiency and energy price(s) (backup may consume another energy type than electricity).

Because of the infinite diversity of real life situations, real life bivalent temperatures may vary significantly. Therefore, as a standard value, depending on each manufacturer strategy, product, application, climate, it may be of interest to market lower price and efficiency to decrease the installation cost and thus show higher Pdesignh for the same capacity even if SCOP value is decreased.

To the knowledge of the study team, Europe is the only economy to apply a freely declared bivalent temperature, which in general are fixed values associated to climates defined in standards. And this seems justified given the diversity of climates, energy prices and main heating system types.

Present bivalent point allowance in Regulation (EU) no 206/2012 is as described in Table 9

Table 9: Bivalent and operation limit temperatures according to climate in Regulation (EU) no 206/2012

Climate	Tdesignh	Tbivalent	Operating limit
Average	– 10	max. 2	max. – 7
Warmer	2	max. 7	max. 2
Colder	– 22	max. – 7	max. – 15

Based on ECC database of certified products[27], it is possible to have an idea of the bivalent choices made by manufacturers (these are model based statistics, not sales weighted based).

For average climate, over 2851 product declarations for Tbivalent, only 18 (0.6 %) have bivalent temperatures above -7 (5 units use -3 °C, 3 units -4 °C, 1 unit -5 °C, 5 units -6 °C). 963 units (34 %) have bivalent temperature lower than -7 °C.

Other climate information is not supplied in public ECC database, but more information supplied by ECC was used to compute the same statistics for other climates[28]:

- For warm climate: out of a total of 2293 units, only 322 units have declarations for warm climate, amongst which 39 (12 %) have bivalent temperatures strictly higher than 2 °C (37 units use 5 °C and 2 use 6 °C).
- For cold climate: out of a total of 2293 units, only 151 units have declarations for cold climate, amongst which only 30 (20 %) have bivalent temperatures strictly lower than -7 °C (down to -20 °C for 3 % of units). Interestingly, declarations with lower than -7 °C bivalent temperatures are more frequent for the average climate than for the cold climate declared products.

So it can be seen that the variable bivalent temperature option is used: quite commonly for cold climate and warm climate, and to a less extent for the average climate for which the very large majority of products are declared for lower or equal to -7 °C bivalent temperatures.

Comments have been received from Northern European EU and EEA Member States that propose to increase the bivalent point at least for cold climates so that the Pdesignh declaration is closer to the real capacity delivered by the unit. It can be easily understood that for a bivalent temperature equal to Tdesignh, the capacity of the unit matches Pdesignh. Nevertheless, installing a low capacity heat pump to cover only half of the

[27] Extract from Nov 2016, http://www.eurovent-certification.com.
[28] See Task 4, paragraph 4.1.1, for more information on the ECC database used in this study.

heating needs or less may still be economical in some situations and the manufacturer may want to market such situations. So, this proposal does not seem in the interest of the end-user.

The problem is rather if the manufacturer at the same time can publish high SCOP values because choosing a high Tbivalent and showing a high heat pump capacity. In this matter, the indication of the backup heater capacity required to reach Pdesignh capacity indicated together with the Pdesignh chosen and the corresponding SCOP as proposed in Task 1 report should help in making the situation more transparent for the end-user.

3.1.6 Case of portable air conditioners

For portable air conditioners the most important factor to consider is the impact of air infiltration. Infiltration of air has been discussed in previous tasks and the impact is discussed in this subsection. Another important factor to discuss is the hours of use and how portable air conditioners are used (as supplementary equipment or instead of e.g. fixed air conditioners).

3.1.6.1 Different types of double duct air conditioners

There are two distinct types of double duct products:

- portable units, that are similar to single duct units but with two ducts

- units with fixed installation; one or two holes are to be made in a wall; these units can operate all year round, providing often both heating and cooling; they look like a radiator and for some models a supplementary indoor unit can be plugged to make a multi-split. These units compete with standard split systems or window / through the wall air conditioners and with other traditional heating means when reversible, as electric radiators. Thus, double duct units with fixed installation are to be considered in the regulation as standard air conditioners. The category double duct should refer to portable air conditioners only (and possibly renamed / redefined categorisation in the regulation is to be proposed). Consequently, SEER and SCOP metrics and similar energy labelling scale should be used for those products. Bivalent point temperatures probably need to be adapted.

What follows regards only portable air conditioners, with one or two ducts.

3.1.6.2 Updated information regarding portable air conditioners

Single duct air conditioners are peculiar because their inlet air at condenser is indoor air. To compensate this air flow, infiltration of outdoor air in the building is necessary.

Simulations in preparatory study were done to simulate the infiltration impact on cooling performance using dynamic simulations. However, there was no common consensus on the values of the preparatory study and there are consequently no hours in (EU) Regulation 206/2012 to compute energy consumption of portable air conditioners in cooling mode, nor for other modes. Manufacturers were mainly concerned by the complicated testing method and by the high number of hours in thermostat-off and standby mode.

Although it was not identified in the Preparatory study, the same effect has been highlighted for double duct products by the US DOE (see Task 1), that showed a difference between inlet and outlet condenser air flow rates and thus an infiltration effect albeit lower than for single duct units. In the USA, portable air conditioners are tested according to the enthalpy method. In Europe, they are tested using the calorimeter room method. As temperatures in the two rooms are different in the standard rating test of double duct air conditioners, the infiltration is already accounted for in the rated performances of the double duct products. The same is true for window and through the wall air conditioners, for which infiltration also occurs.

As reported in Task 1, the US DOE proposed a quite simple method to rate the seasonal performance in cooling mode for portable air conditioners. But to compute the infiltration effects, it is necessary to measure the air flow at the condenser exhaust for single duct air conditioners, and at condenser inlet and exhaust for double duct units. This is not necessary to measure infiltration for double duct because this is already taken into account in the test method. For single duct, it is possible to make a supplementary test only to measure the condenser (cooling mode) air flow, according to EU test laboratories.

A US study by LBNL[29] for US DOE reports the field metering results (electricity consumption broken by compressor mode/fan mode/standby and off mode) of 19 portable air conditioners (named hereafter Burke study). It appears that fan mode (which is a thermostat-off mode according (EU) Regulation 206/2012 wording) power consumption above 100 W (average value for products of comparable size to this study base case). This is thus certainly a mode that should be integrated in regulation.

In addition, thermodynamic heating (reversible portable air conditioner using the electric vapor compression cycle in reverse mode for heating and not only an electric resistance), which had not been identified in the Lot 10 preparatory study, has been integrated in the Regulation 206/2012; efficiency is rated by the unit COP rated at max capacity and at 20 °C indoor (so at both condenser and evaporator inlets) using the calorimeter room method (see Task 1 section 1.2.1.1; this method is described in the paragraph dedicated to EN14511 standard).

3.1.6.3 Thermodynamic heating with portable air conditioners

To evaluate the importance of infiltration on the functional performance of the unit and on its efficiency, it is necessary to fix typical infiltration values; these are retrieved from US DOE tests presented in Task 1 (section 1.2.1.4: US DOE 10 CFR Parts 429 and 430. Energy Conservation Program: Test Procedures for Portable Air Conditioners; Final Rule):

- Single duct: typically 200 $m^3/h/kW$ cooling (measured air flow per declared kW cooling)
- Double duct: typically 70 $m^3/h/kW$ cooling (measured air flow per declared kW cooling)

Apparently, portable units have similar heating and cooling capacities, so that we use the average size of 2.6 kW; COP is 3 for both units. It is supposed that the unit is sized according to its rated heating capacity, which is likely given that this is the only information available to the end-user for these products. With these values it is possible to correct the capacity measured in standard EN14511-3 conditions with infiltration. This is done for single duct and double duct air conditioners in heating mode in Figure 13 and in Figure 14 respectively.

[29] T. Burke, et al., Using Field-Metered Data to Quantify Annual Energy Use of Portable Air Conditioners, Lawrence Berkeley National Laboratory, Report No. LBNL-6868E (December 2014). Available at: www.osti.gov/scitech/servlets/purl/1166989

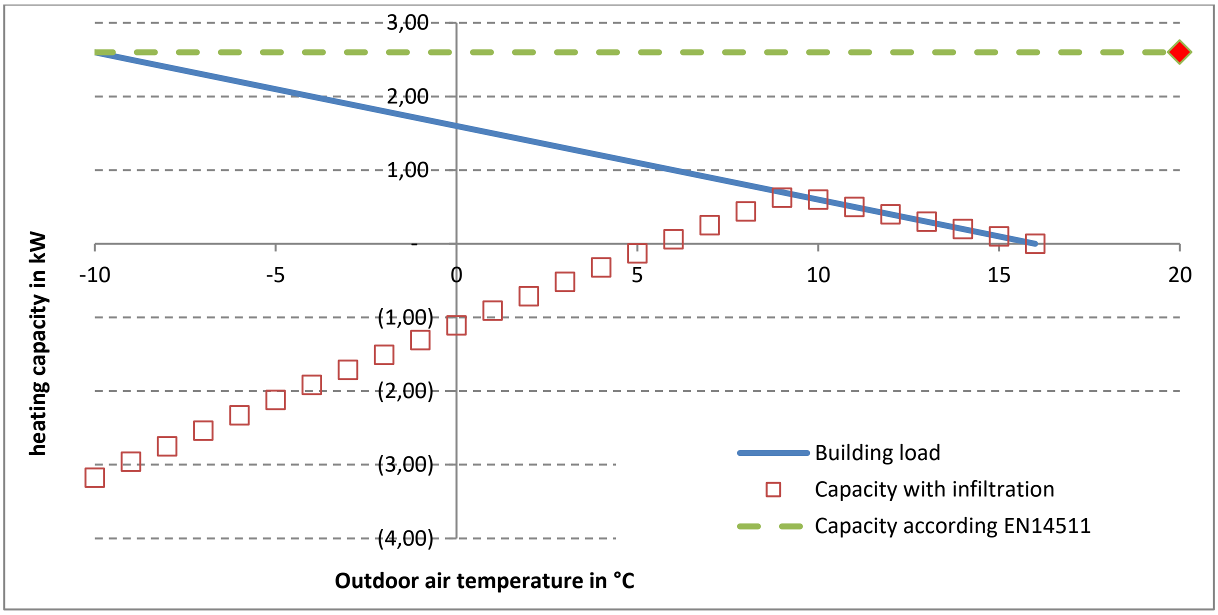

Figure 13: Impact of infiltration on heating capacity of reversible single duct unit in average climate. Red point is EN14511 rated heating capacity.

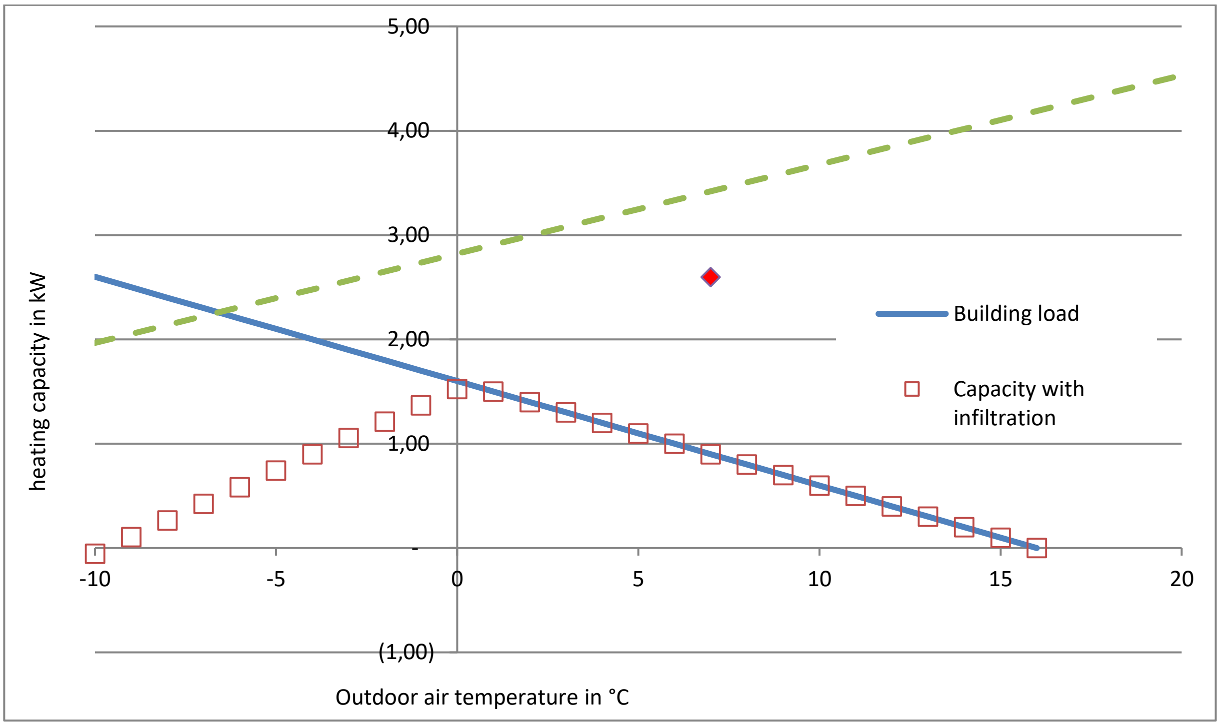

Figure 14: Impact of infiltration on heating capacity of reversible double duct unit in average climate. Red point is EN14511 rated heating capacity.

To plot these figures, it is assumed that the indoor air temperature is maintained at the indoor temperature set point, i.e. that another heating system compensates in case of negative net balance of the reversible portable air conditioner. For double duct, a capacity slope of 2.5 % per degree is used to calculate heating capacity. In addition, to the heating capacity, the net SCOP of the unit is calculated for average and warm climate; a Cd factor of 0.25 as in EN14825:2016 standard is used to correct the COP when the net unit capacity (unit rated capacity minus infiltration) is higher than the building load. The result is as follows:

- for single duct: below 10 °C, the unit cannot cope with the heating load; below 6 °C, the unit cools the building; the average SCOP if working all year long is negative (-0.4) for the average climate and equals 0.9 for the warm climate.
- for double duct: below 1 °C, the unit cannot cope with the heating load; but its contribution remains positive down till - 9 °C, reaching - 0.1 kW at -10 °C versus 2.6 kW capacity bought; the average SCOP if working all year long (but it does not satisfy the heating load below 1 °C) is 2.0 for the average climate and 2.9 for the warm climate.

Hence, reversible single duct operation in heating mode is likely to be worse than electric heating. Double duct operation is better in terms of performances, still it cannot deliver the capacity that the customer is buying.

Manufacturers suggest these products are used only in mid-season, when people are waiting for central heating systems to start, but there is no data available to sustain this claim. In the meanwhile, these products are competing with electric portable heaters and possibly in some cases with fix reversible air conditioners and the information to choose product based on functionality (here heating capacity) and on the often used "heat pump" wording advertisement is highly misleading.

Consequently, the present rating system for heating mode of portable products in Regulation (EU) 206/2012 and the associated label in Regulation (EU) 626/2011 is to be revised.

For single duct, estimated seasonal performance coefficient lies between -0.4 for average climate and 0.9 for the warm climate. In order to keep the thermodynamic benefit for highest outdoor temperatures and to maintain heating capacity at low outdoor temperature, a hybrid version combining a 2.6 kW electric resistance in addition to the 2.6 kW thermodynamic heating system can be added. However, this gives a SCOP of 1.005 for average climate and 1.09 for warm climate (below 10 °C, only the electric resistance operates).

Thus, SCOP of thermodynamic heating with single duct is presently well below 1 and including an electric heater could only bring little improvement over the present archetype, with SCOP only slightly better than 1. It then seems useless to use single duct thermodynamic heating, as an electric resistance alone, with a SCOP of 1, would be just as efficient for the end-user. A simple option is then to ban thermodynamic heating for single ducts.

But, for double duct products, a seasonal performance metrics could be developed.

For double duct, it is necessary to use the same scheme as for fixed air conditioners (SCOP measurement and calculation) and to declare comparable heating capacity and performance at design temperatures (note that infiltration impact is taken properly into account for double duct tests following EN14511-3). Because of the negative capacity at -10 °C, an evolution could be to add a complementary electric resistance. Bivalent point temperatures should be adapted in EN14825 for these new products.

For single duct thermodynamic heating, manufacturers may want to pursue the development of such reversible products. In that case, EN14825 procedure should be adapted to include the impact of infiltration on capacity and performance. Air flow rate of evaporator (in heating mode) should be measured in addition to performance test to enable capacity and SCOP calculation. Bivalent point temperatures should be adapted in EN14825 for these new products. While waiting for a standard to measure the air flow rate, default values should be used, as the ones indicated above. It should be noted to the purpose of correcting capacities by infiltration, large uncertainties on the evaluation of the infiltration air flow may still give a better information on the unit that present rating values without the infiltration impact.

3.1.6.4 Capacity and performance in cooling mode for portable air conditioners

In the EU, the same requirements, 2.6 minimum EER value (Regulation 206/2012) and same energy label grades (Regulation 626/2011) are used for single duct and double duct despite significant differences in test procedures:

- Single duct air conditioners are tested at 35 °C dry bulb (24 °C wet bulb) both indoor and outdoor while double duct are tested like other air conditioner types at outdoor conditions 35 °C dry bulb (24 °C wet bulb) and indoor conditions 27 °C dry bulb (19 °C wet bulb).
- Infiltrations are accounted for in the performance tests of double duct air conditioners but not for single duct air conditioners.

It is of interest to notice that portable double duct products seem to have disappeared in Europe while these are still present in the USA. In the USA the rated performances are different: temperature conditions are 27/27 for single duct in the USA instead of 35/35 in Europe and 35/27 for double duct in the USA, same as in Europe, but infiltration is not taken into account in present rating for double ducts in the USA while it is accounted for in Europe. The USA regulation proposal[30] is now to include infiltration for both types of products and to rate the performance of products at equal outdoor temperature conditions. This goes into the direction of ensuring a level of playing field between both types of products and also to better inform the consumer by integrating infiltration impact.

In addition, it has been commented by EU test laboratories that some units can be supplied with one or two ducts so that the end-user may choose between both operating modes, albeit not having information on the consequences of operating in the two modes. In that case, it is clear that manufacturers should respect requirements for both types of configuration and supply corresponding information. The impact for the end-user is analysed hereafter.

Impact of part load and infiltration on seasonal performance

It is thus required to explore the impact of the differences in rated performances of portable air conditioners with one or two ducts and to evaluate the impact of infiltration (which are already part of the difference between both product types).

As infiltration depends on outdoor air temperature and humidity, it is necessary to show the impact of infiltration on a typical profile of use. A typical profile of use is similar to other air conditioners but keeping in mind the specifics of portable air conditioner use and considering only the residential sector (about 80 % of sales).

To that purpose, the average bin hours distribution of the Regulation 206/2012 is used but cut below an outdoor temperature of 23 °C (starting bin 22.5 °C), which according to (VHK, 2014)[11], corresponds to an average temperature of 26 °C indoor in residences, and can then be considered as the starting point of cooling for these units. Starting to cool at higher

[30] USA Federal Register, Vol 81, No. 105, Wednesday June 1, 2016.

outdoor (and thus indoor) air temperature than for fixed air conditioning installations translates the fact that most units are residential and that end-users may be reluctant to use these units unless really needed because of higher noise (compressor and 2 fans indoors). Hours per bin are shown in Table 10.

Table 10: Bin number j, outdoor temperature Tj in oC and number of hours per bin hj corresponding to the reference cooling season

j #	0	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17
Tj °C	23	24	25	26	27	28	29	30	31	32	33	34	35	36	37	38	39	40
hj h	218	197	178	158	137	109	88	63	39	31	24	17	13	9	4	3	1	0

By definition, 0 % building load condition occurs at an outdoor temperature of 23 °C. 100 % load is supposed to occur at 35 °C. However, the capacity corresponding to this maximum load is presently different for single and double duct products:

- For single duct: the rated cooling capacity is measured at 35 °C db / 24 °C wb without accounting for the impact of infiltration. The sizing decision is however likely to be based on this value (as it is the only one available) or on indications supplied by manufacturers or salesmen based on room size equivalence ("this air conditioner is adapted for a room of XX m^2").
- For double duct: the rated cooling capacity is defined in the same conditions as for fix air conditioner types and so the rated cooling capacity is also the maximum cooling load at 35 °C outdoor condition but including infiltration.

Hence building load can be written as:

$$BL(Tj) = P_{designc} \times PLR(T_j) = P_{designc} \times (T_j - 23) / (35 - 23) \qquad \text{(Eq 1)}$$

with Pdesignc = Pc$_{rated}$, the portable rated cooling capacity.

In order to compute the capacity and efficiency of these units along the building load curve, it is necessary to compute the impact of infiltration on the unit capacity. This is done using a method similar to the one used by the US DOE to account for infiltration and also by integrating the potential variation of the cooling capacity with Tj.

$$Pc\_corr(Tj) = Pc(Tj) + Q_{INF}(T_j) \qquad (Eq\ 2)$$

Where:

- *Pc_corr(Tj): maximum capacity of the unit corrected with infiltration*
- *Pc(Tj): maximum capacity in bin Tj without accounting for infiltration*
- $Q_{INF}(T_j)$ *: Heat loss by infiltration (W)*

The infiltration impact is calculated with the following formulas:

$$If\ Tj < 27, Q_{INF}(T_j) = \frac{27 - T_j}{27 - 20} \times \left[AF \times \left(rho_{air_{27}} \times h_{27} - rho_{air_{20}} \times h_{20}\right)\right]$$

$$If\ Tj > 27, Q_{INF}(T_j) = \frac{27 - T_j}{35 - 27} \times \left[AF \times \left(rho_{air_{35}} \times h_{35} - rho_{air_{27}} \times h_{27}\right)\right]$$

Where:

- $Q_{INF}(T_j)$ *: Heat loss by infiltration (W)*
- *Tj: outdoor temperature of bin j*

- *AF: infiltration air flow (m³/s)*
- $rho_{air_{20}}$ *= 1.20 kg / m3, density of dry air at 20 °C (1 atm)*
- $rho_{air_{27}}$ *= 1.17 kg / m3, density of dry air at 27 °C (1 atm)*
- $rho_{air_{35}}$ *= 1.15 kg / m3, density of dry air at 35 °C (1 atm)*
- h_{20} *= 42.2 kJ/kg$_{da}$ specific enthalpy of infiltration air at 20 °C dry bulb and 15 °C wet bulb temperature per kg of dry air*
- h_{27} *= 54.2 kJ/kg$_{da}$ specific enthalpy of infiltration air at 27 °C dry bulb and 19 °C wet bulb temperature per kg of dry air*
- h_{35} *= 72.5 kJ/kg$_{da}$ specific enthalpy of infiltration air at 35 °C dry bulb and 24 °C wet bulb temperature per kg of dry air*

In order to show the impact of infiltration on capacity and EER, 4 different cases are considered:

- A: single duct base case (EER 2.65; Pcrated = 2.6 kW)
- B: double duct (EER 2.65; Pcrated = 2.6 kW)
- C: same as A but declared and used as double duct

A. Case of single duct

The indoor set point is rather 27 than 35 °C. Thus the capacity to be used at 27 °C is in fact lower than 2.6 kW and EER is higher[31]: capacity is about 2.43 kW and EER of about 3 at 27/27. Average infiltration air flow is of 200 m3/h and per kW of rated cooling capacity (as per EN14511-3).

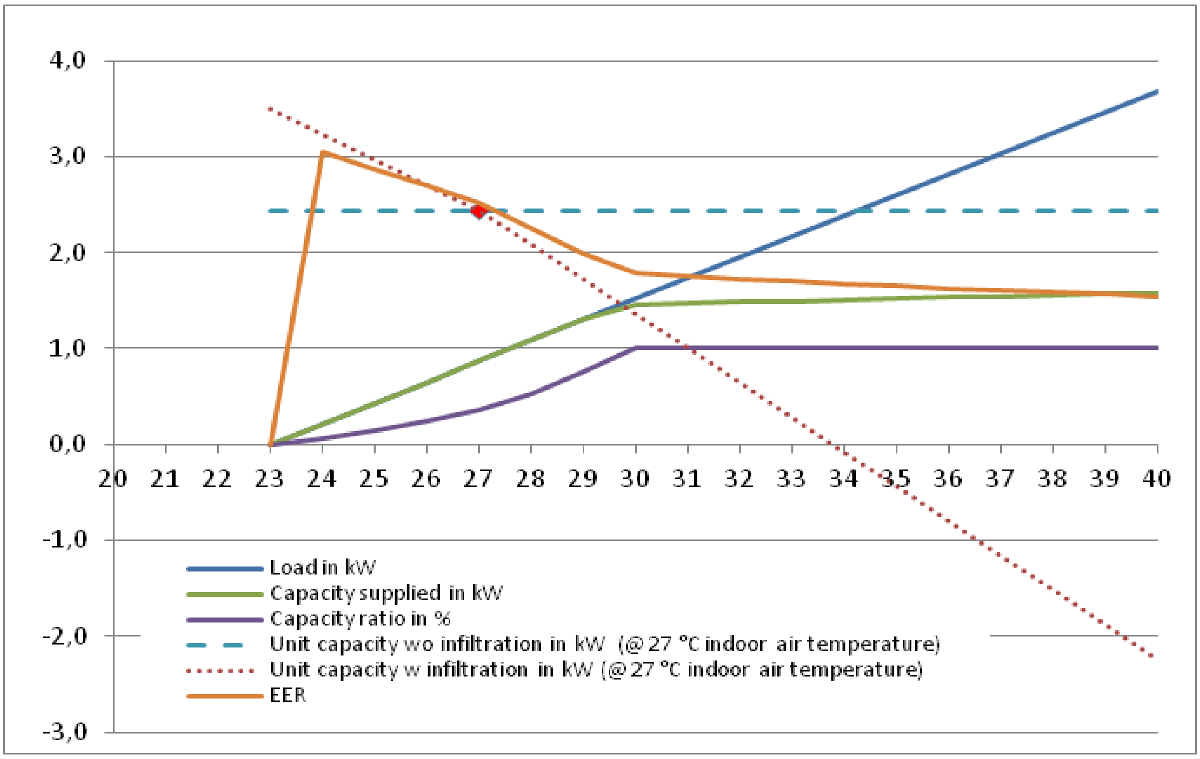

Figure 15: Evolution of load, capacity supplied, capacity with and without infiltration, capacity ratio and EER for base case single duct

[31] According to CECED information supplied during the Preparatory study regarding the variation of capacity and electric power at 27/27 and 35/35.

The Figure 15 shows the evolution of different variables for each temperature bin:

- the building load is a straight line between 0 kW at 23 °C and 2.6 kW at 35 °C.
- unit capacity at constant indoor temperature of 27 °C and without infiltration impact is constant and equal to rated capacity, here 2.44 kW.
- unit capacity corrected by infiltration is equal to unit capacity at 27 °C as the effect of infiltration is 0 kW; but it decreases sharply when outdoor temperature increases. At about 30 °C and 1.5 kW, the unit capacity corrected of infiltration is equal to the load for this temperature (equilibrium point). This intersection is computed by equalizing equations 1 & 2. It is interesting to note that in the US DOE regulatory project, the final capacity accounting for the effect of infiltration corresponds to the capacity at 29.7 °C (close to this intersection). This value is also observed in the Burke study. In Figure 16 below from this study, when the time spent in fan mode goes to zero, it means that the unit compressor is running continuously and so from that temperature and above the cooling load cannot be met. The point with nearly zero fan mode time occurs at temperature between 29.4 °C (85 °F) and 32.2 °C (90 °F).

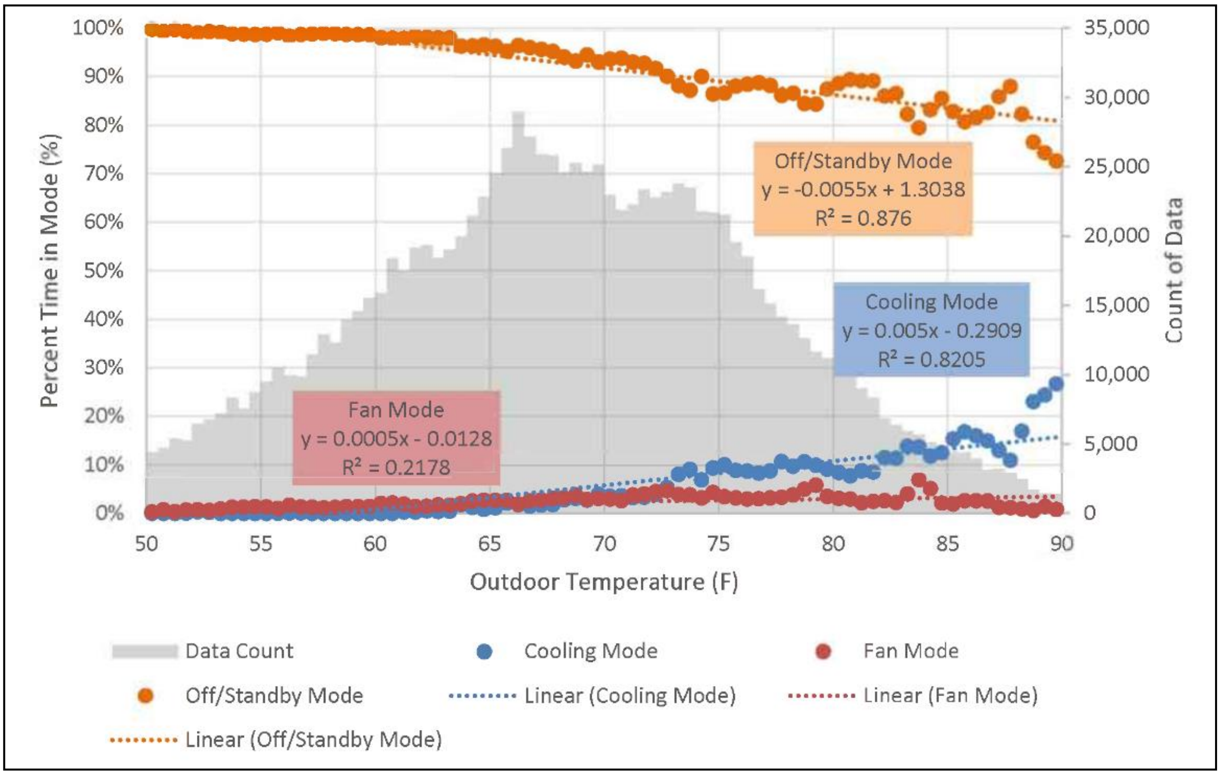

Figure 16: Hours of use per mode in the Burke study for residential portable air conditioner sites

- Above the equilibrium point temperature, the compressor is running continuously but the indoor set point of 27 °C cannot be maintained and indoor temperature begins to increase. On Figure 15, the capacity slightly increases because of the higher indoor temperature. At 35 °C outdoor air temperature, the indoor temperature reaches about 32.3 °C. So that the base case single duct is only able to maintain about 2.7 °C indoor / outdoor temperature difference.
- To compute the indoor air temperature equilibrium temperature above the intersection, equations 1 and 2 are modified by replacing the set point 27 °C by the equilibrium indoor air temperature ITj to be identified, as follows:

$$BL(Tj) = P_{designc} \times PLR(T_j) = P_{designc} \times (T_j + (ITj - 27) - 23) / (35 - 23) \ (Eq\ 1\ bis)$$

$$Qc\_corr(Tj) = Qc(Tj) + Q_{INF}(Tj) \ (Eq\ 2\ bis)$$

$$With,\ Q_{INF}(T_j) = \frac{ITj - T_j}{35-27} \times \left[AF \times \left(rho_{air_{35}} \times h_{35} - rho_{air_{27}} \times h_{27}\right)\right]$$

In conclusion, regarding capacity, present rated capacity is 2.6 kW, but the unit can only supply 1.5 kW at 2.7 K temperature difference indoor / outdoor. This information is not communicated to the end-user today, which thinks capacity indicated for single duct is comparable to kW cooling indications for fixed installations or mobile double ducts.

Regarding performances, infiltration modifies the capacity of the unit and modifies the capacity ratio and so the part load performances. This needs to be accounted for to compute seasonal performances.

The load ratio is computed by dividing the building load by the unit capacity corrected with infiltration below the equilibrium point. Above that it is equal to 1. It can be seen that below the equilibrium point, the capacity ratio of the unit is no longer a straight line because of the infiltration effect. EER is computed based on the capacity supplied and the electricity consumption of the unit, corrected by a cycling coefficient Cdc of 0.25 when load is lower than the supplied capacity. A SEERon value can be computed with this method using the following formula:

$$SEER_{on} = \frac{\sum_{j=1}^{n} hj \times Pc(Tj)}{\sum_{j=1}^{n} hj \times \left(\frac{Pc(Tj)}{EER(Tj)}\right)}$$

Where:

- *Tj = the bin temperature*
- *j = the bin number*
- *n = the amount of bins*
- *Pc(Tj) = below equilibrium point: the cooling demand of the building for the corresponding temperature Tj; above the equilibrium point: the capacity of the unit for the corresponding temperature Tj*
- *hj = the number of bin hours occurring at the corresponding temperature Tj*
- *EER(Tj) = the EER values of the unit for the corresponding temperature Tj.*

The SEERon value reached is of 2.13 versus 3 presently rated EER at 27/27 or a 29 % gap. The net effect of infiltration on SEERon is a 23 % gap, cycling then representing a 6 % loss.

B. Case of standard double duct

Double duct air conditioners need to reach an EER of 2.6 to satisfy Regulation (EU) 206/2012. We use here the same capacity as for single duct of 2.6 kW (at conditions outdoor 35 °C / indoor 27 °C) and same efficiency of 2.65. SEERon reaches 3.60.

Evolutions of capacity, load and load ratio, and efficiency with outdoor temperature are shown in Figure 17.

It is interesting to note that this unit if tested at 27 °C / 27 °C would have a capacity of 4.6 kW and an EER close to 5.15 (note it is lower on Figure 17 because of cycling losses). For matter of comparison, the scale of EER values on the EU market is 2.6 - 3.6. Thus, despite the minimum performance level for EER for single and double duct is the same, this corresponds to very different units because of the differences in test conditions:

- the same published capacity for single duct and double duct products in fact correspond to a double duct unit with a size being 190 % (4.6/2.4) of the single duct unit;

- the same published EER for single duct and double duct products in fact correspond to a double duct unit with an EER being 170 % (5.15/3) of the single duct unit; the SEERon would be also 170 % (3.6/2.13) of the single duct unit.

Conversely, no single duct on the market fitted with a second air duct could reach the double duct minimum performance requirement using the present standard.

Hence, the present rating system needs to be modified as it does not give a level playing field for single duct and double duct air conditioners.

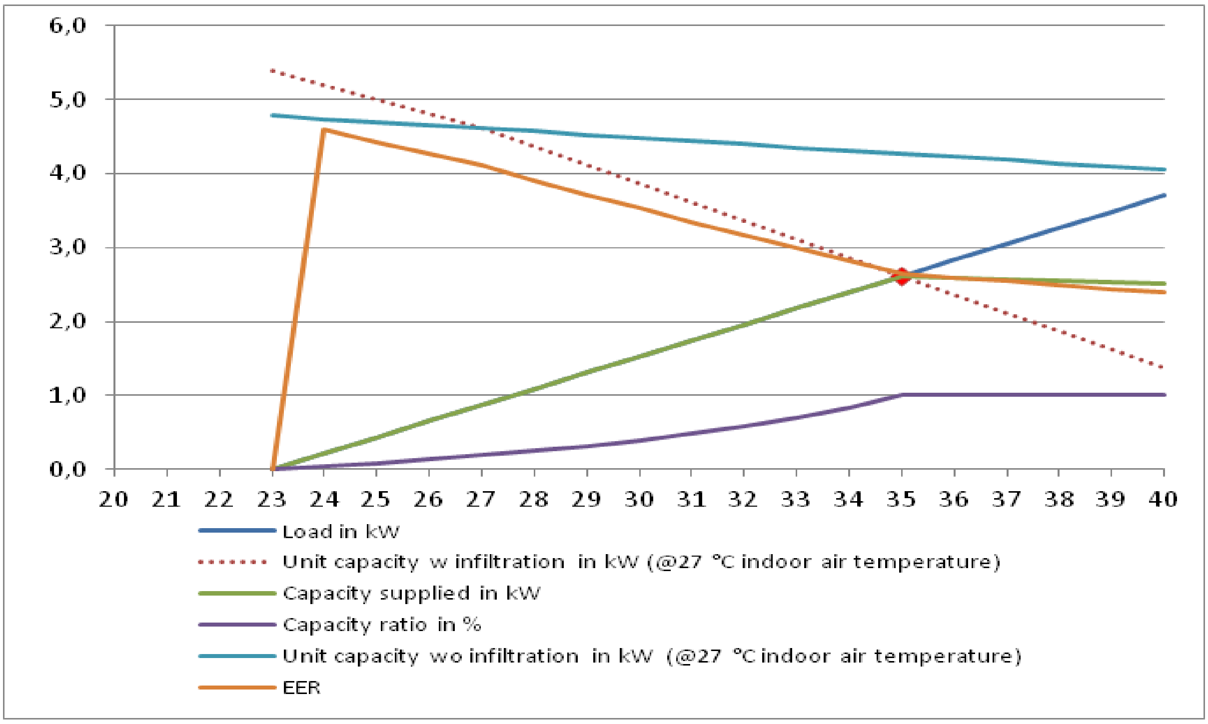

Figure 17: Evolution of load, capacity supplied, capacity with and without infiltration, capacity ratio and EER for standard double duct in the EU

C. Case of single duct (unit in case A) declared as single duct and used as double duct

In case of products being sold with 2 duct connections and that can be operated as single duct or double duct, it is likely that manufacturer declare their performances as the ones of a single duct only (as rating conditions are more favorable). We use the base case single duct product of case A of rated capacity 2.6 kW and EER of 2.65 (at 27/27, capacity is 2.44 and EER is 3). A supplementary consumption of 30 W is accounted for the fan because of the supplementary duct pressure losses, which reduces the EER at 35/35 to 2.57. Sizing would be done at 2.6 kW. Thus, the load treated by the double duct is much higher if declared as single duct than if declared as double duct. Despite being the same unit, this difference in sizing makes a difference. The standard rated capacity reached for the product in double duct configuration is of 1.4 kW accounting for the effect of infiltration using 70 m3/h/kW (the kW used here is the one of the rated capacity at 35 °C without infiltration). The capacity at 35 °C (outdoor) / 27 °C (indoor) without infiltration is of 2.3 kW. Indeed, capacity is supposed to decrease of 1 % per °C outdoor temperature increase; electric power input increases by 1.3 % per °C increase[32].

Evolutions of capacity, load and load ratio, and efficiency with outdoor temperature are shown in Figure 18.

Maximum outdoor air temperature until which the unit can maintain the indoor temperature set point is 31.5 °C and thus at that point the maximum temperature difference between outdoor and indoor is 4.5 °C. Maximum capacity delivered at 35 °C is close to 1.8 kW, EER at 35 °C close to 1.6, and indoor temperature increases up to 30.7 °C at 35 °C. SEERon is 2.28 (versus 2.12 for single duct used as single duct).

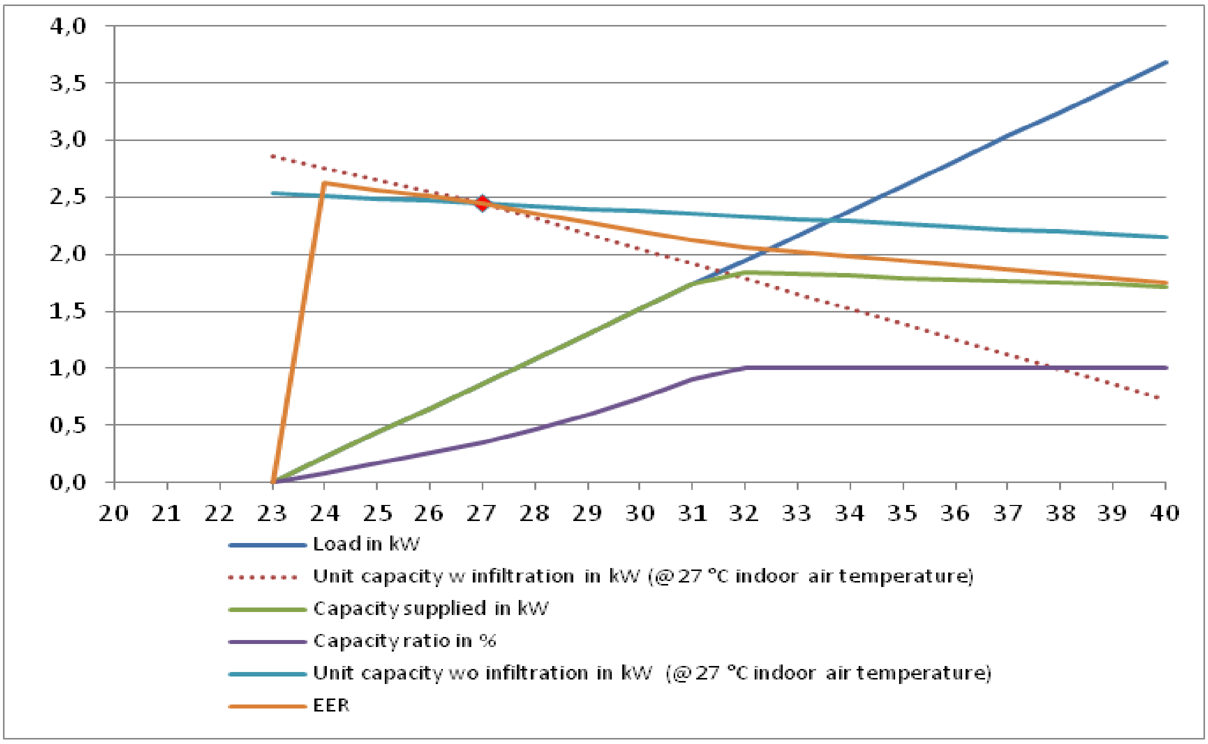

Figure 18: Evolution of load, capacity supplied, capacity with and without infiltration, capacity ratio and EER for single duct used declared as single duct and used as double duct

<u>Conclusion</u>

Table 11 below summarizes the results of the different scenarios envisaged.

[32] Values for split air conditioners in Preparatory study, Task 4.

Table 11: Summary of seasonal performance calculations, case A to C. Note 1: equilibrium is the intersection between load and unit capacity corrected for infiltration at 27 °C indoor temperature.

	Case A		**Case B**	**Case C**
	Base case SD		**Base case DD**	**Base case SD**
				Decl. as SD and used as DD
Rated values	Pc (kW)	2.6	2.6	2.6
	EER	2.65	2.65	2.65
	Ind. Out. Temp. °C	35 / 35	35 / 27	35 / 35
Design Load @ 35 °C	kW	2.6	2.6	2.6
Max capacity with infiltration	kW	1.5	2.6	1.8
Out. Temp. at equlibrium	°C	29.7	35.0	31.5
Temp. Diff. Ind. Out. at equlibrium	k	2.7	8.0	4,.5
SEERon	-	2.13	3.60	2.28
Cooling supplied (mult. 10/24)	kWh/a	365	405	384
Electricity consumption (mult. 10/24)	kWh/a	172	113	168

Despite that they have the same MEPS and labelling requirements, performances of single and double duct with present rating conditions are not comparable. Rated capacity and EER of double duct include infiltration and are measured at more classical 35 °C / 27 °C conditions. Thus, to get the same ratings, a double duct unit need in fact to be 90 % bigger and 70 % more efficient at comparable operating conditions. So, ratings should be reviewed to make rated performances comparable for all portable type products.

Capacity that can be expected from single duct decreases with the increase of difference between the outdoor air temperature and the indoor air temperature; from about 94 % of rated capacity for 0 K difference to about 0 kW at 34 °C (for the base case simulated). Typical conditions of operation were simulated: the maximum capacity of 1.5 kW is reached at about 30 °C outdoor for a 2.7 K outdoor / indoor temperature difference. Above 30 °C, the set point cannot be maintained and only a very limited outdoor / indoor temperature difference can be ensured. This is the main difference with a standard air conditioner which, if sized properly, should enable to maintain 8 K temperature difference between indoor and outdoor temperatures at 35 °C, and so to maintain the indoor temperature at the required set point. So, this type of unit is not working as it is generally expected from a standard air conditioner, but this is not known from the end-user. Capacity bought is not the one available above about 30 °C and the unit operates like an undersized fix type air conditioner. Capacity of single duct products is not comparable to the one of fix air conditioners.

In addition, based on a standardised load profile, it appears that seasonal performance (SEERon) of portable air conditioners is significantly influenced by infiltration and part load behavior.

It is then necessary both to revise rating conditions, to inform that the unit is not able to maintain the required set point above the equilibrium temperature, and to adopt a seasonal metrics.

Update of rating conditions

It appears that despite the points mentioned above regarding the lack of capacity of single duct and their limitation as regards outdoor / indoor temperature difference, there is a solid market for these products. Regarding portable double duct air conditioners, their market share is presently very low (can hardly be found in stores in Europe). This is probably at least partly due to the unfavorable rating conditions. Clearly, revised ratings should use the same temperature conditions including infiltration, if occurring for those conditions, to make both product types comparable.

In the USA, the US DOE kept a weighted average capacity (SACC) including infiltration corresponding to the following conditions 29.7 °C outdoor temperature for 27 °C indoor. This tends to decrease significantly the capacity and EER of single duct air conditioners (close to 40 % loss; for the hypothesis kept on infiltration, capacity decreases from 2.6 to about 1.6 kW) and to increase the capacity and EER of double duct air conditioner (from 1.5 kW to about 2.2 kW with infiltration).

Decreasing the capacity rating of single duct air conditioners may lead to increase the size of installed products, with limited comfort gains and with increased consumption (for case A before, oversizing by a factor 2 leads to 32 % increase in energy consumption, with only 5 % more cooling energy delivered with equilibrium occurring slightly above 31 °C instead of 30 °C; increase in consumption is thus mainly due to seasonal performance decrease because of cycling).

On the other hand, decreasing the capacity of single duct by including infiltration may make the comparison with fix air conditioners more realistic. It has been seen before that for a typical profile of use, a single duct air conditioner would not deliver more than 1.5 kW (against 2.6 kW rated). However, this maximum capacity delivered depends on the load curve (and so on the design condition) and on the capacity corrected by infiltration. And so, the maximum capacity delivered, and coincident outdoor temperature vary for each product and installation. Using this intersection between capacity and standardised load would lead to have a variable temperature for a given capacity (comparison of 1.5 kW at 30 °C or 1.9 kW at 32 °C is not straightforward).

On the other hand, the risk of oversizing is limited because capacity is not thought to be the main driver to choose a portable unit. Instead it is more likely that the end-users choose the size of the unit based on indications supplied by manufacturers regarding the area for which the product is adapted. This information may vary depending on the climate of the country where the product is sold.

This also means that it is not really an issue if capacity corresponds to a variable outdoor temperature depending on the unit.

So, capacity at equilibrium temperature should be kept as the unit capacity to be published on the product information. It gives the best possible estimate of the real service supplied by a specific portable unit. In addition, in order to show the difference with fixed appliances, it is necessary to add the following note to warn end-users:

above an outdoor temperature of XX °C (the "equilibrium" temperature), the equipment may not allow to maintain the indoor temperature at YY °C lower than the outdoor air temperature. Both XX and YY should be computed based on ratings. The formula are given below in the seasonal performance index part.

This applies to single duct and to double duct portable air conditioners.

Seasonal performance index

Using Regulation (EU) 206/2012 and taking a cooling season corresponding to hours above 23 °C, this leads to 1289 hours. As for fix installations, they are supposed to be used only 10 hours over 24, or about 58 % of the time, which leads to 750 hours in standby mode and leaves 549 hours for hours when the unit is on.

Using these figures and the standard SEERon simulation here before (2.6 kW single duct unit with EER 2.65, air flow at condenser of 520 m^3/h), this leads to the following figures:

- cooling energy supplied: 365 kWh/a

- electricity consumption (cooling mode only): 172 kWh/a

It is clear that integrating standby power consumption makes no difference; with 1 W power, standby power consumption is 0.75 kWh or 0.4 % or yearly electricity consumption for cooling. So further progress on standby power consumption cannot be expected from including standby in seasonal performance metrics.

In the Burke study, the fan mode includes both thermostat-off hours (hours with unit on and without cooling need in Regulation (EU) 206/2012) and fractions of hours when the unit is cycling. For these later hours, it is proposed to account for cycling by using a Cd factor and so these hours are not accounted for here. So the Burke study cannot be used to define thermostat-off hours. To do so the hours in bin 23 °C (no load) are considered; this corresponds to 218 hours, to be multiplied by 10/24 to only keep the hours when the unit is on or 91 hours. If thermostat-off power is slightly above 100 W, as measured for units of similar size as the one of the base case, this is still significant: 9 kWh or 4.5 %.

So, the following equations should be used to compute SEER, and maximum temperature operating conditions for rating portable products.

$$SEER = \frac{\sum_{j=1}^{n} hj \times Pc\_corr(Tj)}{\frac{\sum_{j=1}^{n} hj \times Pc\_corr(Tj)}{SEER_{on}} + H_{TO} \times P_{TO} + H_{SB} \times P_{SB}}$$

$$SEER_{on} = \frac{\sum_{j=1}^{n} hj \times Pc\_corr(Tj)}{\sum_{j=1}^{n} hj \times \left(\frac{Pc\_corr(Tj)}{EER_{bin}(Tj)}\right)}$$

Where:

- *Tj = the bin temperature*
- *j = the bin number*
- *n = the amount of bins*
- *Pc_corr(Tj) = below equilibrium point: the cooling demand of the building for the corresponding temperature Tj; above the equilibrium point: the capacity of the unit for the corresponding temperature Tj*
- *hj = the number of bin hours occurring at the corresponding temperature Tj*
- *$EER_{bin}(Tj)$ = the EER values of the unit for the corresponding temperature Tj.*
- *H_{TO}, H_{SB}: the number of hours the unit is considered to work in thermostat-off mode*
- *P_{TO}, P_{SB}: the electricity consumption during thermostat-off mode*

Calculation of *Pc_corr(Tj)* and *$EER_{bin}(Tj)$* for single duct air conditioners

The capacity Qc(Tj) for temperature of bin j should be computed as follows.

$$Qc(Tj) = Qc(27) \quad (Eq\ 1)$$

Where:

- *Qc(27): rated capacity at 27(19) indoor and outdoor*

Capacity should then be corrected for infiltration as follows:

$$Qc_{corr(Tj)} = Qc(Tj) + Q_{INF}(Tj) \quad (Eq\ 2)$$

Where:

- Qc_corr(Tj): maximum capacity of the unit corrected with infiltration
- Qc(Tj): maximum capacity in bin Tj without accounting for infiltration

The infiltration impact is calculated with the following formulas:

$$If\ Tj < 27, Q_{INF}(T_j) = \frac{27 - T_j}{27 - 20} \times \left[AF \times \left(rho_{air_{27}} \times h_{27} - rho_{air_{20}} \times h_{20}\right)\right]$$

$$If\ Tj > 27, Q_{INF}(T_j) = \frac{27 - T_j}{35 - 27} \times \left[AF \times \left(rho_{air_{35}} \times h_{35} - rho_{air_{27}} \times h_{27}\right)\right] = \frac{27 - T_j}{35 - 27} \times INF$$

Where:

- $Q_{INF}(T_j)$ *: Heat loss by infiltration (W)*
- *Tj: outdoor temperature of bin j*
- *AF: infiltration air flow (m³/s)*
- $rho_{air_{20}}$*= 1.20 kg / m3, density of dry air at 20 °C (1 atm)*
- $rho_{air_{27}}$*= 1.17 kg / m3, density of dry air at 27 °C (1 atm)*
- $rho_{air_{35}}$*= 1.15 kg / m3, density of dry air at 35 °C (1 atm)*
- h_{20} *= 42.2 kJ/kg*$_{da}$ *specific enthalpy of infiltration air at 20 °C dry bulb and 15 °C wet bulb temperature per kg of dry air*
- h_{27} *= 54.2 kJ/kg*$_{da}$ *specific enthalpy of infiltration air at 27 °C dry bulb and 19 °C wet bulb temperature per kg of dry air*
- h_{35} *= 72.5 kJ/kg*$_{da}$ *specific enthalpy of infiltration air at 35 °C dry bulb and 24 °C wet bulb temperature per kg of dry air*
- *INF = infiltration in kW (cooling capacity loss - negative capacity value due to infiltration)*

Equilibrium temperature, which is the intersection between building load curve (Eq 3) and capacity corrected with infiltration (Eq 2) is determined and noted Teq.

$$Qc\_corr(Tj) = Qc(27) + \frac{27 - T_j}{35 - 27} \times INF\ (Eq\ 2)$$

$$BL(Tj) = Qc(27) \times (T_j - 23) / (35 - 23)\ (Eq\ 3)$$

$$T_{eq} = \frac{Qc(27) + \frac{27}{35-27} \times INF + \frac{23}{(35-23)} \times Qc(27)}{\frac{Qc(27)}{35-23} + \frac{INF}{(35-27)}} \quad (Eq\,4)$$

Pc_corr(Tj) is then computed as follows:

$$If\ Tj\ \leq\ Teq: Pc\_corr(Tj) = BL(Tj)$$
$$If\ Tj\ >\ Teq: Pc\_corr(Tj) = Qc\_corr(Tj)$$

The unit capacity information to be published is Pc_corr(Teq).

To compute EERbin(Tj), two cases may occur. In both cases, the capacity ratio should be computed as follows:

$$CR(Tj)\ =\ min\,(1\,;\ Pc\_corr(Tj)\,/\,BL(Tj))$$

Case 1: on-off unit

$$If\ CR(Tj)\ <\ 1;\ EERbin(Tj)\ =\ EERrated\ x\ (1\ -\ Cdc\ x\ (1\ -\ CR(Tj)))$$

With Cdc cycling coefficient with a value 0.25 by default.

Case 2: inverter unit

A supplementary test should be made at 27 (19) / 27 (19) temperature conditions and at 33 % capacity ratio. The part load coefficient of EER variation noted PLc should be computed as follows:

$$PLc = \frac{\frac{EER(27;33\%) - EER(27;100\%)}{EER(27;100\%)}}{\frac{Qc(27;100\%) - Qc(27;33\%)}{Qc(27;100\%)}} \quad \text{(Eq 5)}$$

And EERbin(Tj) should be computed as follows:

$$If\ CR(Tj) \geq\ 0.33;\ EERbin(Tj)\ =\ EERrated\ x\ (1\ +\ PLc\ x\ (1 - CR(Tj)))$$

$$If\ CR(Tj)\ <\ 0.33;\ EERbin(Tj)\ =\ EERrated\ x\left(1\ +\ PLc\ x\ (1 - 0.33)\right)x\left(1 - 0.25\ x\ (1 - CR(Tj)/0.33)\right)$$

<u>Calculation of *Pc_corr(Tj)* and *EER$_{bin}$(Tj)* for double duct air conditioners</u>

$$Qc(Tj) = Qc(27)\ +\ (Qc(35) - Qc(27))/8\ x\ (Tj - 27)\ (Eq\ 1bis)$$

Where:

- *Qc(27): rated capacity at 27(19) indoor and outdoor*
- *Qc(35): rated capacity at 27(19) indoor and 35(24) outdoor*

Capacity should then be corrected for infiltration as follows:

$$Qc\_corr(Tj) = Qc(Tj)\ (Eq\ 2bis)$$

Equilibrium temperature, which is the intersection between building load curve (Eq 3) and capacity corrected with infiltration (Eq 2) is determined and noted Teq.

$$Qc\_corr(Tj) = Qc(27) + (Qc(35) - Qc(27))/8\ x\ (Tj - 27)\ (Eq\ 2bis)$$

$$BL(Tj) = Qc(27) \times (T_j - 23)\ /\ (35 - 23)\ (Eq\ 3)$$

$$T_{eq} = \frac{Qc(27) - 27 \times \frac{Qc(35) - Qc(27)}{8} + 23 \times \frac{Qc(27)}{(35-23)}}{\frac{Qc(27)}{35-23} - \frac{Qc(35) - Qc(27)}{(35-27)}}\ (Eq\ 4)$$

Pc_corr(Tj) is then computed as follows:

$$If\ Tj \le Teq: Pc\_corr(Tj) = BL(Tj)$$

$$If\ Tj > Teq: Pc\_corr(Tj) = Qc\_corr(Tj)$$

To compute EERbin(Tj), two cases may occur. In both cases, the capacity ratio (CR(Tj)) should be computed as follows:

$$CR(Tj) = min\ (1\ ;\ Pc\_corr(Tj)\ /\ BL(Tj))$$

For double duct units, the efficiency at maximum capacity should be computed as follows:

$$EER(Tj) = EER(27) + (EER(35) - EER(27))/8\ x\ (Tj - 27)$$

Where:

- *EER(27): rated EER at 27(19) indoor and outdoor*
- *EER(35): rated EER at 27(19) indoor and 35(24) outdoor*

Case 1: on-off unit

$$If\ CR(Tj) < 1;\ EERbin(Tj) = EER(Tj)\ x\ (1 - Cdc\ x\ (1 - CR(Tj)))$$

With Cdc cycling coefficient with a value 0.25 by default.

Case 2: inverter unit

A supplementary test should be made at 27 (19) / 27 (19) temperature conditions and at 33 % capacity ratio. The part load coefficient of EER variation noted PLc should be computed as follows:

$$PLc = \frac{\frac{EER(27;33\%) - EER(27;100\%)}{EER(27;100\%)}}{\frac{Qc(27;100\%) - Qc(27;33\%)}{Qc((27;100\%)}}\ (Eq\ 5)$$

And EERbin(Tj) should be computed as follows:

$$If\ CR(Tj) \ge 0.33;\ EERbin(Tj) = EERrated\ x\ (1 + PLc\ x\ (1 - CR(Tj)))$$

$$If\ CR(Tj) < 0.33;\ EERbin(Tj) = EERrated\ x\ \left(1 + PLc\ x\ (1 - 0.33)\right) x \left(1 - 0.25\ x\ (1 - CR(Tj)/0.33)\right)$$

Calculation example for a standard single duct product

For an on-off unit of 2.44 kW capacity at @ 27/27 and an EER of 3 @ 27/27, the infiltration air flow is set to 520 m^3/h, the SEERon value is 2.12. and the SEER is 2.08 with P_{TO}=25 W and P_{SB}=1 W.

The capacity to be published as product information is

Above an outdoor air temperature of 30 °C (rounded from 29.89 °C), the equipment does not allow to maintain the indoor temperature 3 °C lower than the outdoor air temperature.

Calculation details are given in Table 12 below. Note that the total cooling energy and electricity consumption is to be multiplied by 10/24 which corresponds to hours during which the unit is on. This gives 343 kWh of cooling energy supplied and 162 kWh of electricity consumed.

Table 12: SEERon calculaiton details for a standard single duct unit

bin-n°	OAT °C	hours	BLc %	BLc kW	Qc(Tj)	INF(Tj) kW	Qc_corr (Tj)	IAT(Tj) °C	Pc_corr(Tj) kW	CR(Tj) %	EER(Tj)	Cool. Ener. kWh	Elec. Cons. kWh
	20				2.4	1.8	4.3	27.0					
	21				2.4	1.6	4.0	27.0					
	22				2.4	1.3	3.8	27.0					
	23	218	0%	0.0	2.4	1.1	3.5	27.0	0.0	0%	0.0	0.0	0.0
1.0	24	197	8%	0.2	2.4	0.8	3.2	27.0	0.2	6%	3.0	40.1	13.2
2.0	25	178	17%	0.4	2.4	0.5	3.0	27.0	0.4	14%	2.9	72.4	25.3
3.0	26	158	25%	0.6	2.4	0.3	2.7	27.0	0.6	23%	2.7	96.4	36.0
4.0	27	137	33%	0.8	2.4	0.0	2.4	27.0	0.8	33%	2.5	111.4	44.6
5.0	28	109	42%	1.0	2.4	-0.4	2.1	27.0	1.0	49%	2.2	110.8	49.7
6.0	29	88	50%	1.2	2.4	-0.7	1.7	27.0	1.2	71%	2.0	107.4	54.7
7.0	30	63	58%	1.4	2.4	-1.1	1.4	27.1	1.4	100%	1.7	88.2	51.2
8.0	31	39	67%	1.6	2.4	-1.4	1.0	28.1	1.4	100%	1.7	54.6	31.7
9.0	32	31	75%	1.8	2.4	-1.8	0.6	29.1	1.4	100%	1.7	43.4	25.2
10.0	33	24	83%	2.0	2.4	-2.2	0.3	30.1	1.4	100%	1.7	33.6	19.5
11.0	34	17	92%	2.2	2.4	-2.5	-0.1	31.1	1.4	100%	1.7	23.8	13.8
12.0	35	13	100%	2.4	2.4	-2.9	-0.4	32.1	1.4	100%	1.7	18.2	10.6
13.0	36	9	108%	2.6	2.4	-3.2	-0.8	33.1	1.4	100%	1.7	12.6	7.3
14.0	37	4	117%	2.8	2.4	-3.6	-1.2	34.1	1.4	100%	1.7	5.6	3.3
15.0	38	3	125%	3.1	2.4	-4.0	-1.5	35.1	1.4	100%	1.7	4.2	2.4
16.0	39	1	133%	3.3	2.4	-4.3	-1.9	36.1	1.4	100%	1.7	1.4	0.8
17.0	40	0	142%	3.5	2.4	-4.7	-2.2	37.1	1.4	100%	1.7	0.0	0.0
											Totals	824.2	389.3
											SEER on	2.12	

Required changes in test standards

Single duct air conditioner seasonal performance calculation only requires one test at 27/27 plus a test to measure condenser air flow rate. Additional measurements required are for thermostat-off and standby (standby is already done). It is proposed that the capacity of single duct to be used is the capacity at 27/27 as it is then comparable with the one of double duct air conditioners.

For double duct, it is necessary to have two tests at 27/27 and 35/27 but there is no need to measure air flow rate. Additional measurements required are for thermostat-off and standby (standby is already done). It is proposed that the capacity of double duct to be used is the capacity at 27/27 as it is then comparable with the one of single duct air conditioners.

3.1.7 Real life versus standard performances

In the following section the real-life performance is discussed for both fixed and portable air conditioners.

For large efficiency increase of split air conditioners due to large indoor air flow increase, several comments have been received regarding the risk of designing units that operate very efficiently under EN14825 conditions but not in real life:

- In heating mode, temperature should not be blown below 32 °C (temperature of the skin) and probably closer to 40 °C to avoid "cold draft" effect. In case the temperature blown is too low, the end-user is likely to increase the indoor set point, thus increasing heating load and decreasing the real life SCOP and thus increasing the difference between standardized performance and real life one.
- In cooling mode, cooling without dehumidification does not allow to enter the comfort zone. This is likely to lead end-user to decrease the cooling temperature set point to feel more comfortable, with increased cooling load and decreased efficiency and thus increasing the difference between standardized performance and real life one.

Hence, building very efficient machines by increasing the indoor air flow may lead to disconnect the SEER and SCOP ratings from real life performances.

These comments were received for very large increase in indoor air flow rates over the base cases in Task 6. However, some of these air flow levels match the ones of existing products on the EU market (for products of the same capacity). It means there is a risk that some of the most efficient units on the EU market already have air flow levels which imply the savings expected by the end-user are not realized in practice.

It is thus proposed, for units in the scope of this study, to complete the data supplied in test report of EN14511-3 with a measurement of the blown air temperature in heating mode and blown air temperature and humidity in cooling mode. These temperatures are normally measured during tests but not disclosed/published. They should now be available in the technical documentation of the products.

These values are to be reviewed at the time of the next revision of Regulation 206/2012 and 626/2011. The review should evaluate the possible impacts and the required changes in the standard and the regulation to ensure that SEER and SCOPs remain representative of real life. Possible measures could include minimum air temperature supply in heating mode, minimum sensible to heat ratio for certain conditions in cooling mode or lowering the cooling temperature set point from 27 °C to lower values, particularly at low loads.

3.1.7.1 In situ continuous performance measurement

As explained in part 3.1.1, there is presently no agreed method to measure in-situ performance of air-to-air heat pump performance on sites, which makes it impossible to compare in-situ and standard performances as is currently done for other energy consuming products.

The reason why is that measuring capacity on the air side is cumbersome in real life. In a lab, indoor air flow and temperature inlet and outlet (and humidity) can be measured to evaluate cooling capacities at indoor unit side (although not precise enough). Stated in the standard, indirect measurement is done as follows: an electric heater heats the room and the air conditioner must stabilize the room temperature and the cooling capacity matches the heater energy and room losses, this is used for all air conditioners in scope. It is a more precise method but not possible to be done on field. As the above methods cannot be used on field, no measured capacity on field means real life efficiency cannot be measured.

However, on-board performance measurement on the refrigerant side is possible and highly precise (with the same or even better precision than standard measurement on the air side as performed today). At least one compressor manufacturer already developed such a method basing upon the compressor performance curves (sold as an option to system assembler with the compressor) and also one air conditioner manufacturer[33]. Several other methods are possible (using the expansion valve or the compressor energy balance)[34], and at least one of these methods is being developed[35].

These methods could be generalized and used by all manufacturers, but an important standardization work has to be done to ensure their reliability. This should include methods to correct performance evaluation for dynamic conditions and the way faults are filtered, as well as possible checks of on-board measurement capabilities by third parties.

This seems an important topic to improve further the efficiency and to reduce real life consumption of air conditioners, as well as to ensure that test standards are aligned with real life operating conditions as regards sizing, equivalent full load hours and other operating hours. As these methods could apply in practice to most types of vapor compression cycles, this would also help the unfortunate inspectors of HVAC systems (EPB Directive 2010/31/EU, articles 14 and 15), that presently have little information on system performance or adequate sizing.

3.1.7.2 Additional information to end-user to reduce real life consumption

UBA[36] suggests other ways to cut real life energy consumption to be studied and possibly become Ecodesign requirements:

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[33] Stefan Vandaele, Hiroshi Aihara, Optimisation of VRF systems in buildings by monitoring, in Heiselberg, P. K. (Ed.) (2016). CLIMA 2016 - proceedings of the 12th REHVA World Congress: volume 9. Aalborg: Aalborg University, Department of Civil Engineering.

[34] Kim, Woohyun and Braun, James E., "Evaluation of Virtual Refrigerant Mass Flow Sensors" (2012). International Refrigeration and Air Conditioning Conference. Paper 1245. http://docs.lib.purdue.edu/iracc/1245

[35] C.T. Tran, P. Rivière, D. Marchio, C. Arzano-Daurelle, In situ measurement methods of air to air heat pump performance, In International Journal of Refrigeration, Volume 36, Issue 5, 2013, Pages 1442-1455, ISSN 0140-7007, https://doi.org/10.1016/j.ijrefrig.2013.03.021.

[36] Jens Schuberth (UBA) et al., Minimum requirements on monitoring equipment for assessing the energy efficiency of heating, ventilation and air conditioning products, Submission by the Federal Environment Agency (UBA) and the Federal Institute for Materials Research and Testing (BAM) on the Ecodesign Process, 14th March 2018.

- Electricity consumption measurement and display: in general, literature indicates small but non-negligible gains; this is already a common feature of many fix air conditioners although electricity consumption may not be measured but deducted from the measured variables of the unit;
- Energy efficiency monitoring by energy signature methods: use the change in correlations between electricity consumption and outdoor air temperature to detect change in consumption pattern and inform the end-user; this requires however large computing / data storage capacity which may not be available in the units and could only be available to smart appliances; plus this method may not work for all installations;
- mandatory saving mode allowing to control a fixed temperature difference between indoor and outdoor temperature, rather than a fixed indoor temperature set point;

It is proposed to include the electricity consumption measurement in the standardisation work to be done regarding in-situ continuous performance measurement. It is indeed necessary to fix a certain number of requirements (e.g. maximum uncertainty, minimum acquisition and averaging times, test of the functionality in the unit based on standard tests).

3.1.7.3 Compensation method

In the present standard test procedure for split air conditioners in EN14825 standard, the test points are obtained by fixing the control parameters of the unit; in a compensation method, the load is fixed (e.g. heat is added in a closed room) and the controls of the air conditioner sense the increase in temperature and adjust the air conditioner capacity to match the load; the intent is to include the impact of air conditioner controls in terms of seasonal performances. Compensation method investigations were already mentioned in Task 1. To note, a round robin test should begin in 2017/2018 to test a new compensation test method proposal for air-to-air units in different laboratories in Europe.

3.1.7.4 Impact of a lack of maintenance

To reach the 500 hours equivalent full load hours in the first EU label for air conditioners[37], a 20 % provision for fouling of the heat exchangers was accounted for following Breuker & al[38] findings. More recently in the IEE project Harmonac[39], it was found that cleaning the surface of indoor heat exchanger not equipped of filter could save 17 % energy consumption.

In the Lot 10 preparatory study, it was supposed that most units were fitted with filters and that for split units a regular maintenance check was the standard and thus the impact of heat exchanger fouling was not taken into account. This is still thought to be the case for fixed installations. If it was not the case, a measure as the one proposed by UBA - to equip units with a visual heat exchanger cleaning signal as in Regulation EU no 2014/1253 - could prove useful and would most likely be cost-effective.

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[37] Commission Directive 2002/31/EC of 22 March 2002 implementing Council Directive 92/75/EEC with regard to energy labelling of household air-conditioners

[38] Value taken from the following reference, M. Breuker&J. E.Braun, Common faults and their impacts for rooftop air conditioners, July 1998, HVAC&R Research, p.303-318

[39] Ian Knight and James Cambray, Quantifying the Energy Conservation Opportunities found in Air-Conditioning Inspections as required by EPBD Article 9, IEECB conference, 2010.

There have been many developments in filtering options for split air conditioners, as reported by (Schleicher et al, 2017)[40] and this includes automatic filtering cleaning.

However, stakeholders report that for the portable range, filters are not always fitted to both heat exchangers and that all are not washable, which should be required for portable air conditioners as most likely the end-user has to perform the maintenance himself. It could then be proposed a mandatory requirement for washable filters on both heat exchangers for portable air conditioners.

3.1.8 Use patterns of comfort fans

The hours of use of comfort fans will be based on the preparatory study. The overall assumption is that comfort fans will be used the same way as air conditioners, but without the possibility of more advanced control e.g. pre-cooling. When a cooling need appears, the users will properly use either a comfort fan or an air conditioner. The preparatory study assumed an average value for EU of 320 hours per year.

3.2 End-of-life behaviours

3.2.1 Product use and stock life

The lifetime of products is of great influence of the products overall environmental performance. If a product has a long lifetime it is beneficial regarding resource consumption but at some points there might be some trade-offs after some years in terms of lower efficiency compared to a new model. The product lifetime can be interpreted in a numerous of ways and there exist different definitions of the lifetime of products. In the below table are three definitions of lifetime presented.

Table 13: Different definitions of lifetime

The design lifetime	The behavioural (or social) lifetime	Definition in current study
Is the intended lifespan regarding functioning time, the number of functioning cycles, etc., foreseen by the manufacturer when he designs the product, provided that it is used and maintained by the user as intended by the manufacturer. The design lifetime must not be confused with the guarantee period of products, which is a service offered by the Manufacturer and fulfils other constraints, namely commercial.	Is defined as the number of years until the device is replaced for other reasons than technical failure or economic unattractiveness. This generally regards social and consumption trends, a product including new feature has been released and is preferred, e.g. a more powerful computer	The term "lifetime" or "Economic product life" used in the current study must be understood as the period (i.e. the number of years) during which the appliance is used and consumes electricity ("actual time to disposal"). Therefore, it is a value included between the social lifetime and the design lifetime.

[40] Tobias Schleicher, Jonathan Heubes, Ran Liu, Pascal Radermacher, Jens Gröger. The Blue Angel for Stationary Room Air Conditioners – a national eco-label with international impact, Final Draft of the technical Background Report for German Federal Environment Agency. Freiburg, June 2017.

An accurate lifetime can be difficult to determine as many factors can affect the lifetime such as location, marine climate and the hours of operating in different modes. Depending on the type of air conditioners other factors can also have an impact e.g. if portable air conditioners are moved frequently they a subject to bump into doors or furniture which in the end can affect the lifetime. For fixed air conditioners, the lifetime can be significantly reduced due to customer installation. The main causes for replacement of air conditioners are listed below[41]:

- breakdown
- drop in performance
- rise in sound noise
- replacement by more energy efficient models,
- building renovation
- the end-user is moving

The overall assumptions regarding lifetime are unchanged since the preparatory study[42] and the previous assumptions used are briefly described below.

ENERGY STAR has published a brochure[43] about room air conditioners where it is written that the typical product life for a room air conditioner is ten years[44]. Furthermore, because models that are at least 10-years old use 20% more energy than a new ENERGY STAR model, ENERGY STAR advocates consumers to replace 10-year-old room air conditioner with a new ENERGY STAR qualified model.

White certificates in France had estimated the lifetime of air conditioners (only single split package and multi split package air conditioners) that were used in French overseas departments and territories at 9 years. However, these conditions were expected to be more intensive since the climate in general are hot and air is charged in salt leading to the accelerated corrosion of the heat exchange areas of the outdoor unit. The lifetime of reversible air conditioners (only single split package and multi-split package air conditioners) used in France was estimated to 16 years.

JRAIA informed at a stakeholder meeting during the preparatory study that the product life for residential units in Japan was estimated to be about 12 years based on an analysis of recycled products and for commercial units it is thought to be between 10 and 15 years. They thought it could be longer in Europe than in Japan because of less hours of usage in average, especially since a large part of the products are not reversible or not used as such, whereas in Japan residential air conditioners are used in winter and summer time. Today are most fixed air conditioners sold in Europe reversible.

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[41] Preparatory study on the environmental performance of residential room conditioning appliances (airco and ventilation), Contract TREN/D1/40-2005/LOT10/S07.56606. CO-ORDINATOR: Philippe RIVIERE, ARMINES, France. Air conditioners, final report. December 2008. This report can be downloaded from DG ENERGY website: https://ec.europa.eu/energy/en/topics/energy-efficiency/energy-efficient-products.

[42] Preparatory study on the environmental performance of residential room conditioning appliances (airco and ventilation), Contract TREN/D1/40-2005/LOT10/S07.56606. CO-ORDINATOR: Philippe RIVIERE, ARMINES, France. Air conditioners, final report. December 2008. This report can be downloaded from DG ENERGY website: https://ec.europa.eu/energy/en/topics/energy-efficiency/energy-efficient-products.

[43] http://www.energystar.gov/ia/partners/manuf_res/downloads/Room_Air_Conditioners_Partner_Resource_Guide.pdf

[44] 29th Annual Portrait of the U.S. Appliance Industry, Appliance Magazine, September 2006.

EERAC study[45] suggested the average lifetime expectancy for four types of products. The products and expected lifetime was:

- Window/wall package air conditioner 12.6 years
- Multi split package air conditioner 12.6 years
- Single split package air conditioner 12.5 years
- Single-duct package air conditioner 10.3 years

It was also found that the average life span of the four main air conditioner types in Spain and Italy was largely due to the desire for a new unit rather than equipment failure, due to good reliability and low annual usage.

CECED informed at a stakeholder meeting during the preparatory study that the estimated product life of portable air conditioners was about 12 years. They also added that premature replacement sometimes might occur as switch from portable air conditioners to fixed split systems. This observation is in good agreement with the fact that moveable air conditioners, that have the compressor located inside the room whose air is to be cooled, are noisier for the end-user that other air conditioner types. Many other causes can justify this willingness to change as being forced to install the air conditioner when it is to be used, having the indoor unit inside the room if space is constrained.

Lifetime assumptions in current study are based on the above-mentioned sources from the preparatory study. Below is a summary of the expected lifetime of fixed air conditioners and portable air conditioners.

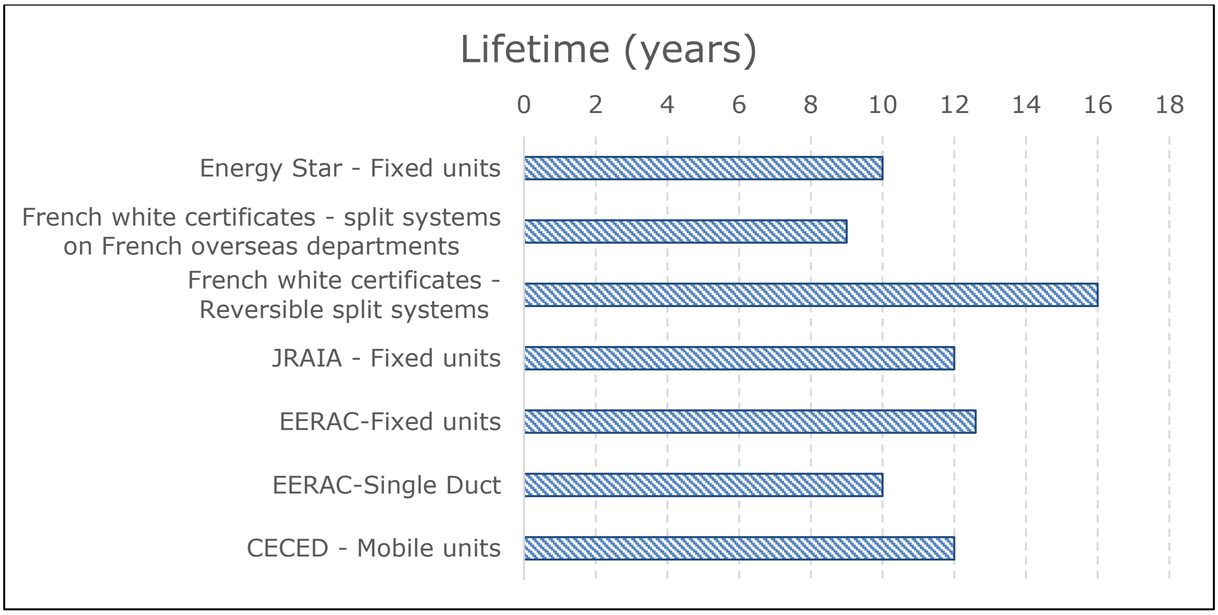

Figure 19 Lifetime for different types of air conditioners according to several sources

The expected lifetime for fixed air conditioners is in the range of 9-16 years and for portable air conditioners it is in the range of 10-12 years. A lifetime of 10 years for portable air conditioners was also used by the US DOE[46] recently. The expected lifetime for fixed and portable air conditioners in the current study are presented in Table 14. Regarding Variable Refrigerant Flow (VRF) units the expected lifetime is 15 years. These assumptions are

[45] ENEA, 1999, Survey on the behaviour of the end users of room air conditioners, Report within the frame of the EERAC study, ENEA, 1999.

[46] https://energy.gov/sites/prod/files/2016/12/f34/PAC_ECS_Final_Rule.pdf

based on the impact assessment[47] for air heating products, cooling products and high temperature process chillers above 12 kW. Regarding comfort fans the assumption on lifetime is kept in the current study. The lifetime was estimated to 10 years.

Table 14: Average lifetime assumptions adopted to the current study

Technology	Average lifetime (years)
Movables + Window Units	10
Small Split (<5 kW)	12
Large Split (>5 kW, incl. ducted)	12
VRF	15
Comfort fans	10

It is difficult to suggest any Ecodesign requirements regarding the lifetime of air conditioners and comfort fans as the expected lifetime currently are 10 years or above. If the lifetime is improved the trade-off between resource efficiency and energy efficiency should be considered, in addition, consumers may not be willing to repair their air conditioners due to high repair cost in comparison with the costs of purchasing a new equipment. This is discussed in the following section. Furthermore, requirements of lifetime would be difficult for the market surveillance authorities to control.

3.2.2 *Repair and maintenance practice*

The repair and maintenance practice are expected to be done by professionals for fixed air conditioner systems. The price of repair is then constituted of the labour costs and the cost of the spare parts, which means that the affordability of repair is very much dependent on the labour costs. The labour cost varies greatly across Europe and are presented in the below figure.

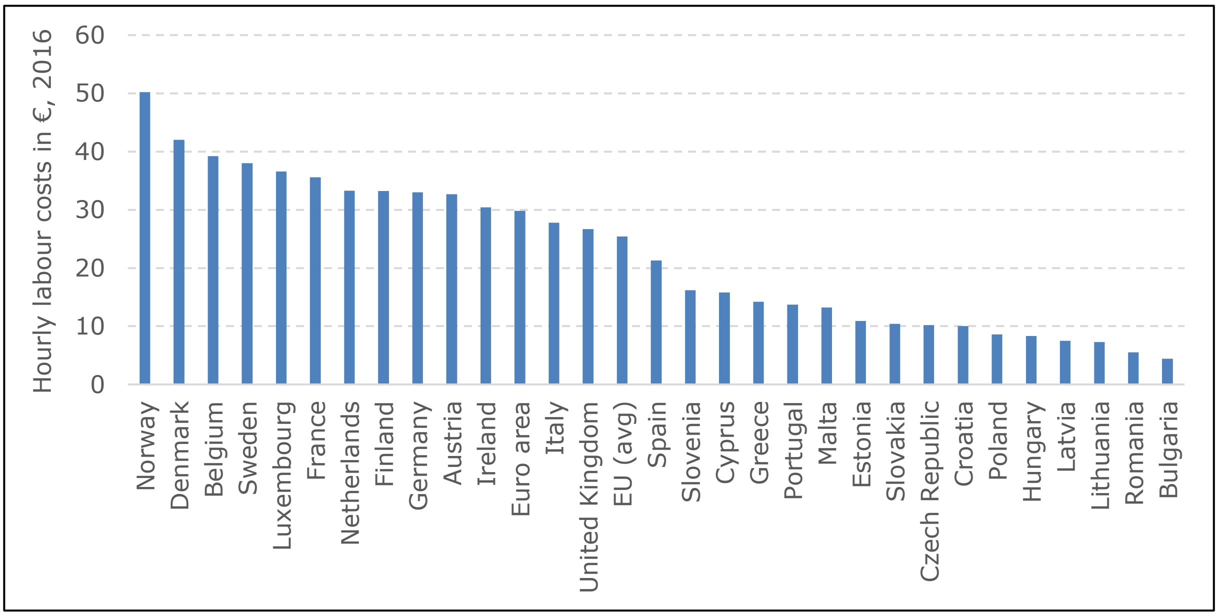

Figure 20 Hourly labour cost in €, 2016 for European countries

Based on labour cost the amount of repair is expected to be low in northern countries and higher in southern and south-eastern countries. Another important factor is also the age of the equipment. At end-of-life (above 9-10 years) air conditioners are probably too expensive to repair compared to the price of a new model. Furthermore, a new model is also expected to be more efficient so that the total cost of ownership is lower for the new

[47] http://ec.europa.eu/smart-regulation/impact/ia_carried_out/docs/ia_2016/swd_2016_0422_en.pdf

model compared to repairing the old one an extending the lifetime. This balance is dependent on the efficiency and price of a new model and the cost of repair. The amount of repair is assumed to be low currently but may change in the future since there is an increased focus on resource consumption within the EU. With an increased focus on resources and lower impacts of electricity (both renewable energy and more efficient products) the impact of the resource consumption may be more important. An improved lifetime of certain products could potentially save energy in the future as less materials are used. But, the electricity consumption is still believed to be the most important factor currently. Regarding comfort fans the repair and maintenance practice is assumed to be negligible. This is due to the low purchase price and simplicity of the product. The same assumption was made in the preparatory study.

Currently there are no observed problems with the repair service of air conditioners, but the high repair cost could pose a challenge even for relatively new products after the warranty expires. To avoid unnecessary replacement of efficient air conditioners it should be considered how Ecodesign requirements potentially could ease the repair and maintenance practice to lower the cost of repair and prevent premature failure. This subject will be discussed further in task 4. Though, it may be difficult for the market surveillance authorities to control.

Spare parts

Spare parts are available on the internet from some manufacturers[48,49] and third-party companies offering spare parts and sometimes also a repair service. In any case the most likely scenario is that the service is performed by a professional which would add extra expenses to the costs of the spare parts. Important technical problems like a compressor breakdown may occur when the appliance is old. Since the lifetime of a compressor is estimated around 12 years[50], it is very unlikely that a compressor change would occur during the lifetime of the air conditioner. If it happens after 10 or 12 years, it is most likely that the end-user will change the complete unit given the cost of the intervention. It is not known if repair actions are often carried out neither which ones and consequently, it is not possible to estimate additional material for repair. Stakeholders have indicated that the most bought spare parts are printed circuit boards. One stakeholder informed that the majority of their air conditioners was sold with a maintenance contract and their most sold spare part was the compressor. The most bought spare parts are properly affected whether the air conditioners are sold with a maintenance contract or not. The availability of the spare parts is assumed to be 10 years based on stakeholder inputs.

Normal maintenance like cleanliness of the filters, emptying and cleaning the condensate container requires no additional material. Manufacturers now propose washable filters. In general, for products in the scope, most of the air conditioners are assumed to be sold without a maintenance contract.

[48] http://www.daikin.com/products/ac/services/parts/index.html
[49] http://www.toshiba-aircon.co.uk/products/spare-parts
[50] Preparatory study on the environmental performance of residential room conditioning appliances (airco and ventilation), Contract TREN/D1/40-2005/LOT10/S07.56606. CO-ORDINATOR: Philippe RIVIERE, ARMINES, France. Air conditioners, final report. December 2008. This report can be downloaded from DG ENERGY website: https://ec.europa.eu/energy/en/topics/energy-efficiency/energy-efficient-products.

Based on the stakeholder consultation, the availability of spare parts seems reasonable, but it could be discussed with stakeholders how to maintain the availability of spare parts from all manufacturers.

Refrigerant fluid leakage rate

The refrigerant fluid leakage rate represents the amount of rejected fluid according to the initial charge and is an important factor both considering the efficiency of the unit and the release of CO_2-equivalents to the atmosphere. The release of CO_2-equivalents is both a consequence of using electricity for powering the air conditioner and by the leaking of refrigerants. The leakage of refrigerants is based on the findings in the preparatory study and stakeholder inputs which are briefly described below.

A French study[51] proposed different values for average yearly emission rates and presented in Table 15.

Table 15: Upper estimates of the annual leakage rate

	Single and double duct	Split	Multi split (<17.5kW)
Nominal charge [kg]	0.5	1	1.5
Lifetime [years]	10	12	12
Leak rate	2%	5%	15%

The values presented in the table above are an upper estimate of the annual leakage rate and will be used in a sensitivity analysis. Variable Refrigerant Flow(VRF) systems was not included in the preparatory study but values from the multi-split category will be adopted.

Portable units are in the preparatory study estimated to have an annual leakage rate of 1 % including the end of life losses. However, stakeholders have previously and currently indicated that many portable air conditioners are filled and hermetically sealed in factory which means that they have very low leakage. One stakeholder claimed that their annual leakage rate was as low as 0.2 %. To determine the correct leakage rate can be difficult since the end of life leakage also must be included. An annual leakage rate of 1 % will be used in the current study.

Split, multi-split systems and VRFs have a higher annual leakage rate than portable air conditioners. Split systems are in the preparatory study assumed to have an annual leakage rate of 3 %. A field study of leakage rates on air conditioning and refrigerating equipment was led in France[52] and estimates of 3.8 % were found in situ for two split units working with R22. The same order of magnitude was identified for small air to water heat pumps. According to this field study leaks are localized at connection of supplementary components (valves, pressure gauges, potential intrusive temperature measurements). Average leak rates of 1.68 % (without End-of-Life losses) was suggested at the "2013 Annual Conference of the Institute of Refrigeration – LEC Leakage & Energy Control

[51] Barrault, 2004, Stéphanie BARRAULT, Denis CLODIC, avec la participation de Carine SAYON, "Inventaire des fluides frigorigènes et de leurs émissions", 2004 Inventaire des fluides frigorigènes et de leurs émissions, France, 2004, Document 2 : Données de base pour les inventaires de fluides frigorigènes.

[52] CETIM, 2004, Centre d'Etudes Techniques des Industries Mécaniques, Résultats des actions collectives N° 695, Confinement des installations frigorifiques, 2004.

system". Including End-of-Life loses an annual refrigerant leakage rate of 3 % seems reasonable.

End-of-Life refrigerant leakage rates are difficult to estimate since the rate is depending on how the units are treated and which functions are available. Some products today are equipped with a "pump-down" function to recover refrigerant from indoor units and connecting pipes into the outdoor unit. These functions are facilitating recovery of refrigerants and are preventing refrigerants to be emitted to the atmosphere. Different scenarios are made in the sensitivity analysis to cover the importance of refrigerant leakage.

As many uncontrollable factors such as the installation of the equipment have an impact on the leakage rate, it is difficult to implement requirements in the Ecodesign regulation. The challenge may also be related the handling at End-of-Life. This leakage at End-of-life may be improved by pump-down systems and is further discussed in Task 4.

3.2.3 *Estimated second hand use*

The estimated second-hand market is based on the preparatory study which made some assumptions based of statement from UNICLIMA and a survey on Ebay carried out in preparatory study and supported by desk research over several online platforms during the current review study. The statement from UNICLIMA stresses that there were no incentive or economic justification for refurbishment or second-hand use of fixed air conditioners since the cost of reinstalling a fixed air conditioner and fill it with a new fluid costs more money than a new product. Regarding portable air conditioners, a second-hand market exists. The situation may change but there is at the moment very little second-hand use of the products. It was confirmed by a survey led on Ebay in the frame of the preparatory study where very few products were available, despite of direct imports from China of new products. Main second-hand market is thought to relate to people buying a portable air conditioner before the summer in case of heat wave and willing to recover the money once the summer has ended. The same assumption as for portable air conditioners can be made for comfort fans. A recent survey on Ebay and sites alike confirmed that a second-hand market also exist for comfort fans despite their low initial price.

If the air conditioner is barely used in its primary life, after being resold as a second-hand product, it could potentially increase its typical lifetime. However, the second-hand market exists for air conditioners is rather small, because of the high costs to uninstall/install and refurbish the equipment. Due to the small market in the EU, it is assumed that it has no or very limited impact on the average lifetime evaluation and can be neglected in the coming tasks. The impacts of a second-hand market could potentially negative impacts due to the trade-offs between material consumption and energy consumption. As reusing old and inefficient air conditioners means that although raw materials are saved but it is at the cost of higher electricity consumption.

3.2.4 Best practice in sustainable product use

Sustainable product use can minimise the energy consumption of air conditioners. A few best practices are listed in this section.

First it is important to purchase a properly sized air conditioner which is described in a previous section. Following the purchase of the air conditioners the end users can also affect the energy consumption of the equipment. The end user should consider the following points[53]:

- **Proper positioning** – The air conditioners should be placed centrally in the room and on the upper part of the wall since cold air tends to go down and mix with warm air. The air conditioners should be free of obstacles that might prevent the diffusing of cold air into the room. Furthermore, is it important that the thermostat is clear of any potential heat source such as televisions which can cause the air conditioner to operate longer than necessary
    - **Number of air conditioners** – Installing one powerful air conditioner in e.g. a corridor is not sufficient as the only really cooled room would be the corridor, not the rest of the rooms.
- **Amount of cooling** – it is often adequate to lower the temperature with two or three degrees compared to outdoor temperature and use dehumidification. It is often the humidity that makes end user perceive a higher temperature than it is.
- **Closed windows and doors** – preventing warm air into the room.
- **Insulate the cooling circuit tubes outdoor** – direct sunlight can deteriorate the tubes. It is also important to make sure that the external part of the air conditioner be not exposed to direct sunlight and bad weather conditions.
- **Use the timer and the 'Night' mode** – reducing the hours of use.
- **Cleaning and correct maintenance** – Regular cleaning of air filters and fans at the beginning of the cooling season (importance of cleaning filters is discussed in previous section). A proper maintenance can reduce the cost for cooling and heating by up to 10 % and prolonging the life of the equipment[54].

Many of the above-mentioned suggestions are also applicable for air conditioners used for heating e.g. to properly place the air conditioner so it can supply heat to a large area/room. The temperature control is also of great importance as the temperature can be lowered during the night and during working hours when the house is empty.

Regarding comfort fans the best sustainable product use is based on preparatory study. The preparatory study suggests using fans with variable speeds and curved blades. Besides this proper sizing is also important. Tre proper fan size dependent on the room area is presented in Table 16.Table 16: Ideal fan width depending on the room area

Room Area (m^2)	Ideal Fan Width
Up to 10	900 mm
10-20	1200 mm
15-30	1400 mm
+30	Two or more fans

3.3 Local infrastructure

3.3.1 Electricity

The power sector is in a transition state moving from fossil fuels to renewable energy. The origin of the electricity is very important factor to consider both regarding the environmental impact by using an air conditioner and how it may affect the consumer behaviour. Within the EU there are a number of renewable energy targets for 2020 set out in the EU's renewable energy directive[55]. The overall target within the EU is 20 % final energy consumption from renewable sources. To achieve this goal the different EU

[53] http://www.enea.it/en/news/environment-10-tips-to-efficiently-use-air-conditioning-and-keep-bills-low-cost
[54] https://energy.gov/sites/prod/files/guide_to_home_heating_cooling.pdf
[55] https://ec.europa.eu/energy/en/topics/renewable-energy

countries has committed to set their own individual goal ranging from 10 % in Malta to 49 % in Sweden. In 2015 the share of renewable energy was almost 17 %[56].

The electricity consumption are a major part of the final energy consumption and the electricity mix is highly relevant for air conditioners. The electricity mix of 2015 is presented in the below figure.

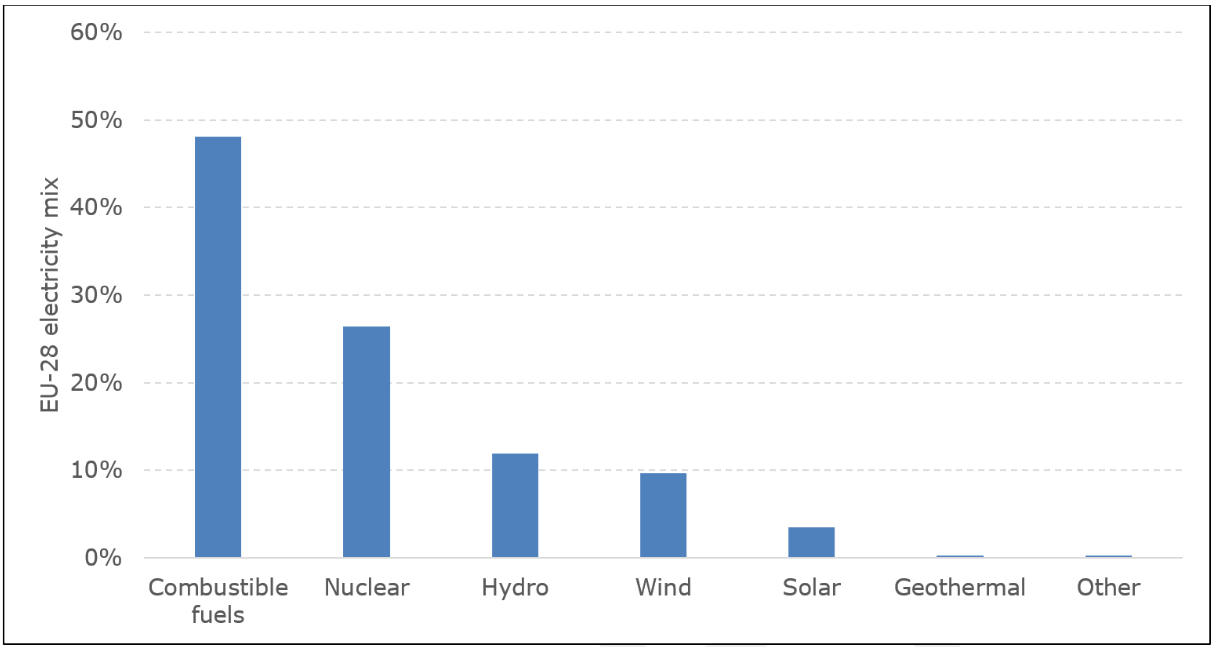

Figure 21: Net electricity generation, EU-28, 2015 (% of total, based on GWh)[57]

Almost half of the electricity consumption still originates from combustible fuels and renewable energy sources only constitutes about 25 % of the electricity generation in 2015.

The reliability of the electricity grid could be in some degree affected by the transition to a renewable energy system. With more renewable energy in the system new challenges occur e.g. with excess production of wind energy and the two-directional transfer of energy. Due to technological development, the reliability in many EU countries is ensured via the expansion of the electricity grid to distribute renewable energy. The quality of the electricity grid in Europe is considered to be high and among the best in the world. Every year the World Economic Forum release a Global Energy Architecture Performance Index report. The report is ranking the different countries on their ability to deliver secure, affordable, sustainable energy. In recent years European countries have dominated the top spots[58].

[56] http://ec.europa.eu/eurostat/documents/2995521/7905983/8-14032017-BP-EN.pdf/af8b4671-fb2a-477b-b7cf-d9a28cb8beea
[57] http://ec.europa.eu/eurostat/statistics-explained/index.php/File:Net_electricity_generation,_EU-28,_2015_(%25_of_total,_based_on_GWh)_YB17.png
[58] https://www.weforum.org/reports/global-energy-architecture-performance-index-report-2017

Country	2017 score	Economic growth and development	Environmental sustainability	Energy access and security
Switzerland	0.8	0.74	0.77	0.88
Norway	0.79	0.67	0.75	0.95
Sweden	0.78	0.63	0.8	0.9
Denmark	0.77	0.69	0.71	0.91
France	0.77	0.62	0.81	0.88
Austria	0.76	0.67	0.74	0.88
Spain	0.75	0.65	0.73	0.87
Colombia	0.75	0.73	0.68	0.83
New Zealand	0.75	0.59	0.75	0.9
Uruguay	0.74	0.69	0.71	0.82

The consumer behaviour might affect the electricity system in some countries since the use of air conditioners often occurs in the same time periods when it is either warm or cold. January is in general the month with the highest monthly electricity consumption across EU except for some of the southern countries like Spain, Italy and Greece which all peaks in July. In the below table are the monthly electricity consumption presented for most of the EU countries[59]. In the below table is the peak consumption marked with red and the lowest consumption marked with blue.

[59] Data provided by ENTSO-E

Table 17: Monthly electricity consumption

MONTHLY CONSUMPTION (IN GWh)													
Country	**Jan.**	**Feb.**	**Mar.**	**Apr.**	**May**	**Jun.**	**Jul.**	**Aug.**	**Sep.**	**Oct.**	**Nov.**	**Dec.**	**Total**
Austria	6498	5984	6203	5542	5468	5376	5588	5436	5271	5900	6005	6234	69505
Belgium	8057	7312	7653	6940	6795	6657	6548	6609	6731	7221	7202	7284	85009
Bulgaria	3455	3068	3111	2639	2404	2363	2611	2537	2416	2703	2766	3171	33244
Cyprus	368	364	338	283	314	343	452	495	441	351	298	358	4405
Czech Republic	6019	5584	5774	5200	4972	4818	4859	4641	4865	5509	5553	5624	63418
Germany	48952	45608	46179	40889	39607	39875	41470	39824	40911	45723	46280	45289	520607
Denmark	3188	2909	2916	2306	2648	2907	2556	2692	2697	1943	2555	3113	32430
Estonia	816	719	743	679	634	573	574	593	624	719	714	751	8139
Spain	23883	22048	22279	19837	21016	21614	24972	22341	20897	20964	20985	22069	262905
Finland	8437	7336	7645	6756	6268	5838	5941	6008	6118	7138	7279	7730	82494
France	52475	48579	45707	36847	33873	33225	34887	31582	33483	39167	40985	44593	475403
United Kingdom	32243	29083	31380	26097	26044	24327	24569	24361	25082	28320	30380	30768	332654
Greece	4829	4299	4504	3772	3823	3965	4855	4687	4086	3835	3895	4610	51160
Croatia	1538	1429	1461	1314	1292	1288	1573	1494	1336	1351	1369	1539	16984
Hungary	3629	3316	3507	3218	3209	3249	3484	3342	3313	3507	3490	3491	40755
Ireland	2498	2279	2397	2154	2192	2055	2100	2087	2120	2276	2353	2445	26956
Italy	26786	24948	26793	24169	25027	26328	31970	24458	26449	25907	25675	25818	314328
Lithuania	1005	891	920	873	862	825	846	863	866	955	958	995	10859
Luxembourg	574	538	579	516	497	503	542	512	492	554	547	514	6368
Latvia	692	616	635	589	571	522	549	568	562	625	626	654	7209
Netherlands	10343	9183	9588	8741	8881	8823	9191	9049	9149	9685	9763	10119	112515
Norway	13526	12065	12244	10410	9989	8880	8391	8585	8979	10632	11726	12872	13526
Poland	13546	12327	13116	12060	12011	11716	12333	12295	12099	13257	13066	13254	151080
Portugal	4713	4232	4167	3727	3939	3964	4280	3907	3883	3987	3977	4189	48965
Romania	5023	4598	4791	4435	4258	4202	4636	4398	4266	4665	4634	4877	54783
Sweden	14100	12610	12851	10967	10494	9602	8907	9561	9888	11578	12242	13130	135930
Slovenia	1233	1130	1178	1067	1092	1088	1149	1073	1099	1175	1164	1199	13647
Slovakia	2470	2277	2393	2194	2157	2115	2191	2136	2128	2360	2350	2405	27176

Spain, Italy and Greece are all among the countries with the highest share of air conditioners installed. This is properly one of the reasons that these countries all have their peak consumption of electricity in July. The lowest monthly electricity consumptions are for most countries within EU in June. The hourly load values for a random Wednesday in March 2015 for selected countries are presented in the figure below.

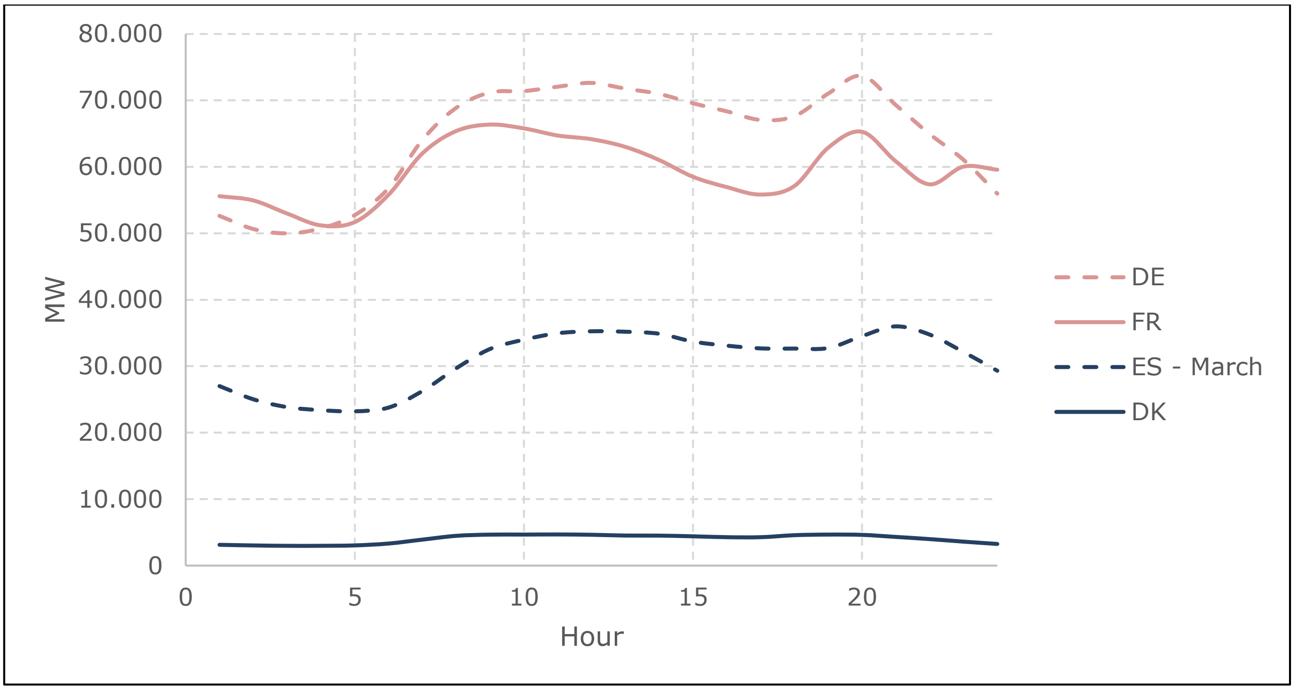

Figure 22: Hourly load values a random day in March

All presented countries have similar hourly load values with two peaks, one in the morning and one in the evening. It is barely visible for Denmark but this is due to scale of the graph. There are though small shifts in the peaks. In Denmark, the peaks occur a little earlier than in Spain. The first peak fits well with the start of the workday and the second peak fits with the end of the workday. Between the two peaks there is a falling trend in the energy consumption. The lowest electricity consumption across the different countries are at 5 AM. For most countries, this hourly load curve fits this description the majority of the days. For months and days with a higher or lower consumption tendency the profile are the same it is just shifted up or down. In the southern countries such as Spain, Italy and Greece the hourly load consumption is different in July. The hourly load consumption for Spain in March and July is presented in Figure 23.

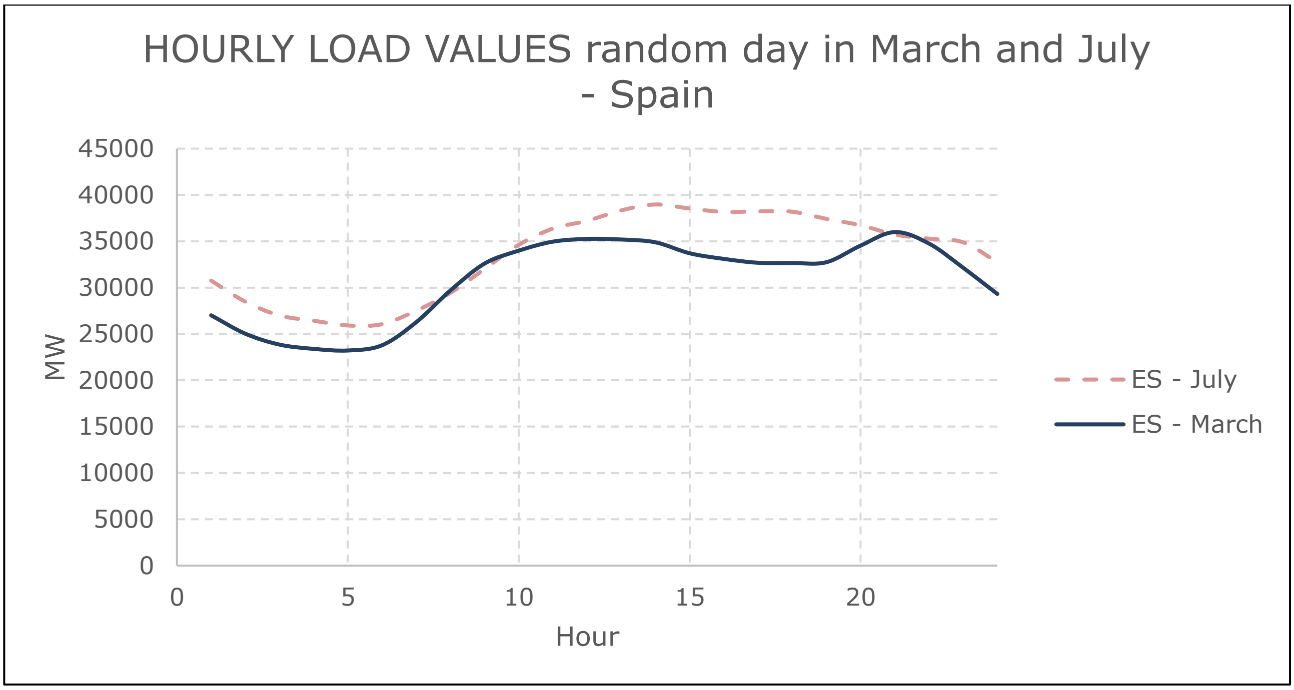

Figure 23:Hourly load values a random day in March and July - Spain

In Spain, the hourly load values only have one peak at 3 PM. The same pattern is also visible for e.g. Greece and Italy in July. This change in values is properly caused by the increased use of air conditioners. In this specific time period, it is also expected that the photovoltaic production in warm countries are peaking. The installed capacity of air conditioners is below 5 GW while the installed photovoltaic capacity is above 5[60] GW. This does not mean that air conditioners solely are operating on renewable energy but a comment on possible synergies between air conditioners and the photovoltaic technology.

3.3.2 Smart appliances

More smart appliances are expected to enter the market and air conditioners in this category. Smart appliances are defined in the ecodesign preparatory study on smart appliances (Lot 33)[61] as an appliance that supports Demand Side Flexibility (in a more recent report of the preparatory study, these called **Energy Smart Appliances**, to be more specific about the type of smart the appliance is capable of):

- It is an appliance that is able to automatically respond to external stimuli e.g. price information, direct control signals, and/or local measurements (mainly voltage and frequency);
- The response is a change of the appliance's electricity consumption pattern. These changes to the consumption pattern are what we call the 'flexibility' of the smart appliance;

Whereby:

- The specific technical smart capabilities do not need to be activated when the product is placed on the market; the activation can be done at a later point in time by the consumer or a service provider.

[60] https://www.worldenergy.org/data/resources/country/spain/solar/
[61] http://www.eco-smartappliances.eu/Documents/Ecodesign%20Preparatory%20Study%20on%20Smart%20Appliances%20_Tasks%201%20to%206.pdf

Overall air conditioners have the potential to have some flexibility in their use patterns and have a certain impact on the electricity system. Although shown in Task 2, there are already air conditioners with network capability available on the market that can be controlled via smart phones, tablets and other devices remotely via network, and grid communication capability that allows control of device according to electricity prices, these are not Energy Smart Appliances as defined in Lot 33 yet. Furthermore, these networked air conditioners on current market do not vary significantly from products without network capabilities.

Depending on the number of energy smart appliances in the system these appliances can provide energy system services both in day-ahead and in real-time by shifting operation. Day-ahead services leads to a reduced cost and CO2 emission compared to a situation without smart appliances, since additional generation by conventional power plants (e.g. coal) could be avoided due to a smart shift in load. The same benefits can be observed regarding real-time services[62]. With energy smart appliances it is possible to fit the demand to the production.

Some of the current barriers for energy smart appliances are estimated high prices (not always the case, but for many premium products), limited consumer demand, long device replacement cycles and the fragmentation within the connected home ecosystem. There are many different ways of connecting to the “smart home”. Without any common standards for the smart home it is difficult and confusing for the consumer to set up and control multiple devices. An additional reason is the fact that the electricity suppliers do not currently offer demand response.

The limited consumer demand for energy smart appliances can very well be related to the low availability of these products and the current electricity prices in the different countries. In most of the EU countries more than half of the electricity prices constitute of taxes and network cost, so if the electricity price is low there is no or only little incentive to use electricity in these periods since it is only a minor part of the costs (note that other ways of paying consumers exist in the smart grid[63]). In Figure 24 is the composition of the electricity prices for household consumers presented[64]. The countries are listed after the lowest energy and supply costs.

Furthermore, although some air conditioners have the potential of being energy smart appliance, savings at large scale can only be realised when the standards and infrastructure are ready for all types of energy smart appliances for the whole EU in the coming years, as a result of the Ecodesign work on Smart Appliances.

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[62] http://www.eco-smartappliances.eu/Documents/Ecodesign%20Preparatory%20Study%20on%20Smart%20Appliances%20_Tasks%201%20to%206.pdf
[63] https://www.smartgrid.gov/the_smart_grid/consumer_engagement.html
[64] Eurostat, http://ec.europa.eu/eurostat/statistics-explained/index.php/Electricity_price_statistics

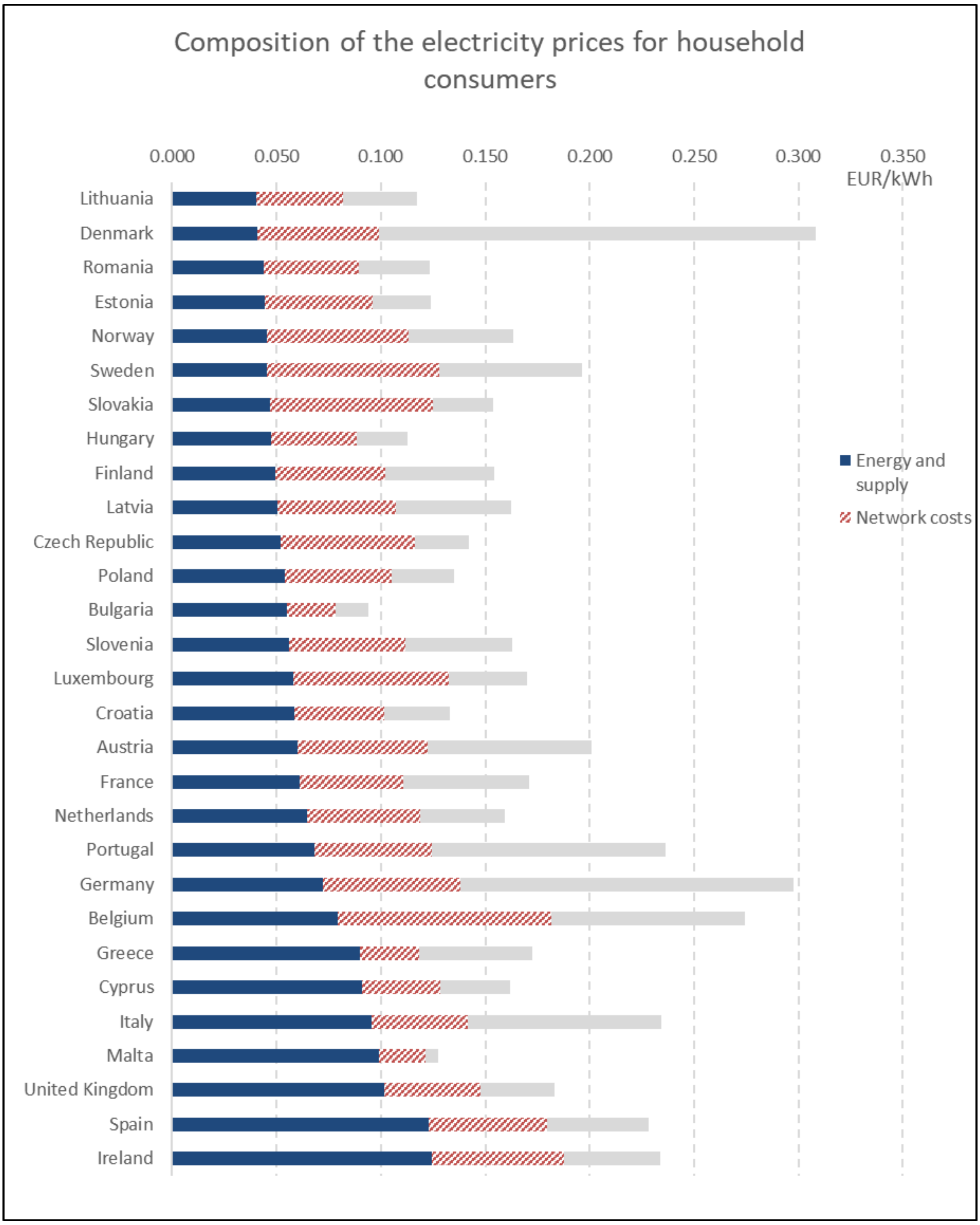

Figure 24: Composition of the electricity prices for household consumers[65]

[65] Eurostat, http://ec.europa.eu/eurostat/statistics-explained/index.php/Electricity_price_statistics

3.4 Conclusions and recommendations

System aspects use phase

Base on the assessment on use phase in section 3.1, the following conclusions and recommendations can be summarised.

One of the most important point for **air conditioner use phase energy consumption** is the number of equivalent full load hours that is used to compute the energy consumption and seasonal performances, this is the ratio of the specific yearly cooling load (in kWh/m^2/year) divided by the sizing coefficient (in W/m^2). For reversible air conditioners, the equivalent full load hours for heating is also important.

The seasonal performance metrics used to rate SEER and SCOP of fix air conditioners was thoroughly reviewed to evaluate the need for changing some of the **SEER and SCOP metrics parameters**, assessment and considerations have been presented.

Available data for **cooling loads** has been reviewed and compared to preparatory study simulations. It is recommended to keep these values presented in section 3.1.2.1. Correlations versus Cooling Degree Days (CDD) are proposed to enable to account for climate change and to derive cooling load values for EU28 countries which was not modelled in the preparatory study.

Significant **oversizing** of air conditioners is observed in practice and begins to be quantifiable due to the development of on-board performance measurement methods. Nevertheless, data is not yet available to properly quantify the oversizing and the consequences in terms of energy efficiency. In addition, present rating of very low loads (impact on performance of inverter compressor cycling below minimum speed) should be improved so that that effect might be included in the SEER metrics. It is therefore proposed to maintain the metrics hypothesis of "perfect sizing" where it is assumed the 100 % load of the building matches the declared rated capacity of the unit for now.

Available data of **sizing coefficients** has been reviewed and compared to preparatory study simulations. It is recommended to keep the preparatory study values.

The evaluation of the weighted average EU28 specific cooling load and sizing coefficients leads to a figure close to the one obtained in the preparatory study for **equivalent full load hours in cooling mode**, which is close to 350 hours. it is thus recommended to keep this value.

Regarding heating, the **heating loads and sizing coefficients** are derived from the space heating preparatory study and there is no change proposed regarding these values. Note that oversizing is also a concern here and might be included in future studies. To evaluate the environmental impact of **reversible air conditioner**s in Tasks 5 and 7, it is necessary to account for the fact that part of the reversible air conditioners is not used for heating in Europe. Most recent data source from Commission decision 2013/114/EU and stakeholder inputs results in the assumption of ca. 30 % of reversible air conditioners used for heating in average for stock units and 50 % for new units.

Low power mode hours have been reviewed. It is proposed to require two distinct **crankcase power values** for reversible units, respectively for cooling mode - measured at 20 °C - and for heating mode - measured at 12 °C. Cooling (or heating) only unit crankcase power measurement should be done at 20 °C (12 °C), as already written in standard EN14825:2016. Minor adjustments of **crankcase hours** are proposed. Above that, it is suggested that to include network standby for products with network capability, present standby hours can be used.

Regarding **portable air conditioners**, the impact of air infiltration on unit's capacity has been studied in heating and in cooling mode for both single duct and double duct units. It appears as a major factor for the product functionality and performance. A seasonal performance metrics is proposed in cooling mode for both products that include this effect. In heating mode, the seasonal efficiency of these units is very low. It is questionable to have at all reversible single duct units on the market as their seasonal performance is most likely lower than the one of a fan electric heater. For single duct and double duct reversible units in heating mode, it is recommended to adopt a SCOP metrics derived from the one for fixed reversible air conditioners and including the infiltration effect. This requires developing a test method to measure the condenser air flow rate of single duct air conditioners.

Stakeholders have underlined a risk that some of the highest efficiency units with high air flow at indoor unit may not be comfortable to the end-users and thus may lead to **higher consumption in real life.** To that purpose it is proposed to complete the test results of EN14511-3 with outlet temperature and humidity conditions and to require these values to be included in the technical documentation of products. Crucial **real life performances** are not available for fixed and portable air conditioners and the development of on-board measurement methods should be encouraged (it is proposed to start a standardisation work on this subject and to include electricity consumption measurement as well). Additionally, the impact of heat exchanger fouling may be important (as high as 20 % consumption increase) but it is supposed to be mitigated by a proper and regular maintenance, and with the help of filtering (including now automatic cleaning of filters). It is proposed to make washable filters on both heat exchangers mandatory for portable air conditioners as maintenance is currently not common for these units.

Regarding **comfort fans use patterns**, there is no new insight since the preparatory study and the number of equivalent full load hours are maintained at an average value for EU of 320 hours per year.

End of life behaviours

Through reviewing preparatory study and different sources, the average lifetime of air conditioners and comfort fans used in the current review study is presented in Table 18.

Table 18 Average lifetime assumptions adopted to the current study

Technology	Average lifetime (years)
Movables + Window Units	10
Small Split (<5 kW)	12
Large Split (>5 kW, incl. ducted)	12
VRF	15
Comfort fans	10

During the lifetime, air conditioners are more likely to be repaired in southern and south-eastern countries due to lower labour costs. The effect of a long lifetime should be assessed in later tasks to reveal to possible trade-offs between a long lifetime and efficiency. The most likely parts to malfunction or to be repaired by replacement is printed circuits boards and compressors based on stakeholder inputs. These parts are also available through the entire life of the products. When comforts fans malfunctions, they are expected to be discarded due to their low purchase price.

There are opportunities for Ecodesign in terms of product lifetime, enabling easy repair and maintenance as well as refrigerant leakage, however these can be problematic for market surveillance to verify, and Ecodesign which sets requirements at the beginning of the product's life may have limited impact on these aspects at the end of life.

The second-hand market for fixed air conditioners are very limited, but for both portable air conditioners and comfort fans this market is more widespread. Though, it is still assumed to be negligible and therefore not included in later tasks.

Regarding the best practice in sustainable product use it is important that the air conditioner is properly sized, well-placed and regularly maintained to reduce the energy consumption. These factors can reduce the cost of heating and cooling by up to 10%. It should therefore be discussed whether these information's should be made mandatory in the user manual.

Local infrastructure

The power sector is in a transition state moving from fossil fuels to renewable energy which challenges the existing infrastructure. In Europe the electricity system is characterized by a high share of renewable energy and high reliability.

Most countries have similar hourly load values with two peaks, one in the morning and one in the evening. The first peak fits well with the start of the workday and the second peak fits with the end of the workday. Between the two peaks there is a falling trend in the energy consumption. Though, it seems like the need for cooling is affecting the hourly load values for southern countries in the summer period. In these countries the peak load occurs in the middle of the day which could indicate a high use of air conditioners.

The consumer demand of energy smart appliance is still low due to the lack of products and the relatively small cost saving as a result of the non-flexible electricity prices and composition. Although some air conditioners have network and grid communication capability, but savings at large scale can only be realised when the standards and infrastructure are ready for all types of energy smart appliances in the coming years.

Annex 1: Preparatory study Building Energy Simulation input

The preparatory study for Lot 10 used simulations to derive cooling loads of air-conditioned buildings. Three different simple buildings were simulated, one house, one office and one shop, envelope insulation characteristics and air change properties for existing buildings and for new buildings built after 2006. The main characteristics of the three buildings are given in this annex.

Geometry

Residence

The residence is modelled as an apartment with characteristics as defined in Table 19. The exposed surfaces are the roof and facades in four orientations, while the floor is detached to a fully symmetric zone with identical boundary conditions. The exterior dimensions are 8 by 12.5 m with a total floor area of 100 m^2. The exposed wall area is 98 m^2. Distinct windows are placed in each wall with a window to wall ratio of 0.15, where operable shades are employed. The conditioned area represents the 30% of the total floor area, with south and west exposed facade.

Office

The office is modelled as a rectangular building with a core, conditioned zone and characteristics as described in Table 20. The exterior dimensions are 10 by 10 m with a total conditioned space of 100 m^2. The flour is detached to a fully symmetric zone with identical boundary conditions.

Retail

The retail store is modelled as a rectangular building with a single zone, as part of a strip mall with adjacent zones on two sides, characteristics as identified in Table 21 and in two orientations (north-south). The exterior dimensions are 25 by 25 m^2 with 50 m^2 of total conditioned floor area. Windows to wall ratio was set equal to 0.6, with operable shades.

Table 19: Construction, Internal Load and cooling equipment characteristics for Residence

Construction	Characteristic	Old	New
Zones	1 (conditioned)		
	1 (unconditioned)		
Floor area	100 m^2		
Roof	U-values range (W/m^{2}K)	0.2-0.9	0.2-0.5
Wall constructions	U-values range (W/m^{2}K)	0.2-1.2	0.3-1.2
Windows	16 m^2		
	Clear with operable shades		
Interior Load			

Infiltration	Air Change per hour (ACH)	0.8-1.5	0.2-0.4
Lighting	7-15 (W/m2)		
Equipment	7-10 (W/m2)		
Occupancy	1 person / 20 m2		
Equipment			
Thermostat	Cooling set point	25-26	25-26
Natural ventilation	Window operation available		

Table 20: Construction, Internal Load and cooling equipment characteristics for Office

Construction	Characteristic	Old	New
Zones	1 (conditioned)		
	1 (unconditioned)		
Floor area	100 m^2		
Roof	U-values range (W/m^2K)	0.2-0.9	0.2-0.5
Wall constructions	U-values range (W/m^2K)	0.2-1.2	0.3-1.2
Windows	25 m^2		
	Clear with operable shades		
Interior Load			
Infiltration	Air Change per hour (ACH)	0.8-1.5	0.2-0.4
Lighting	15-18 (W/m^2)		
Equipment	12-15 (W/m^2)		
Occupancy	1 person / 9 m2		
Equipment			
Thermostat	Cooling set point	24-26	24-26
Natural ventilation	Window operation available		

Table 21: Construction, Internal Load and cooling equipment characteristics for retail

Construction	Characteristic	Old	New
Zones	1 (conditioned)		
Floor area	50 m^2		
Roof	U-values range	0.2-0.9	0.2-0.5
Wall constructions	U-values range	0.2-1.2	0.3-1.2
Windows	30 m^2		
	Clear with operable shades		
Interior Load			
Infiltration	Air Change per hour (ACH)	0.8-1.5	0.2-0.4
Lighting	15-25 (W/m^2)		
Equipment	5 (W/m^2)		
Occupancy	1 person / 5 m2		
Equipment			
Thermostat	Cooling set point	22-26	22-26
Natural ventilation	Window operation available		

Building Envelope

Since the focus is to perform a comparative analysis, building characteristics were selected to be representative and adaptive to weather variation. Opaque and transparent surfaces exhibit local variation in Europe, so they were not selected to be uniform.

Roof, wall, floor and window characteristics, were initially identified through national questionnaires and the calculated U-values were in good agreement with Ecofys/Eurima data concerning old and new constructions. U-values for both opaque and transparent elements, as well as for old and new constructions, were classified in five groups, being representative of building characteristics adaptive to climate Table 22.

Table 22: U values for old and new construction in different climates

U-values (W/m^2K)	Old Construction	New Construction
COLD: FI, SE		
Roof	0.2	0.15
Wall	0.3	0.2
Floor	0.2	0.18
Windows	2	1.4
MODERATE: AT, BE, DK, FR, DE, IE, LU, NL, UK		
Roof	0.5	0.25
Wall	1.2	0.45
Floor	0.8	0.53
Windows	3.44	2.95
WARM: GR, IT, PT, ES, CY, MT		
Roof	0.9	0.5
Wall	1.2	0.6
Floor	0.8	0.53
Windows	5.68	3.44
CENTRAL EASTERN EU COUNTRIES: PL, SK, SI		
Roof	0.5	**0.23**
Wall	0.6	0.34
Floor	0.53	0.44
Windows	3.44	1.4
BALTIC REPUBLICS: EE, LV, LT		
Roof	0.5	0.2
Wall	0.6	0.26
Floor	0.53	0.29
Windows	3.44	1.4

Internal gains

Internal loads in buildings are due to occupancy (people / m^2), lighting and electrical equipment. We have assumed that there is no local variation, but there is a clear dependence on the type of use. Magnitude and schedules are listed in Table 23, Table 24, Table 25 and Table 26, respectively.

The heat input per person was considered according to ISO 7730, while for the artificial lighting it was assumed that 60 % of the input contributes to the zone heat balance as convective heat and 40 % as radiative.

Table 23: Internal Gains

	Office	Resident	Shop
Lighting (W/m^2)	15	10	25
Equipment (W/m^2)	15	7	10
Occupancy (person / m^2)	1/9	1/20	1/5

Table 24: Schedules for shop

Working Day				
HOUR	**OCCUPANCY**	**TEMPERATURE**	**LIGHTING**	**EQUIPEMENT**
00-08	0	SetBack	0.05	0.05
08-09	0.1	SetPoint	1	1
09-10	0.25	SetPoint	1	1
10-19	0.5	SetPoint	1	1
19-24	0	SetBack	0.05	0.05
Weekend				
00-08	0	SetBack	0.05	0.05
08-09	0.1	SetPoint	1	1
09-10	0.25	SetPoint	1	1
10-19	0.5	SetPoint	1	1
19-24	0	SetBack	0.05	0.05
00-08	0	SetBack	0.05	0.05

Table 25: Schedules for office

Working Day

HOUR	OCCUPANCY	TEMPERATURE	LIGHTING	EQUIPEMENT
00-08	0	SetBack	0.05	0.05
08-09	0.1	SetPoint	1	1
09-10	0.25	SetPoint	1	1
10-19	0.5	SetPoint	1	1
19-24	0	SetBack	0.05	0.05
Weekend				
00-24	0	SetBack	0.05	0.05

Table 26: Schedules for residences

Working Day				
HOUR	**OCCUPANCY**	**TEMPERATURE**	**LIGHTING**	**EQUIPEMENT**
00-07	0	SetBack	0	0.05
07-09	1	SetPoint	1	1
09-17	0	SetBack	0.05	0.05
17-23	1	SetPoint	1	1
23-24	0	SetBack	0	0.05
Weekend				
00-09	0	SetBack	0	0.05
09-23	1	SetPoint	0.05	1
23-11	0	SetBack	0	0.05

Infiltration and Ventilation

Infiltration rates were set equal to 0.4 and 0.6 air change per hour for new and old constructions respectively. Ventilation rates have been calculated as a function of occupancy (number of people and schedule of use) using the standard value of 8 l/s/person.