Service Contract to DGTREN

Preparatory study on the environmental performance of residential room conditioning appliances (airco and ventilation)

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TECHNICAL ANALYSIS OF EXISTING PRODUCTS

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http://www.ecoaircon.eu

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4 TECHNICAL ANALYSIS EXISTING PRODUCTS

Scope: This entails a general technical analysis of current products on the EU-market and provides general inputs for the definition of the Base cases (task 5) as well as the identification of part of the improvement potential (task 7), i.e. the part that relates to the best existing product on the market.

Introduction

The technical analysis of existing air conditioners on the EU market is a large piece of work because of the complexity of the product itself, which performances vary with climatic conditions, and also because of the large number of different technical products covered by the present scope.
A subtask as then been added to explain the main principles of functioning of air conditioners and also to present the technical characteristics of the components that have a direct influence on its energy performances. This is the first subpart of this analysis in paragraph 4.1.1. Among those components, some are used only for some of the technical categories of air conditioners previously identified; this is also described in that subtask. Following subtasks follow the structure of the MEEuP, 4.1.2 – production phase, 4.1.3 – distribution phase, 4.1.4 – use phase (product) , 4.1.5 - use phase (system), 4.1.6 – end-of-life phase.
Because conditions of use are so important to determine energy consumption, we have included the reconstitution of operating conditions of air conditioners using building and system simulation tool in part 4.1.5 – use phase (system). Of course, the ecodesign of the unit will not enable to change these conditions but it will be greatly affected by those ones.

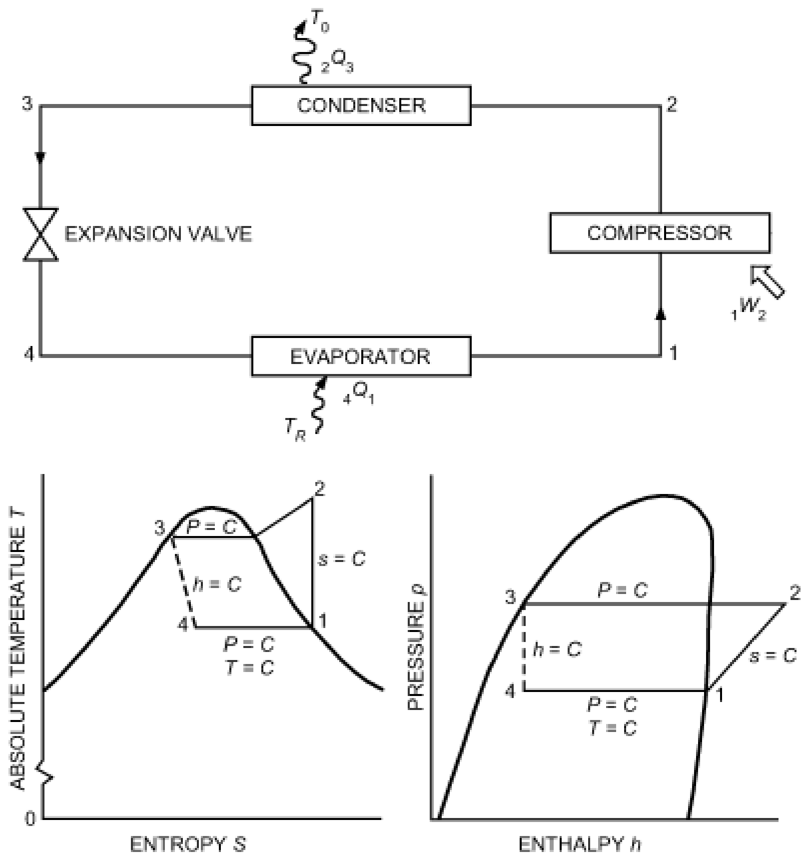
Above specialised literature, it has been necessary to access wider information to cover the existing products on the EU market. Different sources have been used depending on the technical categories of products. For Eurovent certified products (split, multi-split, window/wall, mini-chillers), public information on main energy performance characteristics has been extensively used. Consultation of the technical database of Eurovent certified products was also possible. Information is then supposed to give a good overview of actual EU market. It is also to be noted that the window/wall category in the Eurovent database also include other package air conditioners of the air to air type than window/wall air conditioners. And that the technical trends for window/wall include all package air to air units below 12 kW cooling capacity in the Eurovent catalogue.
About portable air conditioners (single duct, double duct and mobile split air conditioners), we had no access to any kind of technical or energy performance database. As a consequence, we collected information as supplied by manufacturers and importers on their websites and/or information brochures. We thus constituted three set of heterogeneous data with available characteristics. The data base covers 80 single duct air conditioners, 40 mobile split air conditioners and 10 double duct air conditioners.
Concerning central air conditioners (US style) are air cooled air conditioners, either single split or self contained. There is no public information on manufacturers' websites for the European market. Sells are believed to be direct imports from the USA and mainly not for the residential sector but for the commercial and industrial sectors. This very little share of the EU market between 0 and 12 kW has been identified in task 2 to amount to a maximum of 20 000 units. Moreover, energy efficiency minimum of performance applied in the USA (please refer to task 1) for these products are among the highest in the world. Whether required, a specific sensitivity analysis may be led in task 8 or alternatively lead to conclusions or recommendations for possible future action within the forthcoming ecodesign activities. But to keep a specific category does not seem necessary in the following tasks for these products.

4.1 Air conditioners generic technical description

This part intends to give a technical representation of air conditioners that lie in the scope of the study. The aim pursued here is to gather basic information needed for the environmental impact of the products but also for the selection of base cases.
Starting from the basic description of the refrigeration cycle, for each of the main technical categories identified, main characteristics are derived. The analysis is led by main components of the refrigerating cycle. For air conditioners that are certified by Eurovent-Certification (this encompasses single and multi-split air conditioners, window/wall air conditioners and cooling only mini-chillers), the publicly available database (Eurovent, 2006) is used to draw statistics. When it relates to more detailed information, the technical database of products of Eurovent-Certification (Eurovent-tech, 2006) has been used. More details on these two sets of data can be found in chapter 5 of this study related to base case analysis, paragraph 5.1.1.2. For other technical information as compressor description incl. part load control, and for other air conditioner types e.g. portable air conditioners-, different sources from manufacturers websites, technical literature … have been gathered.

4.1.1 Air conditioning and heat pump thermodynamic cycle

Air conditioner products in this study are based on the vapour compression cycle, based itself on the Carnot refrigeration cycle, the reference ideal thermodynamic cycle with two sources at different temperatures. Following Carrier words, the air conditioning system takes heat where it is not desired where it does not matter, for instance outdoors for air cooled air conditioners.



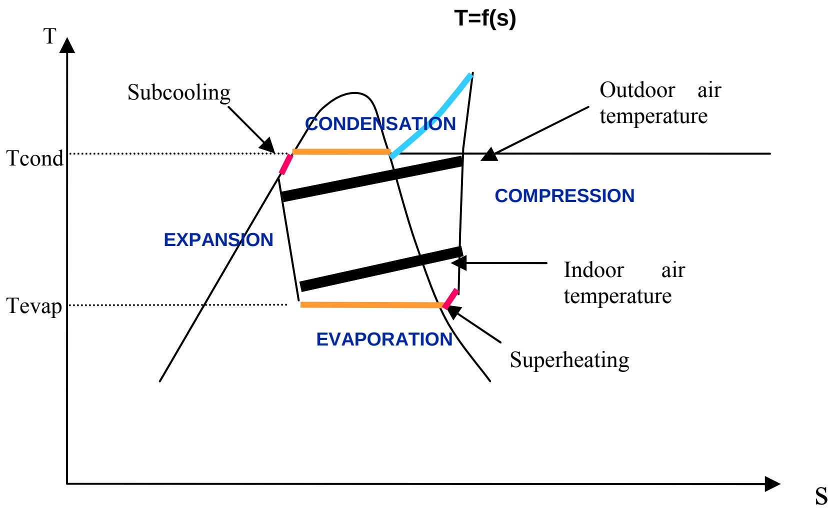


Figure 4-1: Air conditioner schematic diagram, thermodynamic cycles in diagrams LnP – H and T-S of air conditioning cycle

The figure above translates the basics of the thermodynamic cycle as used in air conditioners and heat pumps.
In the **evaporator**, heat is extracted from the indoor air to the refrigerant. During this process, refrigerant vaporizes from binary state to superheated vapour.
In the **compressor**, the refrigerant vapour is then compressed to high pressure: high pressure enables to reach high temperature level.
In the **condenser**, high temperature level enables to release heat to outdoor air while the refrigerant condenses.
Then, in the **expansion device**, refrigerant liquid expands through a flow restrictor in order to recover low pressure and temperature. It then enters the evaporator to capture indoor air heat again.

The **reverse cycle** is the same as the cooling cycle except that the heat is extracted outdoor in cold ambient and released indoor at higher temperature. Then, outdoor temperature and indoor temperature are reversed on Figure 4.1.

Then, main components of the cycle are: the refrigerant fluid, both heat exchangers (evaporator and condenser), the compressor and the expansion device.

Thermal performance, EER, COP, heating and cooling capacities

Cooling capacity is defined as:
$Pc = m . (H1 – H4)$
With m the refrigerant mass flow rate and H the enthalpy of the fluid at point 1.

The ideal fluid compression work is defined as:
$W = m . (H2 – H1)$

The condenser released heat is defined as:
$Ph = m . (H3 – H2)$

Then the coefficient of performance EER (cooling mode) can be calculated as Pc / W and COP (heating mode) as Ph / W.

Then, the larger the difference between outdoor and indoor temperature sources, the larger the difference in condensing and evaporating temperatures and consequently the larger the compression work and the lower the EER and COP.

4.1.2 Refrigerant fluid

There are very few refrigerant fluids used in air conditioners (for products in the scope). This restricted offer is primarily linked to the high number of criteria the refrigerant fluid must fulfil (Duminil, 1996), energy performance, safety –toxicity and flammability, ODP and GWP, technical: it must not interact with components of the system (in that direction, ammonia despite its excellent energy efficiency characteristics is not compatible with copper), economic – it must be affordable, meaning used by a significant part of the industry and not demanding too much modifications on the systems if possible.

HCFC

There are several methods established to select the best refrigerant for a given application, e.g with the knowledge of the temperature levels of hot and cold sources. R22 was for a long time the universal refrigerant because very performing for a wide range of application including air conditioners.

CFC and HCFC phase out

Signed in 1987, the Montreal Protocol on Substance that deplete the Ozone Layer aimed at phase out the consumption and production of several ozone depleting substances among which the chlorofluorocarbons (CFCs) and hydrochlorofluorocarbons (HCFC) that were used in air conditioning appliances. Noticeably, R22 (an HCFC) was the more common refrigerant and is nowadays banned in new equipment in Europe. As explained by (Dieckmann and Little, 1999), the Copenhagen Amendments to the Montreal Protocol in November, 1992 established an HCFC phase-out timetable, as shown in Figure 4-2. The basic approach is to establish a "Cap" based on combined CFC and HCFC use, and then periodically to downsize refrigerant production as a percentage of the Cap. The Cap is expressed in terms of ODP weighted production (units: ODP-kg) and was established as 3.1% of ODP weighted CFC consumption plus ODP weighted HCFC consumption in 1989. As shown in Figure 4-2, consumption is limited to the Cap beginning in 1996, and stepped reductions from the Cap occur in 2004, 2010, 2015, and 2020, with a final phase out in 2030. In December, 1995, at the Seventh Meeting of the Parties, an adjustment was adopted reducing the cap by approximately 5% overall -- the new formula being 2.8% of ODP weighted CFC consumption plus ODP weighted HCFC consumption. In addition, HCFC consumption from 2020 to 2030 is restricted to servicing existing air conditioning and refrigeration equipment. The Montreal Protocol HCFC phase-out schedule seems to have stabilized in terms of both the timing of phase-out steps and the associated consumption limits (i.e., the formula for determining the Cap).

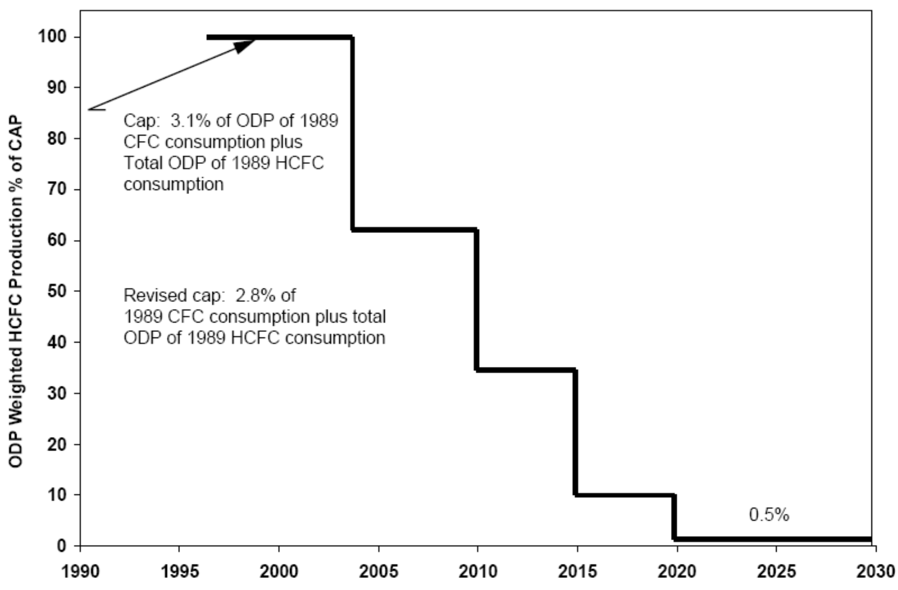


Figure 4-2: HCFC phase-out timetable

As reported by (UNEP, 2005), Europe (EU 25) is in advance over the timetable as far as it concerns HCFCs consumption. In 2004, ODP emissions linked to EU consumption of ODP chemicals was only already 1,450.9 ODP tones with the new initial cap being 8,228.1 which amounts around 18 %. As mentioned in EC regulation 2037/2000, HCFC-22 is forbidden in air conditioning application with cooling capacity inferior to 100 kW as follows: "the use of hydrochlofluorocarbons shall be prohibited from 1 July 2002 in equipment produced after 30 June 2002 and of reversible air conditioning/heat pump systems where the use of hydrochlorofluorocarbons shall be prohibited from 1 January 2004 in all equipment produced after 31 December 2003". Concerning refill of existing installations, "from 1 January 2010, the use of virgin hydrochlorofluorocarbons shall be prohibited in the maintenance and servicing of refrigeration and air-conditioning equipment existing at that date; all hydrochlorofluorocarbons shall be prohibited from 1 January 2015." As a consequence, since 2005, no reversible R22 air conditioner is sold on the European market and since 2003 for cooling only air conditioners. It is difficult to evaluate whether these measures enable this specific air conditioner sector [0-12 kW] to respect the protocol objectives because of the lack of detailed information on potential necessary fluid refill of HCFC 22 split and multi split systems and of emission levels in 1989. However, still following the same EC regulation 2037/2000, "before 31 December 2008 the Commission shall review the technical and economic availability of alternatives to recycled hydrochlorofluorocarbons". Since R407C has been developed as a possible candidate for direct drop-in, refrigeration industry seems well prepared to a complete ban that would then ensure a complete achievement of the Montreal protocol for this air conditioning segment.
Replacement options were investigated and finally two main fluids are now sharing the European market of air conditioners with cooling capacity inferior to 12 kW (Eurovent, 2006): R410A and R407C. Some single duct units are also designed with propane (R290). Whether propane has already been in use for a few years in domestic refrigerators and freezers, its use in small capacity air conditioners has been authorized only recently, as mentioned in task 1, with tight limits on the quantity of gas and then on the cooling capacity range of package air conditioners it can apply to. Both other fluids, R410A and R407C, belong to the hydrofluorocarbons (HFC) family.

GWP

The fluids in use have no depletion effect on the stratospheric ozone but have a global warming impact and are included in the Kyoto protocol. An assessment of global warming impact of an appliance containing this type of refrigerant must take into account direct emissions during lifecycle (leaks, reclaiming) and indirect emissions (emissions linked to the electrical consumption), for which matters the thermodynamic performance or energy efficiency of the refrigerant. This is generally done with the TEWI calculation as already explained in task 1 in the Nordic Ecolabeling scheme for heat pumps. Global warming potential (GWP) is used to compare the abilities of different greenhouse gases to trap heat in the atmosphere. GWP is based on the radiative efficiency (heat-absorbing ability) of each gas relative to that of carbon dioxide (CO_2), as well as the decay rate of each gas (the amount removed from the atmosphere over a given number of years) relative to that of CO_2. GWP is calculated over a specific time interval which value must be stated whenever a GWP is quoted. The GWP provides a construct for converting emissions of various gases into a common measure, which allows climate analysts to aggregate the radiative impacts of various greenhouse gases into a uniform measure denominated in carbon or carbon dioxide equivalents (EIA, 2007).

Refrigerant	Composition	GWP (100yr)
R22 (HCFC)	Pure fluid	1700
R407C (HFC)	R32/R125/R134a	1653
R410A (HFC)	R32/R125	1954
R290 (HC)	Propane	3 - ~20 (*)

Table 4-1: Global warming potential of some refrigerants, for 100 year integration, regulation EC/842/2006. (*) Reference of MEEuP for GWPs is the IPPC 2001 third assessment report, but no value is available for R290 whose GWP is low and exact value varying between 3 and about 20 depending on references.

Thermodynamic performances

Phasing out R22 for HFCs or propane mixture corresponds to an energy efficiency loss and also higher cost to manufacture air conditioning systems. Starting from the thermodynamic cycle in Figure 4.1, the refrigerant properties in indeed the primary component that dictates the air conditioning cycle energy efficiency. If thermodynamics basics only acknowledges the temperature of the sources, the refrigerant saturation curves give the correspondence between temperatures of use and pressure levels and then the compressor design and efficiency. Cooling capacity is a direct function of the vaporization or latent heat of the refrigerant. Superheat at aspiration that is needed for compressor safe functioning degrades the cycle efficiency and is also dependent on the refrigerant used. The same apply to temperature discharge of the compression.

Efficiency comparison with HCFC22 of different replacing fluids for a specific cycle (with a certain number of hypothesis on temperatures and components efficiency needed for the evaluation) by (Devotta, 2001) is reported as an illustration in the table below. Condensation temperature is 55 °C, evaporation temperature is 7.2 C.

Refrigerant	% Relative to HCFC-22				
	COP	Cooling capacity	Pressure ratio	Compres-sor power	Discharge pressure
R-407C	−1.76	1.72	6.60	1.75	7.67
R-410A	−8.90	41.21	−2.29	9.81	55.60
HFC-134a	4.40	−33.00	13.75	−4.27	−31.45
HC-290	1.00	−14.13	−6.87	−1.00	−12.42
HFC-32/HFC-134a (30/70 by wt.%)	1.00	−1.00	8.80	−1.00	1.30
HFC-32/HFC-125 (60/40 by wt.%)	−8.32	45.40	2.00	8.30	57.72
HFC-32/HFC-125/HFC-134a (30/10/60 by wt.%)	0.34	2.93	7.44	0.35	6.71

Table 4-2: Performance comparison of different refrigerants for a standard air conditioning cycle (Devotta, 2001)

R407C and propane (HC-290) have comparable COP (COP in cooling mode, generally called EER in Europe) values. R410A exhibits higher COP losses but a significant capacity gain.

The advantage of R-407C was it could be used as a drop-in refrigerant in existing systems, which were initially designed for operating with R-22. R-407C has nearly the same cooling capacity and operating pressures as R-22. Nevertheless, the efficiency of systems is usually lower with this drop-in refrigerant (5% or so lower than R-22). Despite, this refrigerant has been widely used in Europe for air conditioners due to the accelerated R-22 phase out.

The vapor pressure of R-410A is 60% higher than R-22, and this refrigerant couldn't be easily adapted to R-22 systems. Due to high pressure, compressors need to be redesigned completely and also the heat exchangers needs to be optimized to accommodate lower volumetric flow rates associated with the use of R410A (Kim, 2000).

If the simple thermodynamic cycle analysis shows that the cycle efficiency of R410A is lower than that of HCFC22, the actual energy efficiency of R410A air conditioners may already be similar or better to that of HCFC22 due to improved compressor efficiency and reduced energy losses in other components of the refrigeration system.

(Kim, 2000) presents results of performance tests for R-22 and four alternative fluids (R-134a, R-32/134a (30/70%), R-407C, and R-410A) at operating conditions typical for a residential air conditioner. The study was performed in an experimental breadboard water-to-water heat pump in which a water/ethylene glycol mixture was used as the heat transfer fluid. The heat exchangers representing the evaporator and condenser were counter flow and cross flow, respectively. In tests performed, R-410A had the highest coefficient of performance but for lower frequency speed of rotation. Test results for the system and data characterizing the performance of the heat exchangers and compressor are presented. The impact of the wide variations in the different alternative fluid properties on the system's operation and performance is investigated.

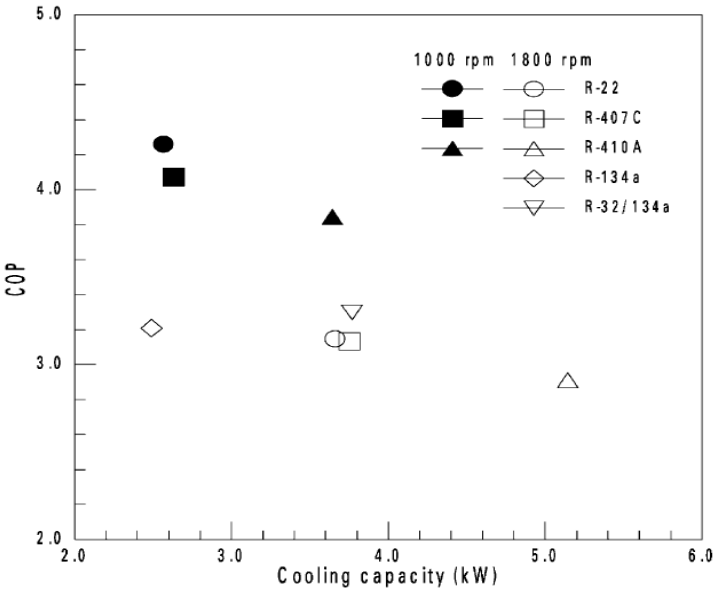


Figure 4-3: Cooling capacity (kW) and coefficient of performance of different fluids (Kim, 2000)

The capacity of R-410A at 1000 rpm (3644 W) is almost the same as the capacity of R-22 at 1800 rpm (3663 W); however, the COP (EER) of R-410A is higher by 22%.

Systems optimized for this refrigerant have smaller heat exchangers for the same duty and efficiency similar to the ones obtained with R-22 refrigerant. On the other hand, keeping large heat transfer surface areas enables to significantly improve system efficiencies. An additional difference of R-410A is the lower liquid density of the refrigerant (approximately 12% lower than for R-22). The refrigerant charge is then approximately 25 to 30% lower than the required charge of a R-22 system for comparable duty and efficiency.

Concerning R290 – propane, (Park, 2006) calculated that energy efficiency of the thermodynamic cycle for air conditioning application is equivalent to the one of R22, from 0 to 5 % better. However, refrigerant charge to obtain the same cooling capacity is cut by 50 %. Cooling capacity decreases by 10 %. Propane is seen by the air conditioning industry as a viable technical alternative refrigerant in case legislation would ban refrigerant fluids with positive GWPs. But safety is very strict in Europe for propane. Moreover, its use would not be compatible with the split system type but only with package air conditioners (Dieckmann, 1999), inducing high transformation costs for the industry and the end users.

The conclusions drawn on alternatives refrigerant are valid also for the COP in the heating mode for reversible air conditioners in the scope of this study.

Temperature glide

The most significant difference between R410A and R407C is the fact that R-410A is a near-azeotropic substance, while R-407C is a zeotropic substance. Zeotropic substances have different equilibrium composition in the liquid and vapour phase, thus having a different pressure-temperature equilibrium at saturation that for liquid and vapour respectively. This results in a temperature glide as shown in the figure below. This means at certain given pressure, the liquid saturation temperature in the heat exchanger is not the same as vapour saturation temperature at the same composition. Refrigerant R-407C has a temperature glide of about 6K (Bigot, 2001). This temperature glide leads to an impact on boiling heat transfer. The pool boiling heat transfer coefficient decreases as temperature glide increases. This cycle of events decreases the heat rate transfer and the overall performance. Moreover, since it is a non-azeotropic refrigerant mixture (NARM), fractionation may occur in case of the leak in the system. Concretely, it also means that when leaks occur, all the refrigerant must be removed and new R407C fluid with the right composition must be used.

Refrigerant type and charge of EU air conditioners

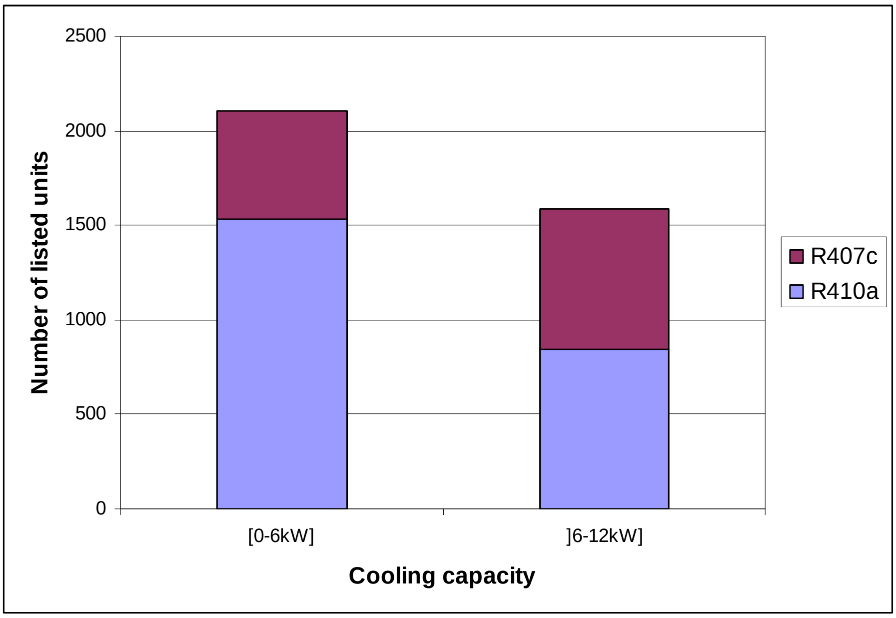


Figure 4-4: Refrigerant type of single split air conditioners cooling only and reversible, (Eurovent, 2006)

R410A refrigerants is the more common refrigerant for single split air conditioners, either cooling only or reversible.

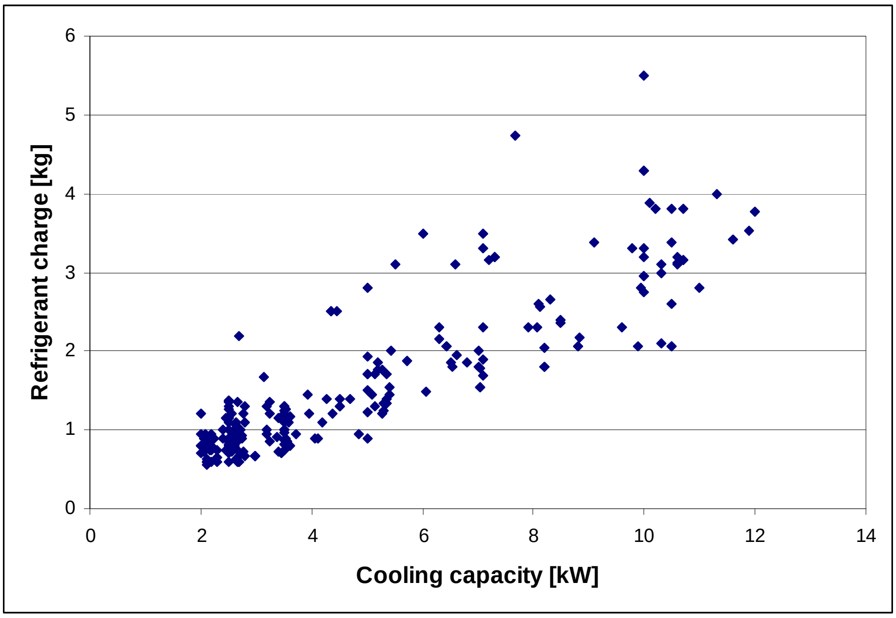


Figure 4-5: Refrigerant charge as a function of capacity of single split air conditioners cooling only and reversible, (Eurovent-tech, 2006)

Values of 0.31 kg of refrigerant by kW of cooling capacity can be kept as the average value for R410A while it is slightly higher for R407C with 0.34 kg/kW.

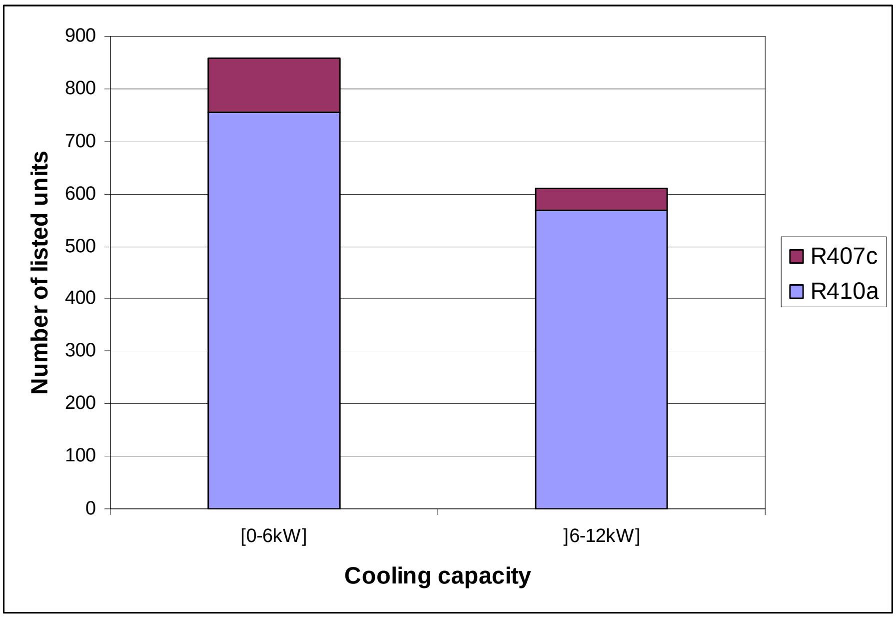
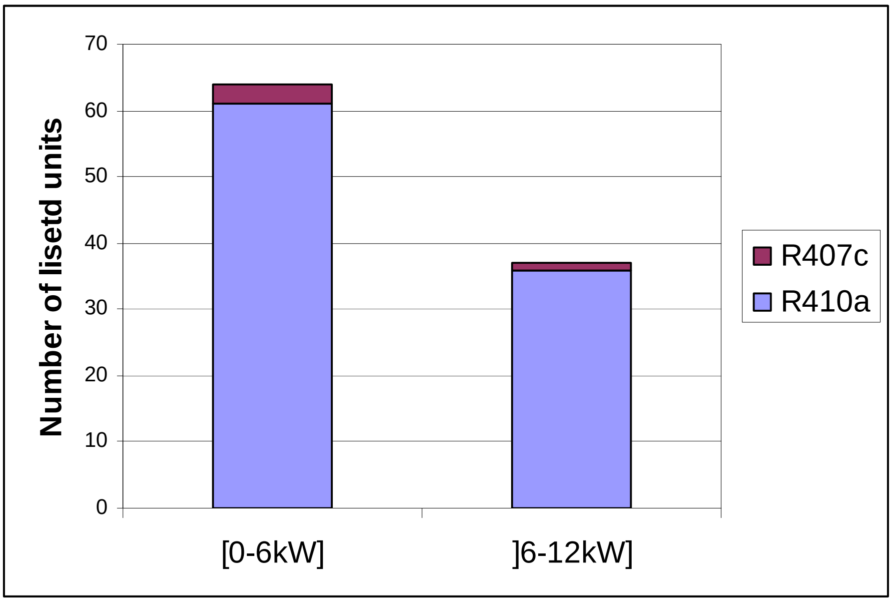


Figure 4-6: Refrigerant type of multi split air conditioners cooling only and reversible, (Eurovent, 2006)

Average charge for multi-split is slightly higher than for single split systems despite mainly R410A is used. With this fluid, average figures of 0.33 kg/kW has been calculated, between 5 and 10 % higher than for single split units.



For single and double duct air conditioners, very few refrigerant charge data have been identified. Refrigerant charge published vary more than for other products, with values ranging from 0.15 to 0.5 kg / kW. For R410A, which seems the main refrigerant in use for this type of air conditioners, the average charge identified is also around 0.3 kg/kW. For split mobile, more common refrigerant is also R410A with 0.4 kg/kW value.

Window/wall units exhibit lower refrigerant charges with average values of 0.27 kg/kW in average. However, representativeness of data used is low and R410A values available are typically lower, between 0.2 and 0.25 kg/kW.

For mini-chillers, R410A is also the standard fluid now with average refrigerant charge of 0.3 kg /kW.

4.1.3 Compressor

There are two ways to increase the pressure of gas or vapors. The first method is by reducing the volume that is occupied by the vapor or gas. Vapors are sucked through suction ports and are geometrically trapped. By exerting work done by the compressor to the vapor, pressure is increased. Compressors that use this method are known as positive displacement compressors. Reciprocating, rotary, screw and scroll compressors fall into this category. Another method applied to increase pressure of vapor and gas is by rotating vapor or gas in the compression chamber by means of impeller. High rotating velocity of vapor or gas produces a large centrifugal force which in turn increases the pressure. This type of compressor, known as centrifugal compressor is not used below cooling capacities of several hundred kilowatts.
Only reciprocating, scroll and rotary compressors are likely to be used in air conditioners in the cooling capacity range of [0-12 kW], screw compressors being reserved to higher cooling capacities (above 70 kW). On this range of cooling capacity [0-12 kW] compressors are hermetic: the motor and compressor are contained in the same housing, with the motor shaft integral with the compressor crankshaft and the motor in contact with the refrigerant.

Rotary compressors
In a rotary compressor the refrigerant is compressed by the rotating action of a roller inside a cylinder. The roller rotates eccentrically (off-centre) around a shaft so that part of the roller is always in contact with the inside wall of the cylinder. A spring-mounted blade is always rubbing against the roller. The two points of contact create two sealed areas of continuously variable volume inside the cylinder. At a certain point in the rotation of the roller, the intake port is exposed and a quantity of refrigerant is sucked into the cylinder, filling one of the sealed areas. As the roller continues to rotate the volume of the area the refrigerant occupies is reduced and the refrigerant is compressed. When the exhaust valve is exposed, the high-pressure refrigerant forces the exhaust valve to open and the refrigerant is released.

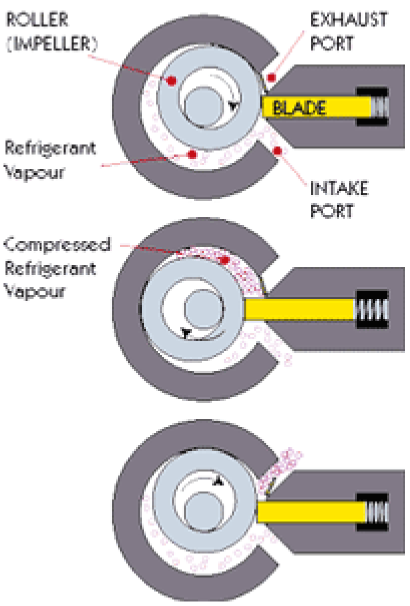


Figure 4-7: Compression cycle in rotary fix-vane compressor (FSCC, 2007)

These rotary fix-vane compressors are typically found in fridges and freezers and small air conditioners with cooling capacity up to 2 kW. R410A series have been developed for cooling capacities until 8 kW (LG, 2007).

Scroll Compressors
Scroll compressors are widely used in automotive, residential and commercial sectors. Scroll compressor capacity ranges from 1 to 20 HP. The scroll compressor uses one stationary and one orbiting scroll to compress refrigerant gas vapours from the evaporator to the condenser. The upper scroll is stationary and contains the refrigerant gas discharge port. The lower scroll is driven by an electric motor shaft assembly imparting an eccentric or orbiting motion to the driven scroll. That is, the rotation of the motor shaft causes the scroll to orbit, not rotate, about the shaft centre.

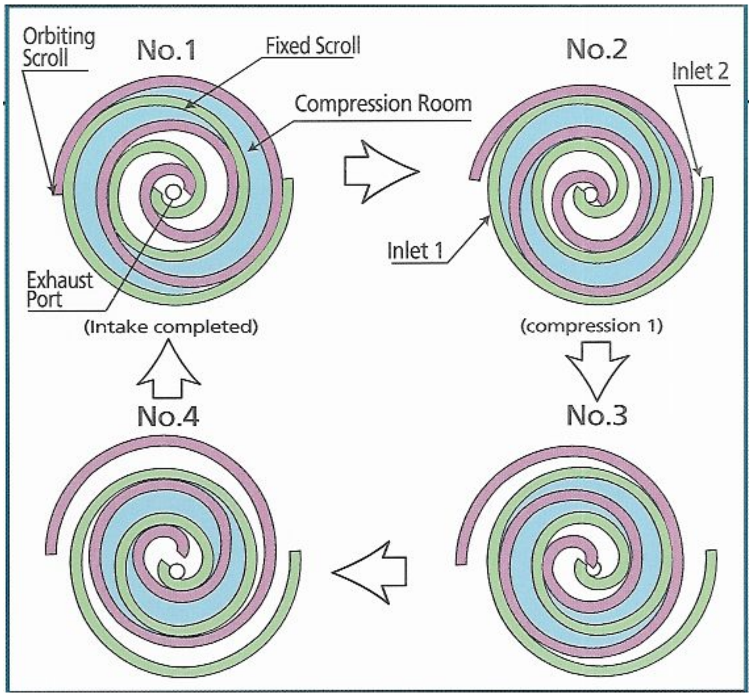


Figure 4-8: Compression cycle in scroll compressor (ACE, 2007)

Piston (Reciprocating) Compressor

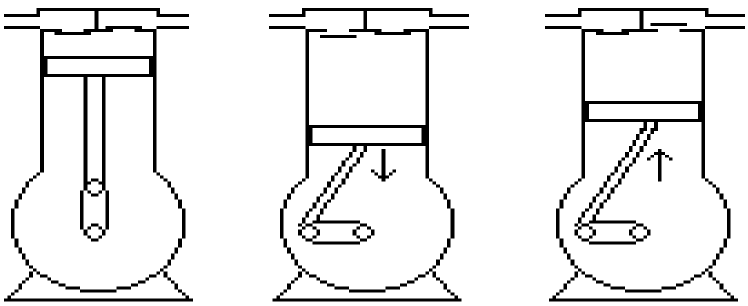


Figure 4-9: Compression cycle in reciprocating compressor (Remotelab, 2007)

Capacity ranges from 1 to several hundreds of HP. It is convenient for refrigerants that require relatively small displacement and condensing at high pressures.
There is small clearance gap between the top of the piston and the valve plate to avoid the piston striking the valve plates. A certain amount of vapour will remain in between the gap and not all of the high-pressured vapour escapes through the discharge valve at the end of the compression cycle. At the beginning of a cycle, the piston moves downward, allowing the vapour to expand and reducing pressure. The lower pressure now in the cylinder compared to the pressure in the suction line forces vapour to be sucked in. When pressure in suction line and in cylinder equalizes, the suction valve

closes. Next, the piston moves upwards and compresses the vapour, increasing its pressure. Now, higher pressure in cylinder than of the discharge line forces the discharge valve to open. High pressured vapour is discharged. Certain amount of vapour remains and the cycle repeats. These valves are responsible of head losses at suction and discharge ports but also enable the compressor to adapt to the varying pressure conditions.

Compressor performance
Compressor manufacturers publish performance maps of compressor performances (mass flow rate, cooling or heating capacity and power input) for a table of varying evaporating and condensing temperature as requested by the standard EN 12900. This table is directly incorporated in assembler tools in order to model the impact of the compressor on the complete performance of the air conditioning cycle. It is to be noted that performances should also be published together with these tables but are not most of the times.
Three ratios are widely used because useful to seize the differences in efficiency of the compressors, motor efficiency, volumetric efficiency, isentropic efficiency.

Motor efficiency is the ratio between the power delivered to the fluid and the power delivered to the compressor group. It includes transmission losses and also inverter losses (for inverter driven compressors).

$$E_m = \frac{W}{P_e}$$

Motor optimization being a cost-optimization compromise, larger motors are generally more efficient than smaller ones. In that power size range accepted efficiencies range from approximately 88 % for a 2 kW motor (electric input) to 95 % for a 75 kW motor (ASHRAE, 2004).

The **volumetric efficiency** is the ratio between the swept volume (displacement multiplied by number of rotations per time unit) and the real volumetric flow rate. It translates for instance the clearance loss of volumetric flow rate.

$$E_v = \frac{D_v}{V_s}$$

Where Dv is the refrigerant volumetric flow rate and ΔH the enthalpy difference between fluid inlet and outlet.

The **compressor efficiency** is a measure of the difference between the isentropic compression enthalpy difference (as a perfect reference) and the real enthalpy difference.

$$E_{is} = \frac{D_m.\Delta H_{is}}{W} \qquad (2.3)$$

Because, without the measurement of the exhaust and shell temperature of the compressor it is not possible to separate the motor and isentropic efficiency ratios, generally, **total efficiency** is used as the product of Em by Eis.

$$E_T = E_m.E_{is}$$

Classical representation of volumetric and total efficiencies are drawn in function of the compression ratio (ratio of the high pressure –condensing- to the low pressure – evaporating). Volumetric efficiency can generally be approximated by a linear curve as a function of pressure ratio for all compressor types. Typical compressor performance curves for a scroll compressor are shown in the figure hereafter.

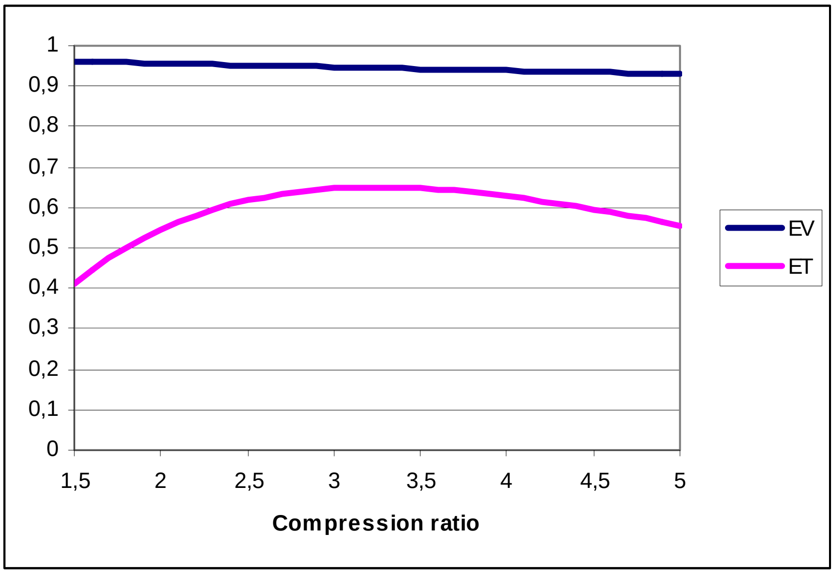


Figure 4-10: Typical performance curves (Total efficiency $–E_T$ and Volumetric efficiency E_V) of a scroll compressor for air conditioning application (optimum compression ratio around 3)

Scroll compressors
Scroll compressors are the more efficient compressors for low capacity applications today at least at peak conditions. Indeed, scroll compressors are designed for a specific built-in volume ratio. It means that there is a clear peak optimum efficiency for a given compression ratio with rapid decrease of isentropic efficiency before and after this point. For air conditioning application, optimum compression ratio lies between 2.5 and 3.5 depending on the manufacturers. For reversible applications, a compromise towards higher compression ratios has to be found. Because of its design, volumetric efficiencies of scroll compressors are high with values of 0.95 at design point. Typical EER would lie between 3 and 3.2 at rating conditions (R410A) for higher capacity ranges. For lower capacity range, below 12 kW cooling capacity, there is less offer and energy efficiency also seems lower in average, between 2.8 and 3.1 (also for R410A).

Reciprocating hermetic compressors
Hermetic compressors have not been developed with R410A refrigerant. As compared to scroll compressors, peak efficiency at rated conditions may be lower. Indeed, most efforts have been made to improve scroll energy efficiency in the last years and little for reciprocating that are progressively replaced by scroll compressors for small cooling capacities and screw compressors for larger ones. The main manufacturing advantage of these two latest categories being the smaller number of pieces in movement during compression that increases the compressor reliability. Lower peak efficiency results of several backdraws of this technology, clearance volume, admission valve that induces pressure losses at the suction side and important mechanical friction losses because of the important number of moving parts. Nevertheless, because of the presence of inlet and outlet valves energy efficiency is better maintained at pressure ratios both lower and higher than the ratio of the rating efficiency point, that may be a pros when comparing air conditioning units not on a single rating point but on their yearly consumption.

Rotary fix-vane
Rotary compressors have a high volumetric efficiency because of the small clearance volume and correspondingly low re-expansion losses inherent in their design (ASHRAE, 2004). For R410A in standard rating conditions (EN 12900), energy efficiency has been found to lie between 2.8 and 3.2. Then, on the cooling capacity range of [0-6] kW, they are already as efficient as available scroll compressors and quite cheaper.

Reduced capacity

Because the thermal load required by the building for cooling or heating varies, air conditioners and heat pumps must adapt their cooling capacity to the cooling or heating load. There are two main principles used in practice, on – off cycles of the compressor(s) and reduction of refrigerant mass flow rates.

ON/OFF or cycling
For all types of compressors, the simplest way to adapt the cooling capacity to the required thermal load is to cycle on and off according to the needs. This is generally controlled by a simple thermostat indoor with a set point and a dead band. For instance, if the cooling set point is 25 °C, and dead band is 4 °C, then the compressor will be set on when temperature rises over 27 °C and will be shut off when, after a on period, the temperature reaches 23 °C. Since performance of air conditioners or heat pumps with indoor air temperature is nearly linear, there is no quantifiable loss (or if it can be neglected) linked to overshooting as for boilers. In average, the compressor will work at 25 °C. However, there are two causes of efficiency degradation related to this mode of control that are quantified in paragraph 4.1: variation of efficiency with load ratio.
1) When the unit starts, there is a delay in cooling or heating capacity production while the compressor uses electric power to establish the pressures of the cycle. When the compressor stops, cooling capacity is still produced for a while with no compressor consumption. This results in a loss of energy efficiency that is used to establish refrigerant pressures.
2) When the unit is OFF, consumption is not exactly null; it can remain crankcase heater electric consumption, electronic control consumption and possibly the evaporator fan that can remain on. When integrating energy efficiency on the whole cycle (on phase + off phase), there is a degradation of performance that cannot be neglected at low load conditions.

Reduction of refrigerant mass flow rate
The reduction of refrigerant flow rate enables not only to overcome on-off losses but also to benefit from higher relative heat exchanger areas as compared to the full load and then full mass flow rate in the heat exchangers. Since there is less capacity to extract from the ambient and to reject outside, the temperature differences between the refrigerant and air (both indoor and outdoor) will decrease. This means that the compression ratio will decrease, leading to improved energy efficiency. The total energy efficiency variation will depend on the efficiency of the compressor for this new compression ratio. There are different means used depending on the application and on the cooling capacity range.

Inverter
The frequency of rotation of the compressor is varied by electronic control. In our range of products, this mainly concern single split and multi-split air conditioners with rotary compressors. (Shao, 2004) gives the COP of an inverter driven rotary compressor for different evaporating and condensing temperatures for a range of varied inverter frequencies. This is reported in the figure below. The curve 7/50 is relatively close of standard measurement conditions of EN 12900 (Tev = 5 and Tc = 50). The shape of the curve lets some degrees of freedom for the air conditioner's manufacturer to adapt the maximum efficiency part of the curve either at full load standard conditions EER or elsewhere in the case he would have incitation to have a more efficient air conditioner at low load, low pressure ratio, as could be the case if units were compared on a scale including reduced capacity and reduced outdoor temperature.

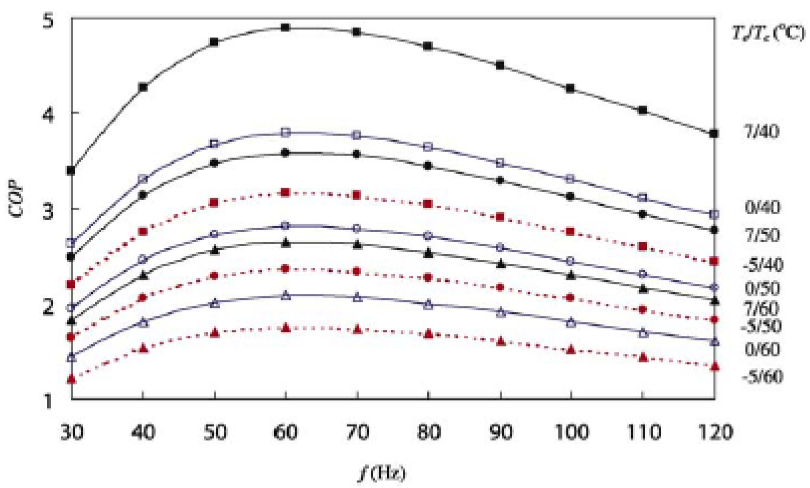


Figure 4-11: COP of performance of an inverter driven rotary compressor as a function of frequency for different couples (evaporating temperature, condensing temperature)

Compressors in parallel
2 or more compressors are used in parallel on the same refrigeration circuit. When load decreases, 1 or several compressors are shut down to reduce refrigerant flow rate and capacity.
Whereas it is generally reserved to larger capacity units than [0-12] kW, it could also be used for smaller capacity units with rotary compressors, since rotary compressors exist until very small capacities (Jang, 2006), it is possible to use two half-sized compressors.

Compressor unloading
For scroll, screw and reciprocating compressors, it is sometimes possible to reduce the swept volume of the compressor by mechanical means. This translates into more or less important refrigeration losses according to the means. It is generally not in use for air conditioners in the scope. For double staged rotary compressors, it is also possible

Digital compressors
This applies to scroll compressors (Copeland Digital Scroll ® compressor and the same principle is used also for rotary compressors by Samsung but to cycle between the two stages of capacity for two cylinders rotary compressors). The general principle is to get very short cycling times to avoid cycling ON-OFF losses of the air conditioning system. For scroll compressors, the compression is stopped by disassembling the two scrolls of the compressor. This is made by hydraulic control. This enables to avoid large off – periods. However, there is not yet any publication on the energy efficiency of this capacity reduction means at reduced capacity. What is certain is that when the two scrolls are not compressing refrigerant, they are still rotating and then compressor still consumes energy. This remaining consumption is not known.

Oil
Compressors in use for products in the scope are hermetic compressors. Oil is needed to ensure proper functioning of the compressor, it forms a protection oil liquid film on moving parts in contact. For R407C and R410A, mineral oils are used. Main properties are viscosity, floc point, dielectric strength, neutralization number, flash point, and fire point. Viscosity is the oils resistance to flow. As the temperature of the oil drops it becomes thicker thereby creating a situation where the oil becomes difficult to pump. The change in viscosity with temperature is measured by the viscosity index. Floc point is a measure of the amount, if any, of wax in the oil. This pertains only to mineral oils. At very low temperature, the refrigerant and oil is likely to form a sluggish and the floc point indicates the minimum temperature to maintain. Dielectric strength is its measure of resistance to electrical current. All oils used with internal electric motors must possess this characteristic. Neutralization number, refers the amount of acid or caustic present in the oil. Flash point and fire point both relate to the oils burning properties and volatility.

Other major properties are lubricity and miscibility and solubility with a particular refrigerant. The lubricity has to do with preventing wear of moving components within the refrigeration system. Hydrodynamic lubrication is defined as the separation of moving parts by a film of oil.
Miscibility is the capability of the two products, oil and refrigerant, to mix in their liquid state. If the two do not dissolve in each other as a liquid then there would be separation and either oil would not come back to the compressor but remain in the receiver (reversible system) and/or would be detrimental to the correct functioning of the expansion device with risks of hunting in case of thermostatic expansion valve. .
Oil continuously cleanse the internal surfaces of the compressor. The oil is the medium used to carry away any microscopic metal particles that may occur from wear. The oil washes the particles down to the compressor sump. There they are suspended in the oil until they eventually drop to the bottom and are picked up and held by magnets that are typically placed at the bottom of the compressor. Refrigeration oil also aids in heat removal. Because it is a liquid it has good heat transfer properties. The heat that the oil absorbs is either rejected in the condenser coil along with the refrigerant or brought down to the oil sump.
When the oil is splashed on the cylinder walls and rings as well as depositing on the suction and discharge valves it helps to seal the gaps of those tolerances thereby decreasing what is known as “blow by” effect. On the opposite side, oil concentration in the refrigerant reduces the overall heat transfer coefficients at the heat exchangers, increases pressure drops. (Stefura, 2006)

Oil charge varies with compressor technology and manufacturers. Orders of magnitude (only indicative) for rotary compressors and scroll compressors range from 0.3 to 0.5 liter / kg of refrigerant fluid.

Compressors is particularly vulnerable when starting. Flooded start occurs when refrigerant is allowed to migrate to the compressor during shutdown. Compressors can be protected with crankcase heaters and automatic pumpdown cycles, where applicable (ASHRAE, 2004). Crankcase heaters are also needed to avoid high viscosity and to go beyond floc point for low temperature application. Temperatures in summer should normally enable air conditioners to operate without crankcase heaters (above 15 – 20 °C) whereas it is needed in winter and then for reversible air conditioners. However, since cooling in commercial buildings can be needed also in spring or even in winter time, some air conditioners may also be equipped of a crankcase heater.

Compressor technologies used in EU air conditioners (in the scope of this study)

For low capacity units (below 6 kW) and multi-split air conditioners, rotary compressor is the dominant type of compressor. For larger capacity units, scroll compressors may be used given that rotary compressors are limited in capacity range. The top runner program (ECCJ, 2006) expects scroll compressors could be one of the main ways to achieve higher COP to increase Japanese air conditioner efficiencies in the near future.

Some R407C units still may use hermetic reciprocating compressor but this type of compressor has now almost disappeared in that capacity range. Only two types of part load control have been identified on the [0-12] kW range of cooling capacity: on-off and inverter control. Whether ON-OFF control still dominates the European market, inverter share is growing rapidly as shown previously in the part of task 2 related to the market trends.

As for fridges, air conditioners, especially reversible air conditioners are likely to be equipped with crankcase heaters. For scroll compressors, values of 70 W in the range 6 to 12 kW have been identified. For rotary compressors, mainly under 6 kW, values of 30 W. These values are in line with values identified in the USA for residential small heat pumps (Max 65 W, average 30 W) by (Floyd, 1998).

4.1.4 Heat exchangers

Heat exchangers enable the refrigerant to exchange heat with indoor and outdoor air (water). Efficient heat exchangers enable to have little temperature difference between air (water) and refrigerant. This efficiency enables to have lower working compression ratio for the compressor and then to decrease energy efficiency. Improving the heat exchange efficiency is then feasible by increasing the heat transfer surface or the heat transfer intensity (means reducing the thermal resistance to heat transfer of the heat exchanger). There are several ways to increase the heat transfer intensity, reduced diameter, increased turbulence of fluids .... One of the limiting factor may be the increased pressure losses that will require more power for forced convection of refrigerant fluids.
Water to refrigerant heat exchangers are more efficient than air to refrigerant heat exchangers because convective heat transfer is higher than for water. This is the main reason for water cooled air conditioners to be more efficient at equal technology of the other components than air cooled air conditioners. The second reason being the difference in test conditions (water cooled air conditioners are tested with lower inlet temperature than air cooled air conditioners).

DX air /refrigerant heat exchangers

Heat exchangers used in air-conditioning applications could be organized in 2 distinct categories: **fin-and-tube heat exchangers with round tubes** (or sometimes oval tubes in order to limit the pressure drops due to the tube bank) and **flat-tube heat exchangers with flat tubes and corrugated fins**. These 2 types of heat exchangers differ of course by their designs, but also by their material (usually copper tubes and aluminum fins for the round-tube configuration and all-aluminum for the flat tube configuration) and by their manufacturing processes. Microchannel heat exchangers are already used for best performing mini-chillers in Europe. This later category enables mainly to reduce the refrigerant charge, or for the same air conditioner size to increase heat transfer performances at condenser side. They are not used in air to air conditioners.

The typical fin-and-tube heat exchangers in air-conditioning application usually consist of mechanically or hydraulically expanded round tubes in a stack of parallel continuous fins as presented in the figure below. Depending on the application, the heat exchanger can be produced with one or more rows. For systems working with azeotropic or nearly azeotropic refrigerants such as R-410A, the trend is to reduce the number of rows to 2 or 3 rows since it is observed that major part of the heat transfer is achieved by the first row. Especially at low air velocities, the heat transfer capability of the downstream rows is limited. For the evaporator of systems working with refrigerants presenting a significant temperature glide, for instance R-407C, it is possible to take advantage of the temperature glide by using a configuration with more than 3 rows (Bigot, 2001 and 2002).

For household applications, the tube diameters may be less than 10 mm, usually 9.52 mm, and recently, even 7 mm and 6.35 mm. This is because higher refrigerant-side heat transfer coefficients (smaller hydraulic diameter) and lower air-side pressure drop can be achieved by using smaller tubes, and this also leads to much more compact fin-and-tube heat exchanger designs. The use of less than 7 mm tube diameters, along with small longitudinal and transverse tube pitches (for instance respectively, 12.3 mm and 21 mm) has become popular in heat exchangers of room air-conditioners. In addition, heat exchangers have gained in compactness with the use of smaller fin pitches (sometimes lower than 1.3 mm) and thinner fins (near 0.1 mm).

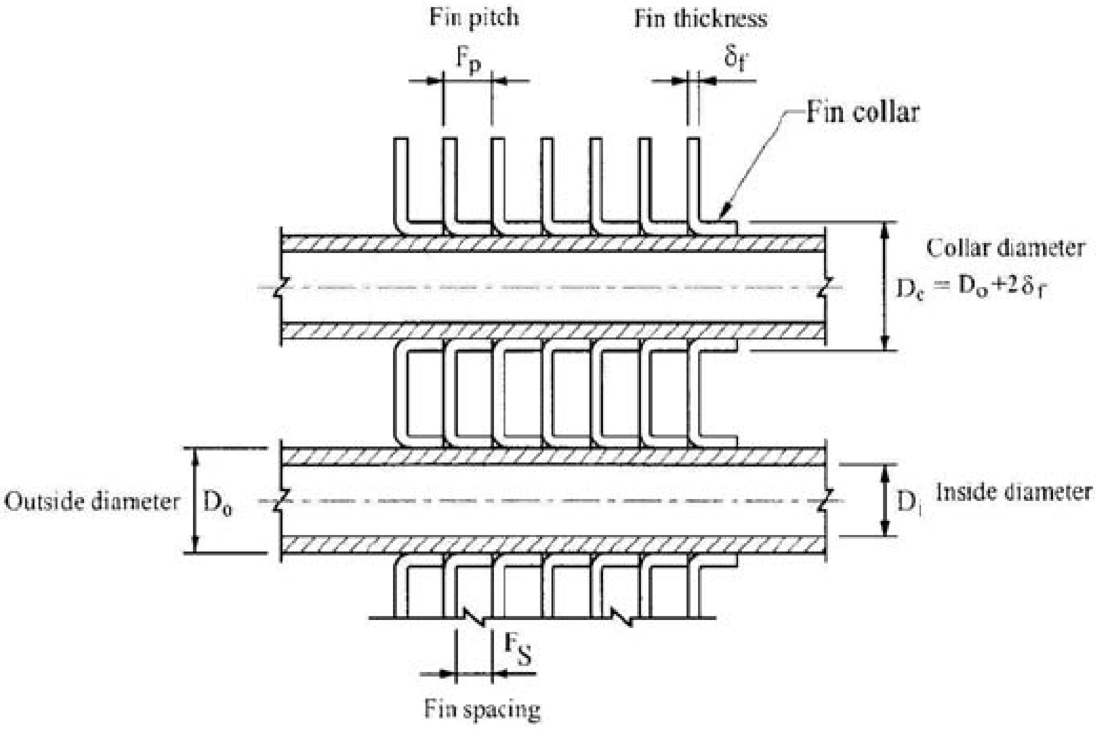


Figure 4-12: Fin-and-tube lay-out and geometrical parameters (WAN, 1999)

Fin types

There are many fin patterns. Three different types have been found in the Eurovent database: louver, flat (plate) and wavy. Flat and wavy fins enable to build plain heat transfer surface whereas louver fins mean that the heat surface is interrupted.

Plain heat transfer surfaces
Plain surfaces denote fins without any surface interruption as presented in Figure 4-13 (a), (b) and (c).

Plate fins
The plate fin configuration in the simplest fin-and-tube design Figure 4-13 (a). In this case, fins are plate aluminum sheets. The plate-fin configuration is still the most popular fin pattern, owing to its simplicity, durability and versatility in application. Moreover it presents superior reliability under long-term operation and lower friction characteristics.

Wavy fins
The wavy fin is a continuous fin with corrugations. The wavy fin geometry is illustrated in Figure 4-13 (b) and (c). Wavy plate fin-and-tube geometries provide a higher heat transfer coefficient than flat fins due to the mixing effect of the corrugated surfaces. There are several variants of the basic wavy fin geometry. Two configurations are usually encountered: the continuous wave curve profile Figure 4-13 (b) and the herringbone wave configuration Figure 4-13 (c). The major difference between the herringbone and smooth wavy fins is the sharp edge corner. Corrugation angle generally ranges from 15◦ to 30◦.

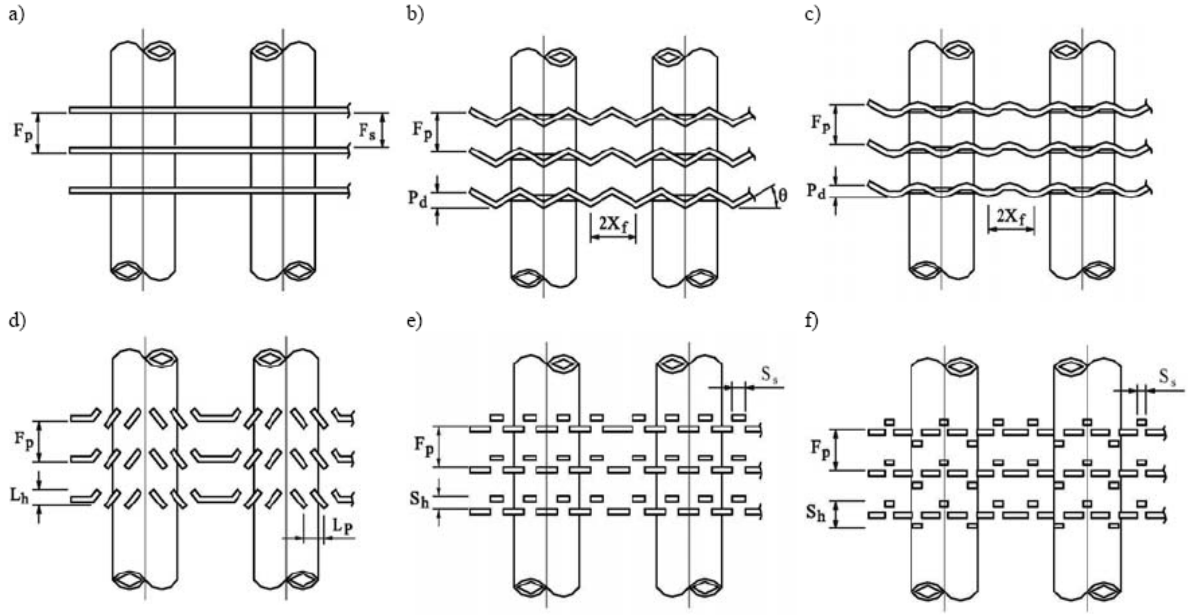


Figure 4-13: Various fin types with round tubes (Wang, 2002); a) Plate fins ; b) Herringbone wavy fins ; c) Smooth wavy fins; Louver fins with redirection louvers ; e) Slit fins ; f) Double side slit fins

Interrupted heat transfer surfaces

Interrupted surfaces denote fins with surface interruptions (louvers or slits) as presented in Figure 4-13 (d), (e) and (f) (but only louver fins have been found in the Eurovent database). In finned-tube heat exchangers, the air-side heat transfer coefficient is largely controlled by the boundary layer growth along the fin. This boundary layer has an insulating effect and the thicker the boundary layer, the more it limits heat transfer to the fin. Moreover, increasing demand for compactness in air-conditioning application influences heat exchanger designs with increased surface density and smaller flow channels. This miniaturization, associated with conventional heat transfer surface designs (plain fins), results in a tendency towards laminar flow and consequently lower air-side heat transfer coefficients.

Louver fins

The typical louver fin configurations are presented in Figure 4-13 (d), Figure 4-14 (a) and Figure 4-19. Louvers are cut and inclined parts of the fin surface and are located perpendicular to the airflow direction. Several louvers are formed in the airflow direction. The louver angle generally ranges from 15◦ to 30◦ and the louver length in the airflow direction, which is also the major louver pitch (*Lp*), could be lower than 1 mm. The louvers provide heat transfer enhancement, even at low Reynolds number. When the Reynolds number is high enough, the louvers direct the flow in an inclined way through the heat exchanger core. Common designs include a redirection louver (Figure 4-19) which tends to reverse the airflow inclination imposed by the first louvers. Louver fins are widely used in aluminum brazed flat-tube-and-fin heat exchanger as presented in figure (Figure 4-14).

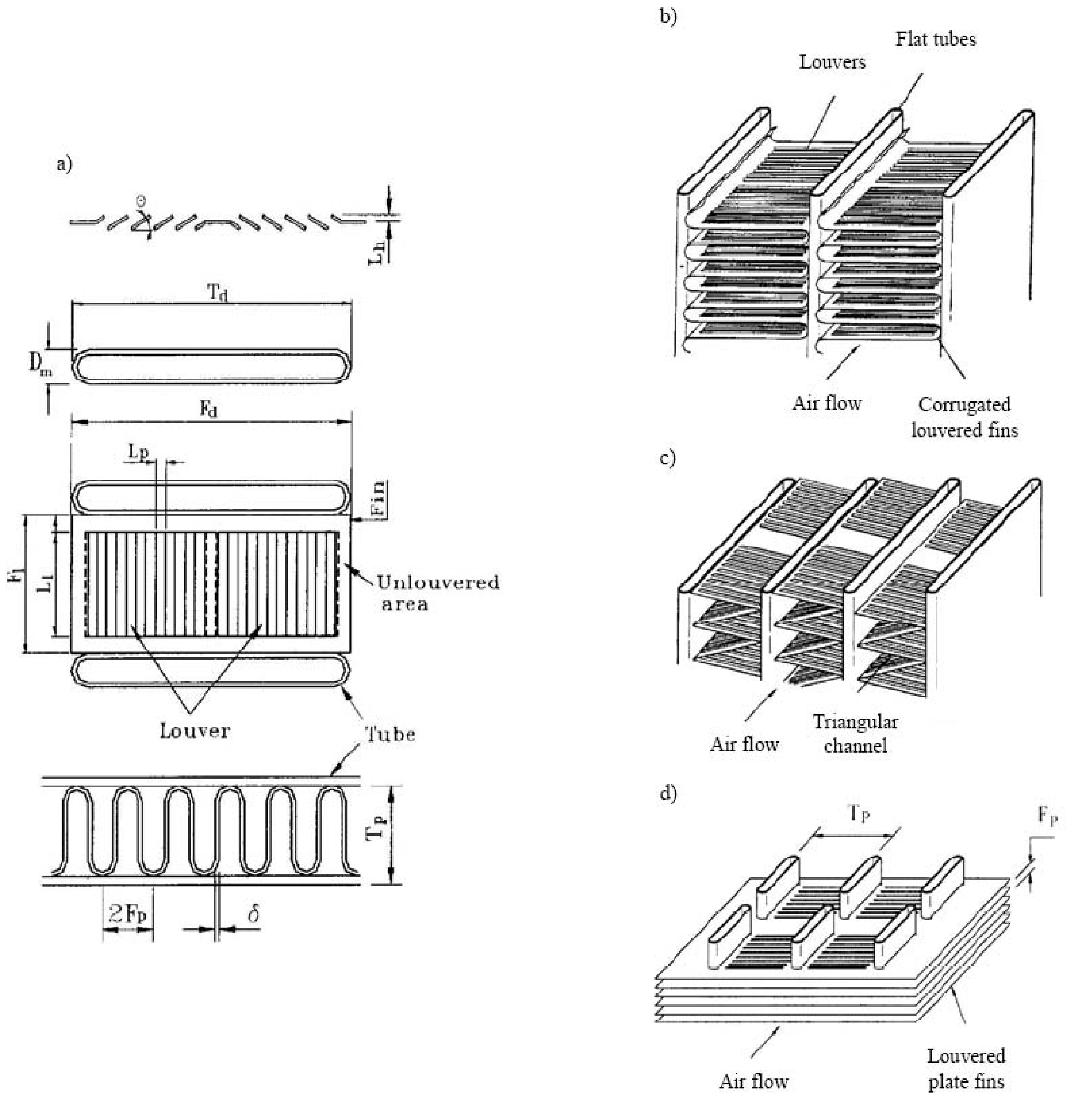


Figure 4-14: Heat exchangers with louver fins and flat tubes; a) Geometrical parameters (Chang, 2000) ; b) Corrugated louvered fins; c) Corrugated fins, triangular channels ; d) Plate fins and flat tubes (Chang, 1997)

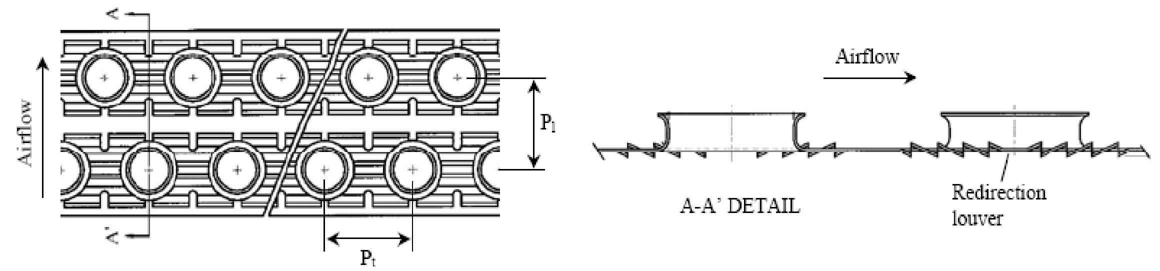


Figure 4-15: Louvered fin-and-tube heat exchanger (Wang, 2000)

Tube types

Two types of tube have been found on the Eurovent database: smooth and grooved ones. Initially, a smooth tube, inner surface of which was not processed like a copper tube, in general, was used for a heat exchanger. For the purpose of saving energy, a tube with internal groove was developed and optimization of a groove shape has been pushed forward (Figure 2.5). As the local heat transfer coefficient on the refrigerant side is much higher than on the air-side, internal groove are shorter in order to achieve a high fin efficiency. The internal surface of the tube is increased by a factor of 1.5 to 3, while on the air-side it is common to find the heat transfer surface area multiplied by 20. Groove on the inner side of tubes are generally straight with a rectangular or trapezoidal shape and are oriented along the tube axis. This is generally obtained by extrusion. The recent trend is to use micro-finned tubes with very small grooves (0.2 mm) in order to enhance the tube-side performance in evaporation or condensation. Tubes could also be helically grooved in one or two directions in order to increase mixing and turbulence.

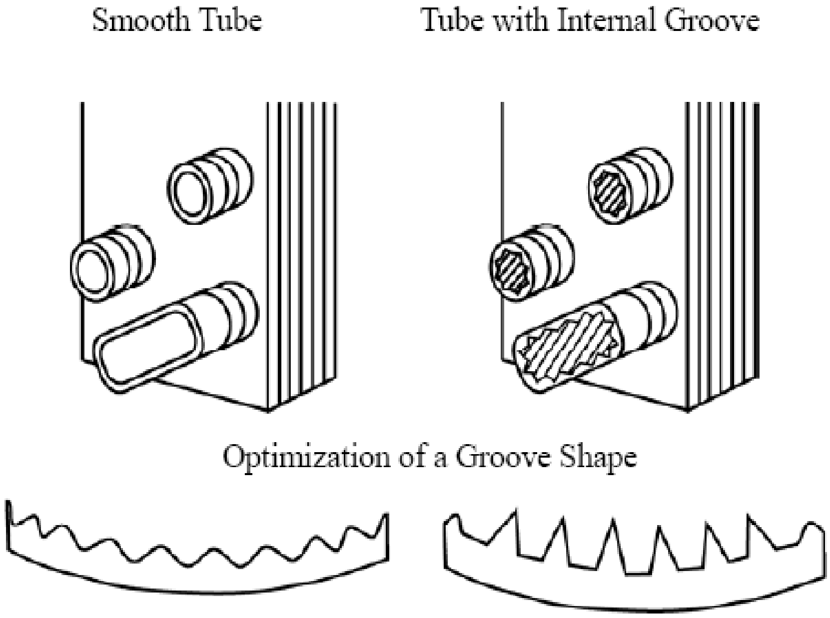


Figure 4-16: Smooth and Grooved tubes (ECCJ, 2006)

Water / refrigerant heat exchangers

For condensers of water cooled air conditioners or for evaporators of mini-chillers, refrigerant to water heat exchangers are used. Mini-chillers, on both evaporator and condenser sides, use plate heat exchangers. It is a plate assembly with circulation of hot and cold fluids alternatively between the plates as shown hereunder. It is also the only remaining type of heat exchanger for water cooled air conditioners.

Figure 4-17: Plate heat exchanger principle, source Alphalaval

Heat exchangers of EU air conditioners

Technical characteristics

For air to air units, copper tube an aluminium fins heat exchangers are used. Main characteristics of tube interior design and fin patterns are reported in the two following figures Figure 4-18 and Figure 4-19 for single split air conditioners and in Figure 4-20 and Figure 4-21 for multi-split units.

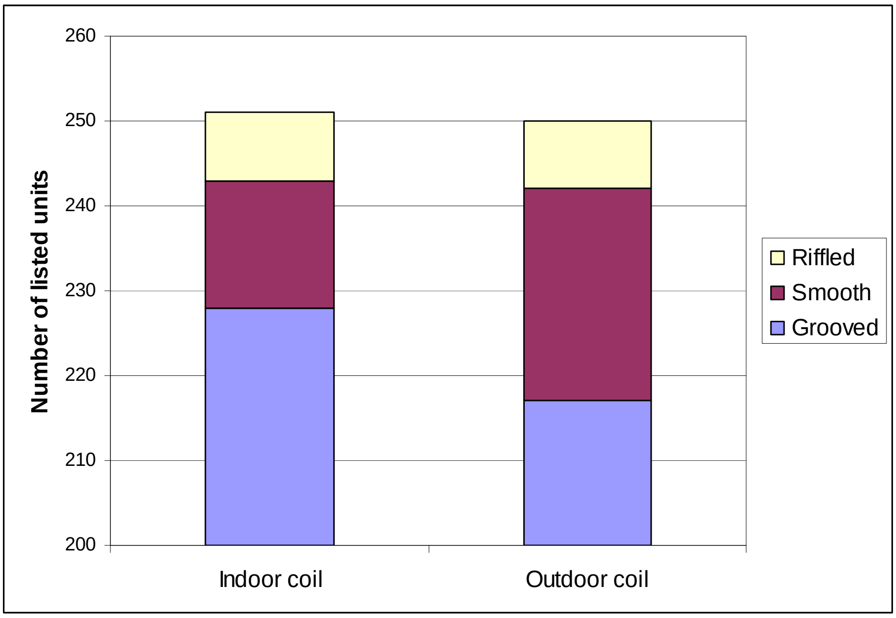


Figure 4-18: type of tube interior design for indoor and outdoor air heat exchangers (Eurovent-tech, 2006) for single split units

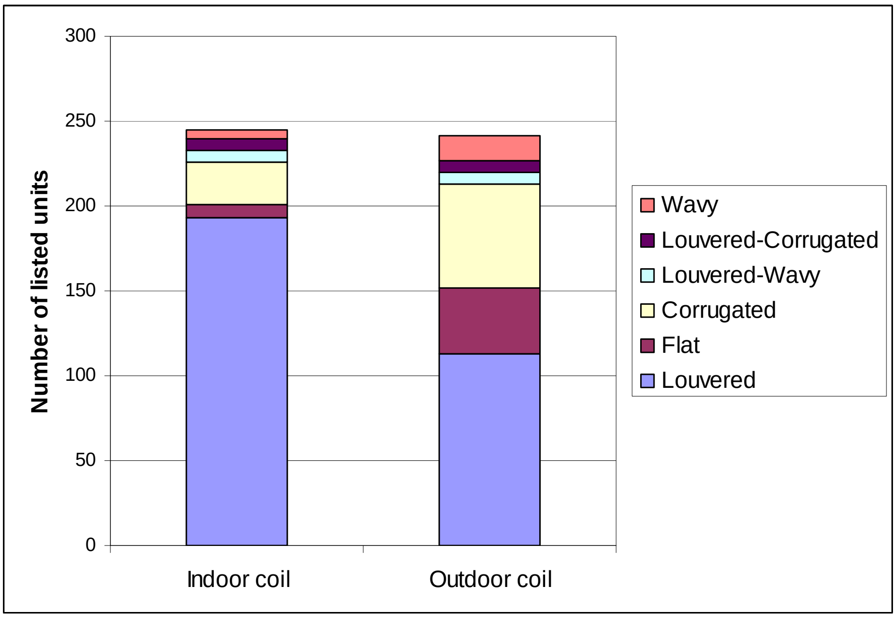


Figure 4-19: type of fin design for indoor and outdoor air heat exchangers (Eurovent-tech, 2006) for single split units

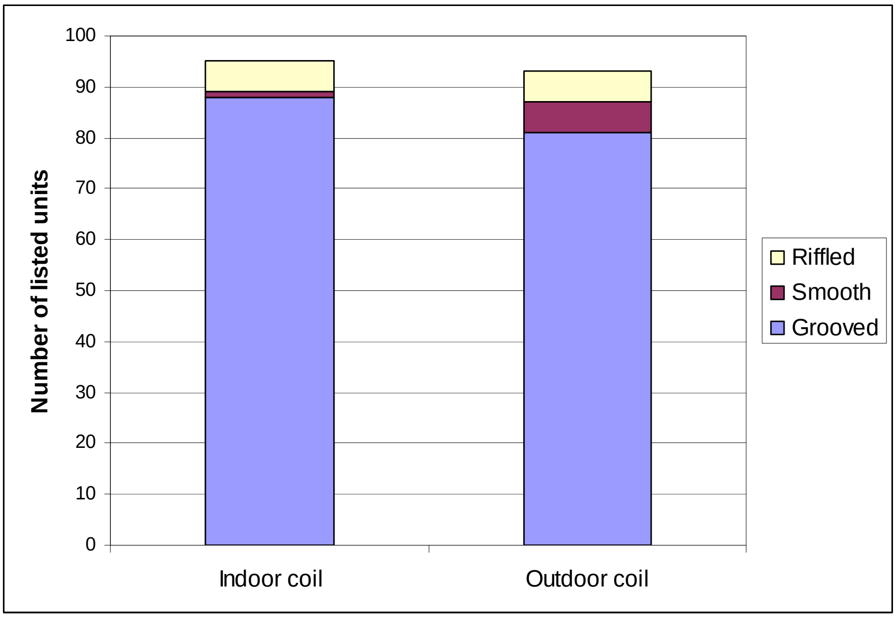


Figure 4-20: type of tube interior design for indoor and outdoor air heat exchangers (Eurovent-tech, 2006) for single split units

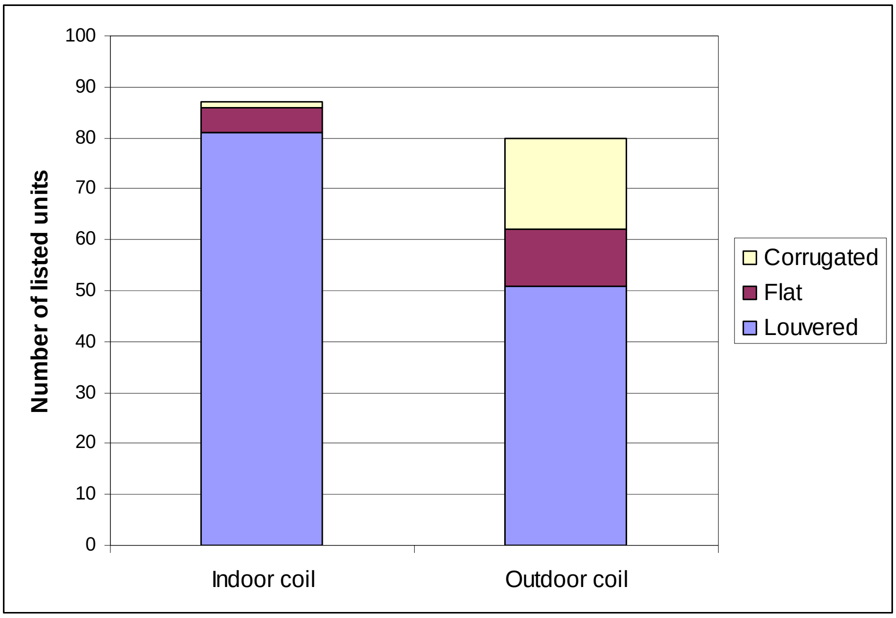


Figure 4-21: type of fin design for indoor and outdoor air heat exchangers (Eurovent-tech, 2006) for multi split units

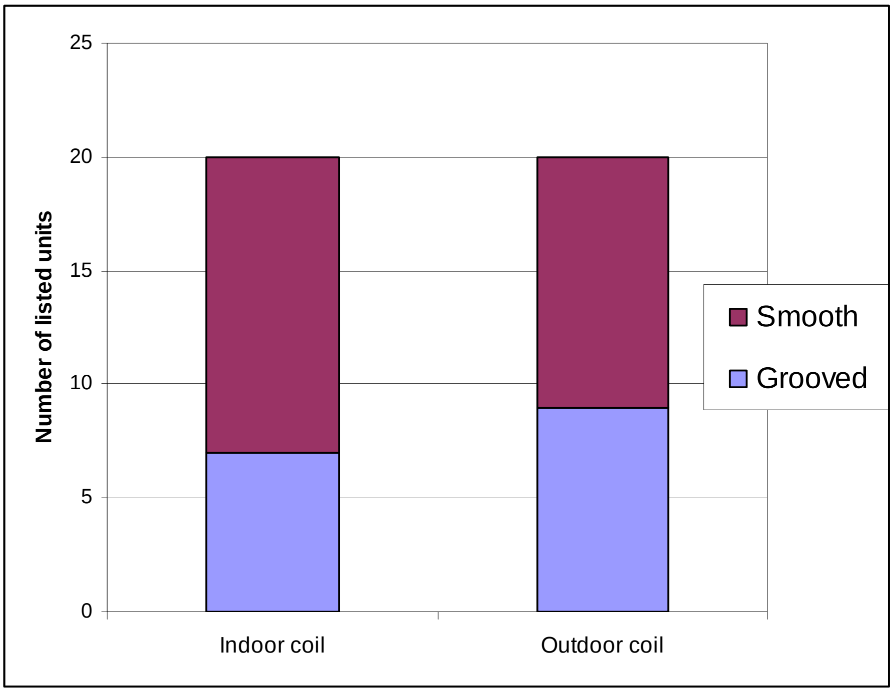


Figure 4-22: type of tube interior design for indoor and outdoor air heat exchangers (Eurovent-tech, 2006) for window/wall air conditioners

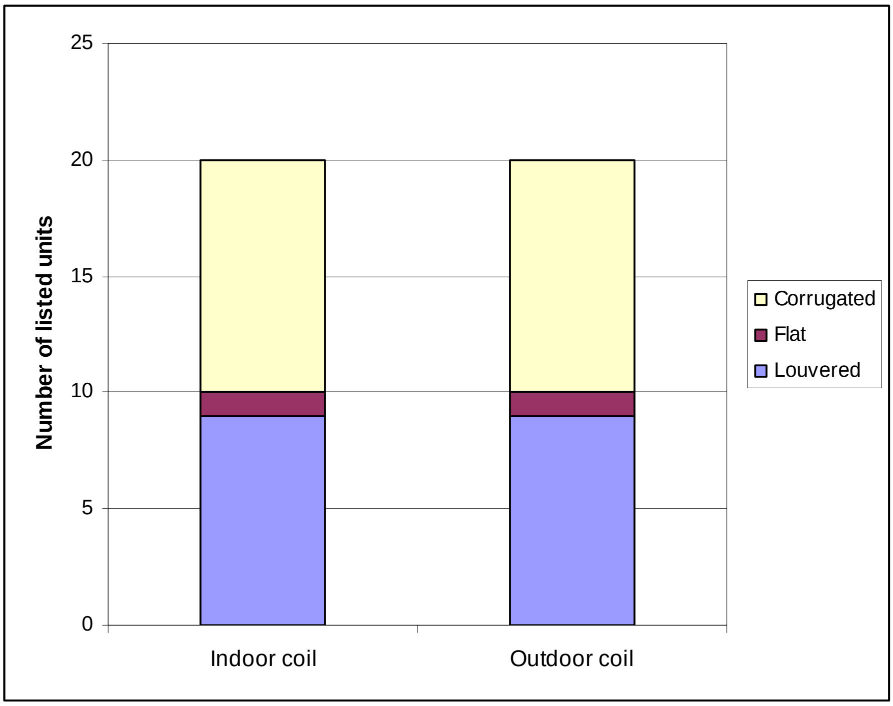


Figure 4-23: type of fin design for indoor and outdoor air heat exchangers (Eurovent-tech, 2006) for window/wall air conditioners

Hence, major design of split and multi-split units are grooved tubes for indoor and outdoor coils and louvered fins. While for window/wall units majority of products have corrugated fin and smooth indoor tubes.

For portable units, no data is available.

Space constraint

Space constraint is a relevant limitation to possible eco-design measures to improve the heat exchange area of indoor and outdoor heat exchangers. (ECCJ, 2006) notes it is an issue in Japan for residential air conditioners because of the standardized space allowed for indoor units installation indoors. This is also particularly relevant for portable units whose size is of primary importance and for window/wall units that are installed mainly in replacement of older units and then whose size should standardized.

4.1.5 Fans

Different types of fans are used in room air conditioners. In indoor units, forced air flow can be achieves with cross-flow fans or centrifugal fans. Axial fans are used in outdoor units. These fans are driven with a variable speed electrical motor or alternatively with several speeds which allows to adjust the air flow rate as desired by the end-user. The technical review hereafter shows that split and multi-split have varied indoor fans according to the type of indoor unit, while outdoor fans are mostly of the propeller (axial) type. Window/wall use primarily centrifugal fans for both sides. Single and double duct are an exception since only one fan is used for both evaporator and condenser coils; it is a centrifugal fans. Mobile split use propeller fan outdoor and centrifugal fan indoor. Air cooled mini-chillers also use propeller fans on the condenser cooling coil.

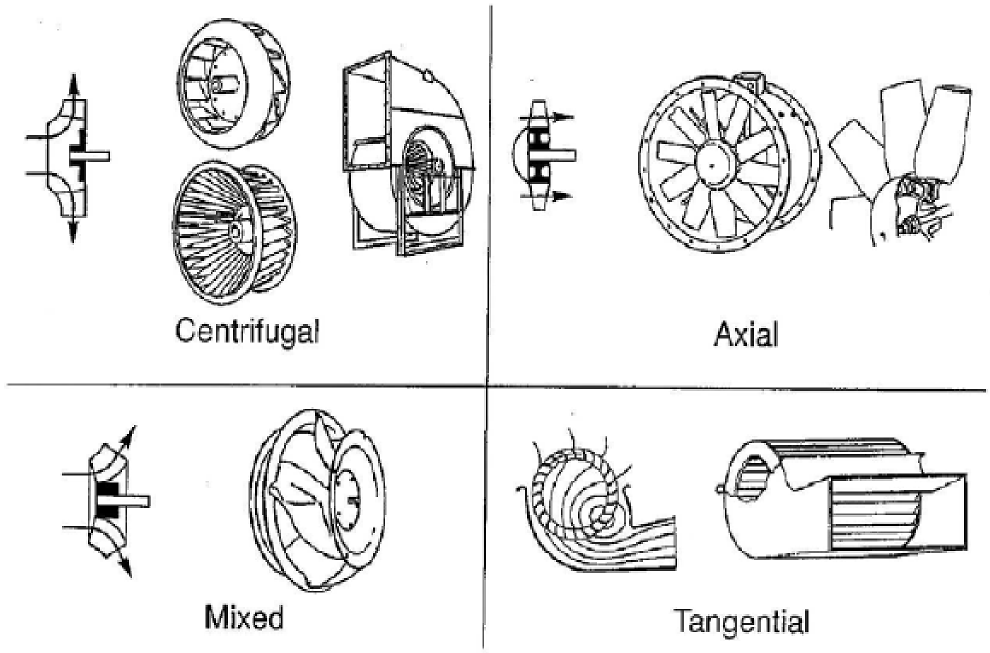


Figure 4-24: Different types of fans (Cory, 1992)

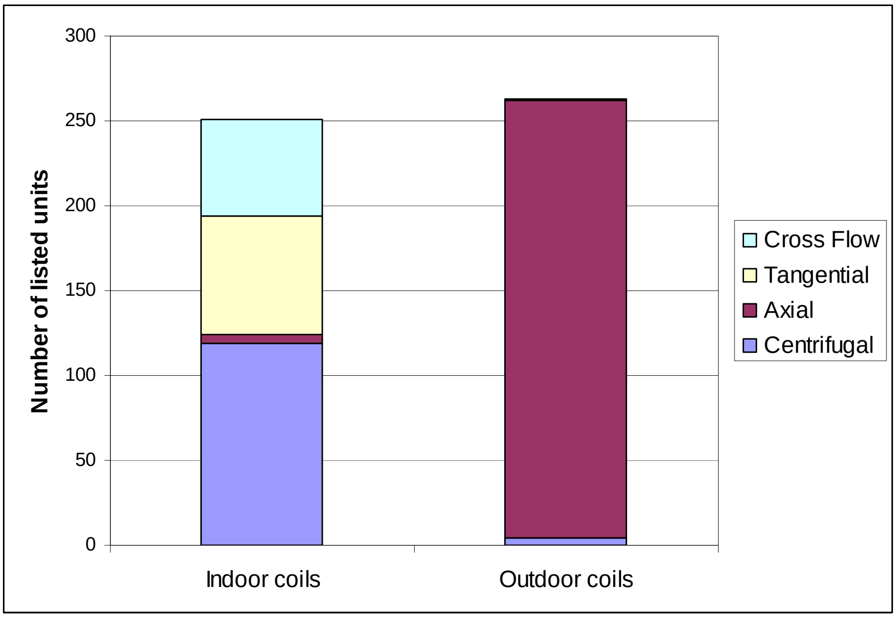


Figure 4-25: Fans used in single split air conditioners

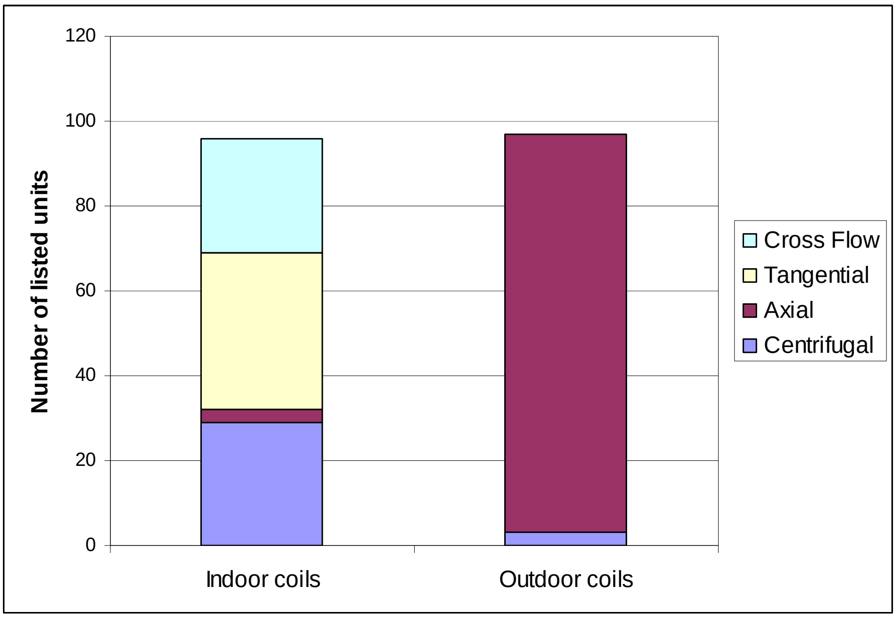


Figure 4-26: Fans used in multi-split air conditioners

Types of fans	Centrifugal	Axial	Tangential
Indoor coils	15	4	0
Outdoor coils	13	5	1

Table 4-3: Fans used in window/wall air conditioners

4.1.6 Expansion device

The expansion valve is placed between the high-pressure side and the low-pressure side of the refrigeration cycle, and its purpose is to maintain the given pressures in these two regions in such a way that condensation and evaporation is carried out under the most convenient circumstances. The expansion valve controls the flow of fluid into the evaporator. Three types of expansion valves have been found in the Eurovent database: capillary, calibrated orifice, thermostatic and electrical expansion valves.

Capillary tubes are small-bore, long tubes used in small cooling systems up to a few kW in size. Tubes, 1 to 2mm in diameter and up to a few meters long are used to produce the high-to-low side pressure drop. As liquid refrigerant passes through the tube, the pressure drops due to friction; the reduced pressure causes evolution of refrigerant gas. As liquid is converted to vapour, the velocity increases; the acceleration causes additional pressure drop. The tube is essentially a passive device and cannot accommodate a large range of load and system pressures. Capillaries have the advantage of being inexpensive and passive. They have the disadvantage of a relatively narrow operating range, susceptibility to clogging by small particles, and the requirement of proper charging within rather narrow limits. They were most widely used because of their low cost and high reliability but control of refrigerant over a wide range of ambient temperature is not optimal. **Short tube orifices**, very popular in automotive application work on the same principles as capillary tubes but the pressure loss is the consequence of a restricted tube passage.

The **thermostatic expansion valve (TEV)** is designed to adjust the refrigerant mass flow rate in order to maintain a constant level of superheat. The TEV is opened when the superheat is too high and closed when the superheat is too low. The TXV is the most popular method of controlling flow and producing the pressure drop in medium capacity systems. The sensor bulb is filled with a small amount of the refrigerant in the system to be controlled. Since the bulb is in close thermal contact with the suction line, the thermodynamic state of the controller fluid represents the state of superheat refrigerant in the evaporator outlet. It is a more complicated system than static expansion devices as capillary tubes and short tube orifices with potential problems as noticed by (Mowris, 2006): TXVs are supposed to optimize refrigerant flow and efficiency as cooling loads vary but in USA, most TXV sensing bulbs are installed with no insulation and improper contact/orientation. In these conditions, hunting may arise (rapid opening and closing of the valve) and harm the compressor while letting liquid enter the compressor in those conditions. If properly used, thermostatic expansion valves enable to maintain superheat in low range of 4 to 7 K for a large set of operating conditions. For low loads, several ports can be used in order to keep flow rate control optimized for all flow rate conditions. Main energy efficiency gains for thermostatic expansion valve is achieved at off-design conditions.

An **electronically controlled expansion valve** enables appropriate degree of throttling based on an electronic signal from a microcomputer determining the operating state of an air conditioner. The valve is such structured that a pulse motor rotates based on an electronic signal, and a gap between the valve and a valve seat is adjusted by converting the rotation into up-and-down motion, thereby controlling the degree of throttling. This could achieve efficient control of the refrigerant flow, depending on the operating state, such as the changing number of revolutions of a compressor used in an inverter air conditioner (ECCJ, 2006). The operating characteristics are close to the thermostatic expansion valve but achieve higher precision and then can enable lower superheat values.

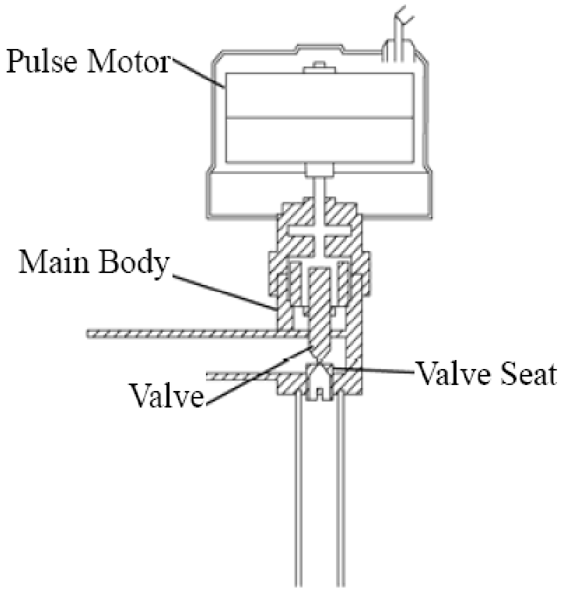


Figure 4-27: Electronic expansion valve (ECCJ, 2006)

Another property of expansion devices is the ability to block refrigerant migration when the unit is OFF or cycling ON-OFF. Only thermostatic expansion valves (with hermetic closure), electronic expansion valves or the use of a dedicated device as a solenoid valve enables to block refrigerant migration.

Expansion devices used in EU air conditioners (in the scope of this study) are reported in the tables below.

	types of expansion valve		
Type of air conditioner	**Capillary**	**Orifice**	**TEV or EEV**
Single split air conditioners	159	17	73
Multi split air conditioners	43	6	41
Window/wall air conditioners	6	12	1

Table 4-4: Expansion device types of single split, multi-split and window/wall air conditioners

4.1.7 Reversibility

A **four-way valve** is located at the exit of the compressor and is used to switch the operating mode (cooling or heating) by changing the direction of the refrigerant flow.
This valve is linked to four distinct and separate flow paths:

- compressor suction (compressor inlet)
- compressor flow back (compressor outlet)
- exchanger of the inside unit
- exchanger of the outside unit

Inside this valve body, a mobile part can move from a bottom to the other enabling two possible configurations:

- The refrigerant can go from the compressor outlet to the outside coil or from the inside coil to the compressor inlet
- The refrigerant can go from the outside coil to the compressor inlet or from the compressor outlet to the inside coil

In the first configuration, the indoor exchanger works as evaporator and the air conditioner operates in the cooling mode. In the second configuration, the indoor exchanger works as a condenser and the air conditioner operates in the heating mode.

Because of the difference of volume content between indoor and outdoor heat exchanger, a **receiver tank** may be needed to compensate the volume difference of refrigerant between cooling and heating modes for reversible air conditioners. It is then generally located at the outlet of the condenser in heating mode and not used in the cooling mode.

In heating mode, the outdoor coil is the evaporator side and then refrigerant evaporating temperature is lower than the outdoor temperature by a few °K. Depending on the characteristics of the heat exchanger and the operating conditions, when the outdoor temperature drops below 3 to 7 °C, the refrigerant temperature in the evaporator becomes negative. At about -3 °C, the evaporator surface becomes also negative and **frost** begins to grow on the outdoor coil (more or less rapidly according to the content of humidity in the air at this particular moment around the air coil). If the temperature is maintained below 0 °C, the frost accumulates over the coil until it blocks completely the air flow rate. **Defrosting** is then necessary. Several solutions may be used (Argaud, 2001). The more popular solution today is to reverse the 4 way valve thus releasing hot gas in outdoor coil for 1 to 3 minutes; the time needed for the temperature of the coil to pass over 0 °C. By the past, and there may remain some in use, fix clock defrost cycles were started when outdoor temperature was below a certain threshold. This was by the past a problem with heat pumps because when there was no frost besides outdoor temperature below the threshold, reversing the cycle may increase importantly the high pressure until the compressor cut out for safety reason. Today, control is actuated by measuring the surface temperature of the outdoor coil or the refrigerant temperature inside the coil.

Concerning design, reversible air conditioners performances are more difficult to increase for both modes and some measures that would improve energy efficiency in the heating mode would degrade the performance in the heating mode. For instance, increasing the fin density to increase cooling performance would be likely to decrease the heating performances in frost conditions.

As seen before, reversible split and multi-split air conditioners are real heat pumps in heating modes. On the contrary, portable air conditioners may be either simple electric heaters or heat pumps. Frosting is an issue for real reversible cycle portable units.
Mobile split are mostly real reversible units. Nevertheless, performances may be low at low outdoor conditions. For instance, some manufacturers of these units explain in the product documentation that the outdoor unit can be put indoors or outdoors indifferently meaning that the COP for low outdoor conditions may be close to 1.
Single duct are mostly reversible with the addition of an electric heater. But some units are advertised as real heat pumps. Coming back to the principle of the single duct in heating mode, condenser is in contact with air of the room for instance at 20 °C. As consequence, there is little risk to have frost

accumulation indoors. However, whether starting heating in a cold room, there may be problems with frost indoor. But no information is available on this subject. The same rationale applies to double duct units. Some fix double duct units are specifically advertised for heating purposes with the risk to freeze the wall with correlated construction problems.

4.1.8 Control

Air conditioners include a large number of controls managed by a micro computer as all modern appliances. Classical safety controls include high pressure cutback to avoid damaging the compressor in case of operating with too low leak or in too high temperature conditions outdoor, some have also low pressure side control. All compressors have a high temperature control. In case of electronic expansion valve, supplementary control are necessary, as well as with inverter driven compressors and fans. In case cooling is necessary with low outdoor temperature, compressor manufacturers may advise to protect their compressor and not to operate at too low difference between high and low pressure side. In that case, a specific control reduces the outdoor flow rate (that may be air or water) to increase the high pressure and accordingly energy efficiency is reduced.

Air to air conditioners also propose a dehumidification mode. To that effect, the evaporator refrigerant side can include a supplementary controlled expansion valve to reduce the evaporator pressure and increase dehumidification capacity while keeping ability to control temperature. In case, this is not enough, direct action on the main expansion valve in order to lower the pressure in the whole evaporator may be needed.

As energy efficiency features, some units also control the air flow passage of the indoor unit by measuring the pressure loss to warn end-users when it is necessary to clean the filter. Some units also use sophisticated sensors to detect the presence of persons in the room to be cooled or heated and to direct the airflow in their direction. In correlation, a mode sometimes called “Economic” can be associated: when nobody is in the room, the cooling or heating operation is stopped.
For water cooled air conditioners operated in open loop, some manufacturers propose to vary the flow rate with load to avoid to waste too much water. As a counterpart, energy efficiency is reduced by operation at higher pressure ratio.

For mini-chillers, they are normally operated at constant chilled water flow rate. Manufacturers do propose nowadays mini-chillers with integrated pumps and several speeds with control integrated to optimize the pump speed as a function of the operating conditions.

4.1.9 Other parts

Air conditioners may be equipped with small pumps that recover the condensate outside or on the water network of waste water.

4.1.10 Noise

As explained in task 3 on Consumer Behaviour and Local Infrastructure, noise has became a real issue for air conditioners both indoor and outdoor. Noise design choices concern primarily fan and air circuit (inlet / outlet) and compressor (either for indoor or outdoor noise) but also the expansion valve. It means trade-offs between higher efficient fans and compressors and also supplementary pieces of furniture to insulate or isolate the compressor or the expansion valve.
Even for reversible units, generally a single value for sound power is given. Very few manufacturers give both heating and cooling sound information. For some of them, it can be in terms of pressure level only. As a consequence, the graph below addresses sound power in cooling mode only. In heating mode however, values identified for some manufacturers are very similar (+/- 2 dBA).

The different graphs below show that noise intensities (sound power) are comparable for all types of units except the window/wall (that includes air to air packages below 12 kW) that shows higher indoor and outdoor noise intensity. For single duct units and mobile split units, values are very low while the compressor is located indoor and there is a centrifugal fan indoor. Some values are certainly underestimated.

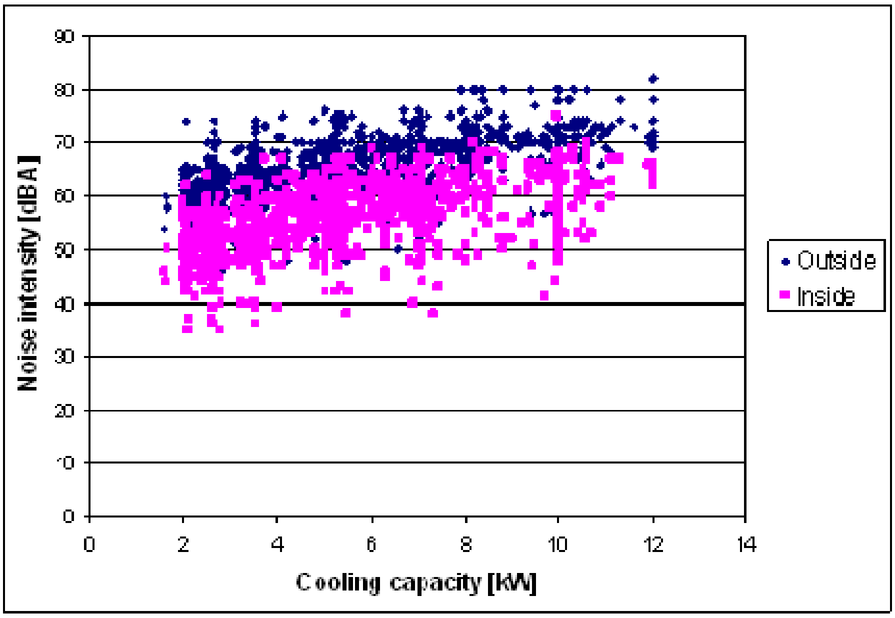


Figure 4-28: indoor and outdoor noise of single split air conditioners

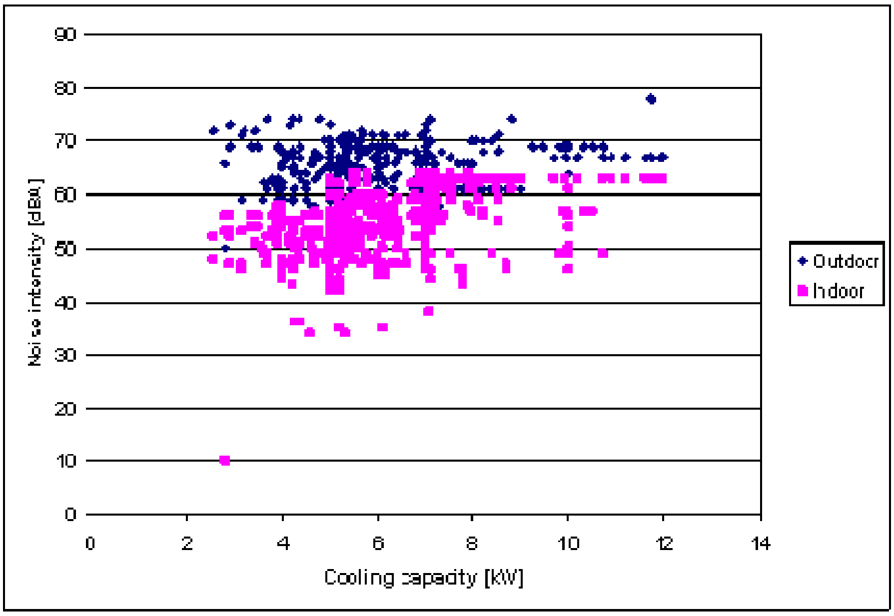


Figure 4-29: indoor and outdoor noise of multi split air conditioners

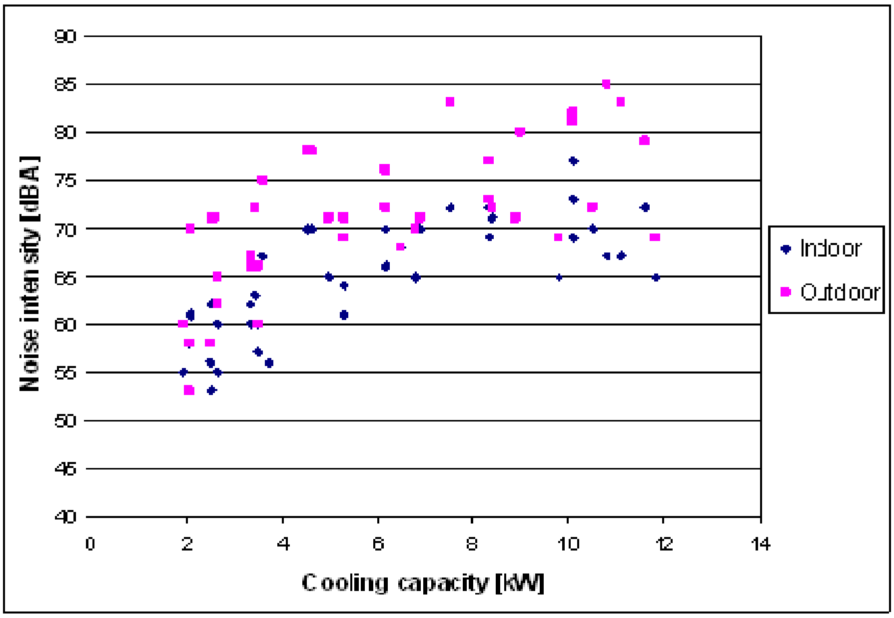


Figure 4-30: indoor and outdoor noise of window/wall air conditioners

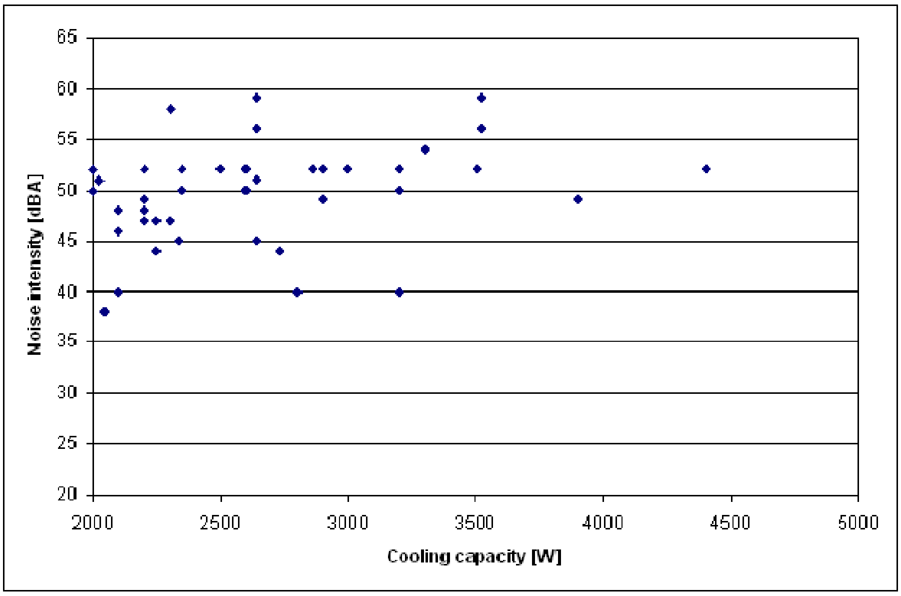


Figure 4-31: noise of single duct air conditioners

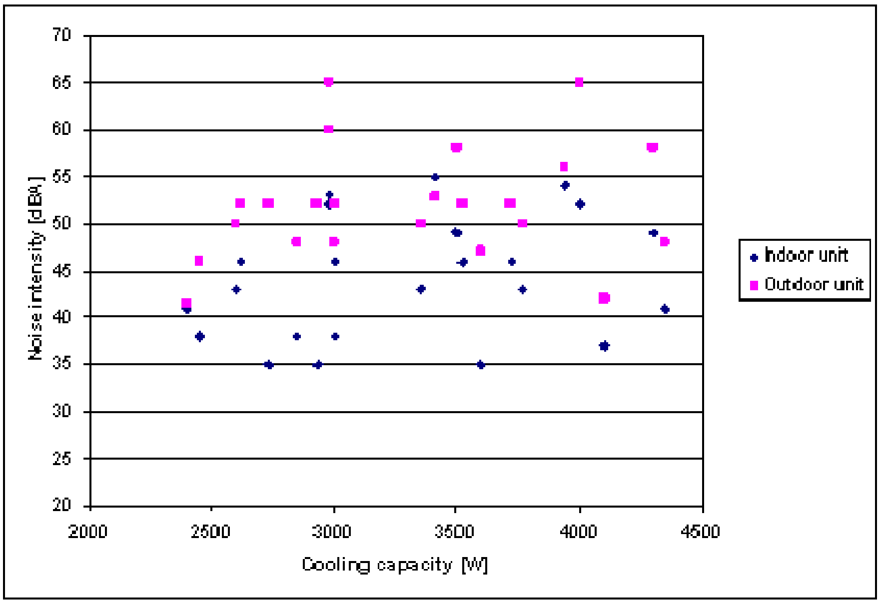


Figure 4-32: noise of mobile split air conditioners

We have not found direct link between power noise and energy efficiency on the dabase of Eurovent products. It appears that most efficient products are not more noisy that less efficient ones despite when designing a unit, one of the primary parameters, air flow, makes a difference regarding energy efficiency.

This is illustrated on the figures below for reversible split units.

Figure 4-33: noise intensity of reversible single split air conditioners – Pc < 6 kW

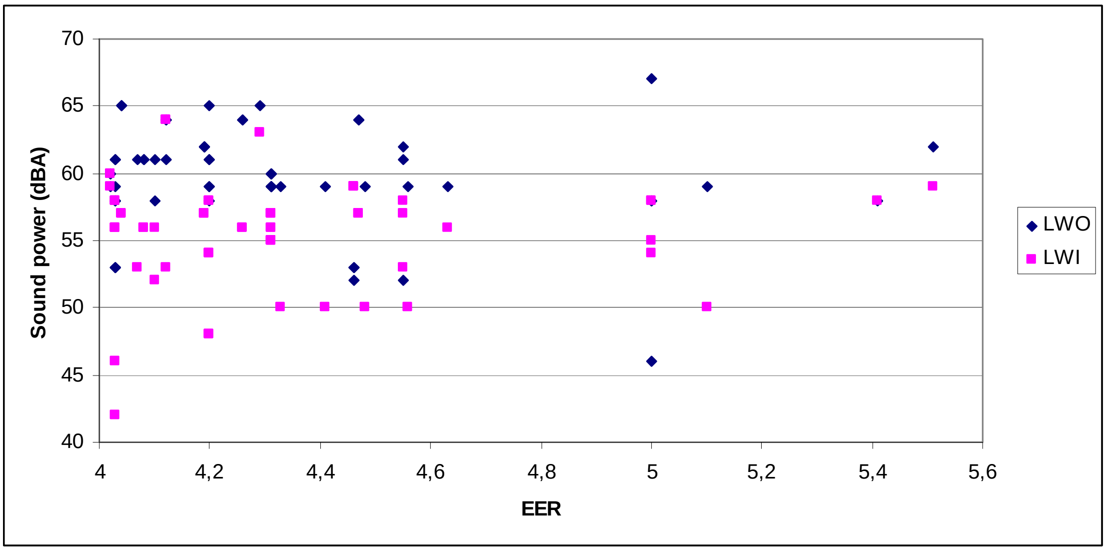


Figure 4-34: noise intensity of reversible single split air conditioners – Pc > 6 kW

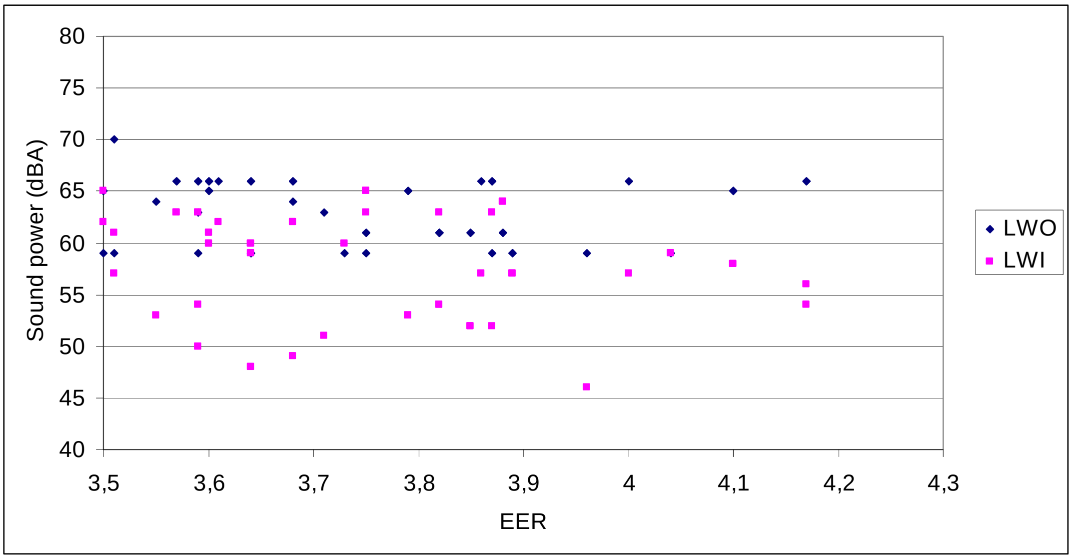
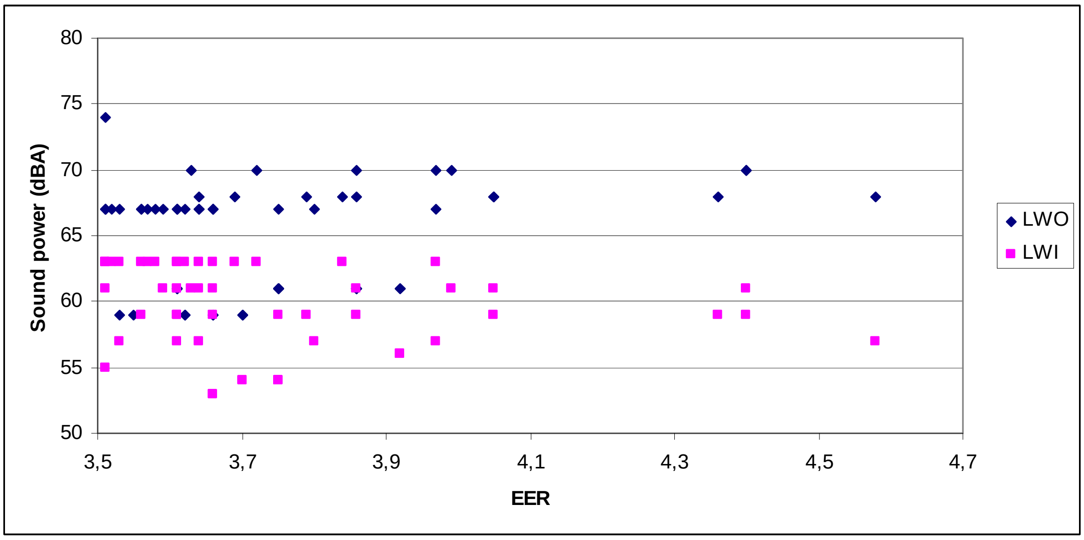


Figure 4-35: noise intensity of reversible multi split air conditioners – Pc > 6 kW



4.2 Production phase

Preliminary Analysis

We currently have 32 Bills of Materials. This preliminary analysis deals with reversible mono-split units, since the majority of the Bills relate to these products. Most of them have cooling capacities between 2.8 and 4 kW.

They are classified by the manufacturers as "typical" or "best" models: some are unclassified. As can be seen in the chart, the relative of material content varies between products, but the variation within any one class is greater than the difference between classes. We therefore propose to combine all three classes for further analysis.

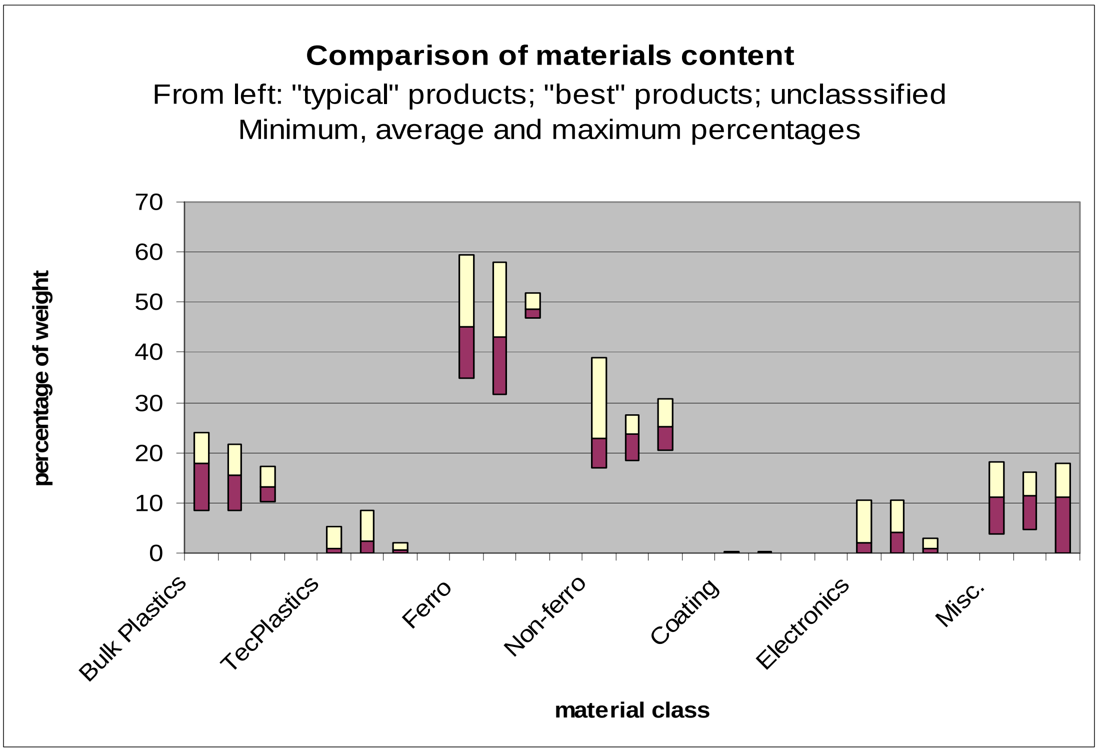


Figure 4-36: Material content for single split units in percentage

There does not appear to be any systematic change in composition with cooling capacity, although units of around 4 kW cooling capacity appear to be lighter than those of lower or higher capacity.

The **average total weight is 14 kg/kW**. This is consistent with the figures of **10kg/kW for outdoor units, and 4 kg/kW for indoor units** derived from the Eurovent database (Eurovent-tech, 2006).

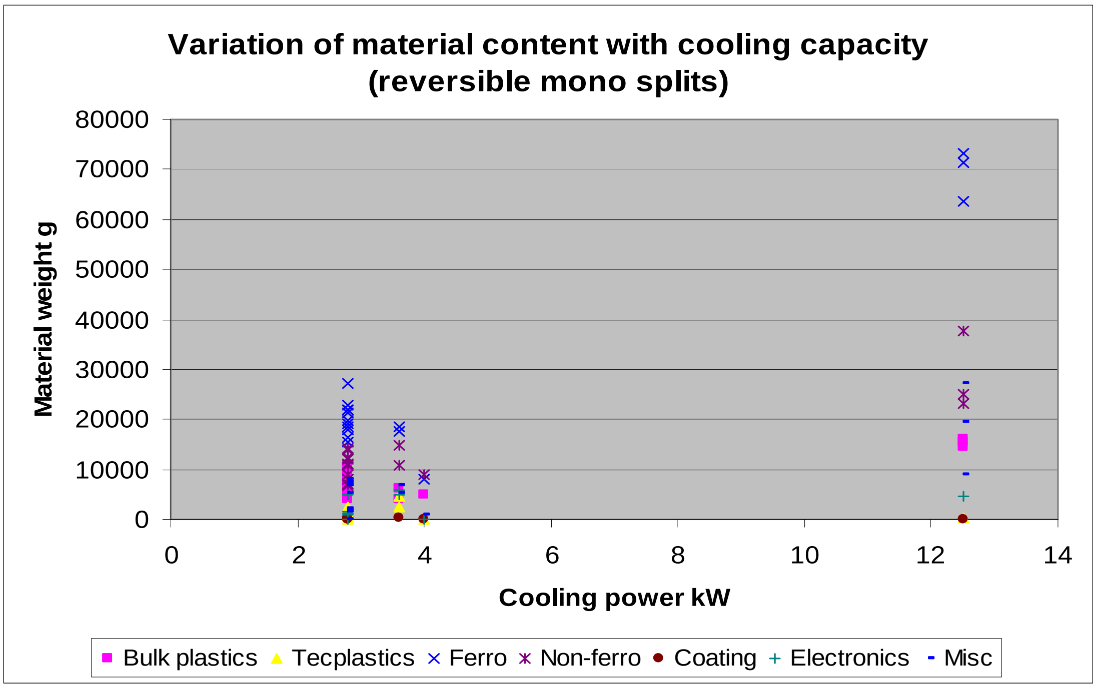


Figure 4-37: Variation of material content with cooling capacity for reversible single split units

We therefore propose an initial estimate of the material breakdown as shown in the "average %" column below. For information, the table shows the modal values for mono-splits and average figures for the four multi-split systems for which we have data.

Material Type	Monosplit		Multi-split % (4 units only)
	Average %	Modal %	
Bulk Plastics	16	(14)	13
TecPlastics	2	(0)	0
Ferro	45	(47)	57
Non-ferro	24	(25)	23
Coating	0	(0)	0
Electronics	3	(<1)	2
Misc.	11	(13)	6

Table 4-5: Average material content of split and multi-split air conditioners

We may refine these figures and address other product ranges and product types whether more data is made available concerning:
- larger split units [6-12 kW],
- multi-split units [0-12 kW],
- portable appliances (single and double duct units and mobile split units),
- window/wall air conditioners,
- mini-chillers.

4.3 Distribution phase

A default value of 0.25 m3 per packaged product for 3.5 kW single split and moveable units is kept.

4.4 Use phase product

4.4.1 Energy use

The cooling (heating) capacity and energy efficiency ratio - EER (COP in heating mode) vary with indoor and outdoor conditions and load ratio. In order to make product performances comparable, standard (also called rating) performances are defined by manufacturers. Nevertheless, the conditions in real life are never the standard conditions and then performances vary importantly along the year.

Standard and off-design performance are then described as follows:
- performances in standard conditions,
- performance variation with indoor and outdoor conditions,
- performance variation with load ratio.

4.4.1.1 Standard energy efficiency

All types of air conditioners in scope have similar conditions for similar combination of fluids at their heat exchangers (air / air …) except single duct air conditioners.

	Outdoor heat exchanger		**Indoor heat exchanger**	
AIR TO AIR // COOLING	**Inlet dry bulb temperature °C**	**Inlet wet bulb temperature °C**	**Inlet dry bulb temperature °C**	**Inlet wet bulb temperature °C**
Split, multi-split, window/wall, double duct	35	24 [a]	27	19
Single duct[b] _test labelling	35	24	35	24
Single duct[c] _test rating	27	19	27	19
AIR TO WATER // COOLING	**Inlet dry bulb temperature °C**	**Inlet wet bulb temperature °C**	**Inlet temperature °C**	**Outlet temperature °C**
Mini-chillers air to water	35	24	12	7
WATER TO AIR // COOLING	**Inlet temperature °C**	**Outlet temperature °C**	**Inlet dry bulb temperature °C**	**Inlet wet bulb temperature °C**
Water to air (split, multi-split and window/wall)	30	35	27	19
WATER TO WATER // COOLING	**Inlet temperature °C**	**Outlet temperature °C**	**Inlet temperature °C**	**Outlet temperature °C**
Mini-chillers air to water	30	35	12	7
AIR TO AIR // HEATING [d]	**Inlet dry bulb temperature °C**	**Inlet wet bulb temperature °C**	**Inlet dry bulb temperature °C**	**Inlet wet bulb temperature °C**
Split, multi-split, window/wall, double duct	7	6	20	15 max

[a] The wet bulb temperature condition is not required when testing units which do not evaporate condensate.
[b] When using the calorimeter room method, pressure equilibrium between indoor and outdoor compartments shall be obtained by introducing into indoor compartment, air at the same rating temperature conditions.
[c] Standard EN 14511 presently indicates labelling should be based on the 35/24 test point and rated capacity on the 27/19 test point.
[d] Reversible single duct are not covered by the EN 14511 standard but are covered by the labelling directive.

Figure 4-38: Standard test conditions for air conditioners (EN 14511, 2004)

In the table below, we have gathered from varied sources (Eurovent, 2006), manufacturers' catalogues, importers' websites … declared performances as in 2006. Analysis of the values in the table is then led by technical product type.

Technical type		Cooling capacity			EER			Heating capacity			COP		
		Min	Max	Ave	Min	Max	Ave	Min	Max	Ave	Min	Max	Ave
Single split (<6kW)	Cool. only	1.6	6	3.99	2.21	4.08	2.81						
	Reversible	1.64	6	3.85	2.21	5.51	3.06	1.64	7.2	4.41	2.42	5.71	3.31
Single split [6-12kW]	Cool. only	6.07	12	8.17	2.1	3.24	2.58						
	Reversible	6.05	12	8.23	1.85	4.17	2.73	6	14.4	9.09	2.25	4.48	3.06
Multi split (<6kW)	Cool. only	2.55	6	4.86	2.21	4.21	3.05						
	Reversible	2.8	6	5.03	2.25	4.21	3.12	3.1	9.26	6.25	2.45	4.63	3.38
Multi split [6-12kW]	Cool. only	6.04	10.7	7.5	2.21	3.26	2.75						
	Reversible	6.1	12	7.86	2.26	4.58	3.02	6.15	13.9	9.30	2.45	4.48	3.44
Window/ wall	Cool. only	2.05	11.8	4.64	2.15	2.99	2.47						
Chillers	Cool. only/Air cooled	3.66	11.7	7.90	1.89	3.01	2.30						
	Cool. only/Water cooled	4.5	11.5	7.86	3.06	4.32	3.91						
Single duct		1.8	4.4	2.67	1.81	4.12	2.67	1.6	4.4	2.71	2.63	3.96	3.2
Double duct		1.87	3.52	2.47	2.21	2.67	2.44	1.35	3	2.1	1	2.67	2.16
Mobile split air conditioner		2.35	4.35	3.25	2	3.73	2.87	1.6	4.22	2.97	1	3.67	2.28

Table 4-6: Standard performances for air conditioners, miscellaneous sources

Single and multi-split air conditioners
The source of data is the Eurovent on line catalogue (Eurovent, 2006). Average EER of single split air conditioners is higher for the [0-6] kW cooling capacity range and reversible products – 3.06 - than for non reversible products – 2.81- and than for more than 6 kW units –cooling only: 2.58 and reversible 2.73. The same trend can be observed for multi-split units with slightly higher EER values. This tendency is linked to the high proportion of highly efficient Asian products, that are mainly in the lower capacity range and reversible. Interestingly, split and multi-split units beyond 6 kW exhibit the same average EER. For reversible units in heating mode, smaller single split units are also more efficient, 3.31 against 3.06. Multi-split units are more efficient than single split units and larger than 6 kW cooling capacity units are more efficient than smaller than 6 kW units - < 6 kW COP = 3.38, and > 6 kW COP = 3.44.
The split and multi-split reversible units present the larger discrepancies between lower and higher energy efficiency ratios and then potential for energy efficiency improvement.
The average values may be underestimated because of the larger sales of lower capacity and higher efficient products. Sales weighted average values are being gathered by Eurovent for split and multi-split air conditioners [0-12] kW by capacity class.

Portable air conditioners: single duct and double duct, mobile split
The dominant portable type (See task 2, market analysis) is the single duct type. Average capacity of the market analysis has been identified to be around 2.7 kW which is in good agreement with products identified in the manufacturers and importers brochures. Average efficiency is 2.67. For only two units advertised as reversible, heating capacities are of the same order of magnitude, around 3 kW, and COP

superior to 3. Apparently, the potential for improvement is high. But in line with comments in the consumer survey led in (Which, 2007), some declared values are doubtful.
As required, double duct manufacturers take benefit of the modified labeling scale of the Directive 31/2002/EC, the same one as for split and multi-split less 0.4.

Window/Wall air conditioners
Average EER are lower than for split air conditioners with small differences between units: energy efficiency ratios lie between class B and F of the labeling directive 31/2002/EC, with grade D as the average.

Mini-chillers
Cooling capacity and energy efficiency are reported in the table below. Results are extracted from the Eurovent directory 2006. Cooling only mini-chillers represent 15 % in number of models of the 0-12 kW range. In the Eurovent Directory 2007, ESEER seasonal performance values can also be found. This point will be discussed with seasonal performance indicator afterwards in this chapter.

Water cooled air conditioners
Concerning water cooled split, multi-split air conditioners and package air conditioners, about 20 products have been identified on the EU market. Average EER is 3.5 and EER extends to EU energy label grades F to A, one model having a 4.59 EER, above grade A, with grade D as the average. The same results is observed for the heating mode.

Standard EER and COP values

For what follows, energy consumptions are calculated for the base cases as identified in task 5, with corresponding EER and COP values, that are reported hereafter.

Base case	Cooling capacity kW	Heating capacity kW	EER	COP
Moveable	2.2		2.3	
Split cooling only	3.5		2.9	
Split cooling only	7.1		2.5	
Split reversible	3.5	4	3.1	3.4
Split reversible	7.1	8.1	2.8	3.3

Table 4-7: Default capacity and efficiency values of base cases, as determined in task 5

4.4.1.2 Variation of performances with source conditions

Variation of cooling performances with outdoor and indoor source conditions

Generic behavior of air conditioning systems is: when outdoor temperature decreases, cooling capacity and energy efficiency ratio (within certain limits) increase. When indoor temperature increases, the cooling capacity and the energy efficiency ratio decrease. Such data are rarely supplied by manufacturers. Default models for simulation tools are generally set up with existing manufacturer information for average efficient models, as the one extracted from (Consoclim, 2004). Indoor air flow rate is set constant as well as inlet air humidity ratio (50 %).

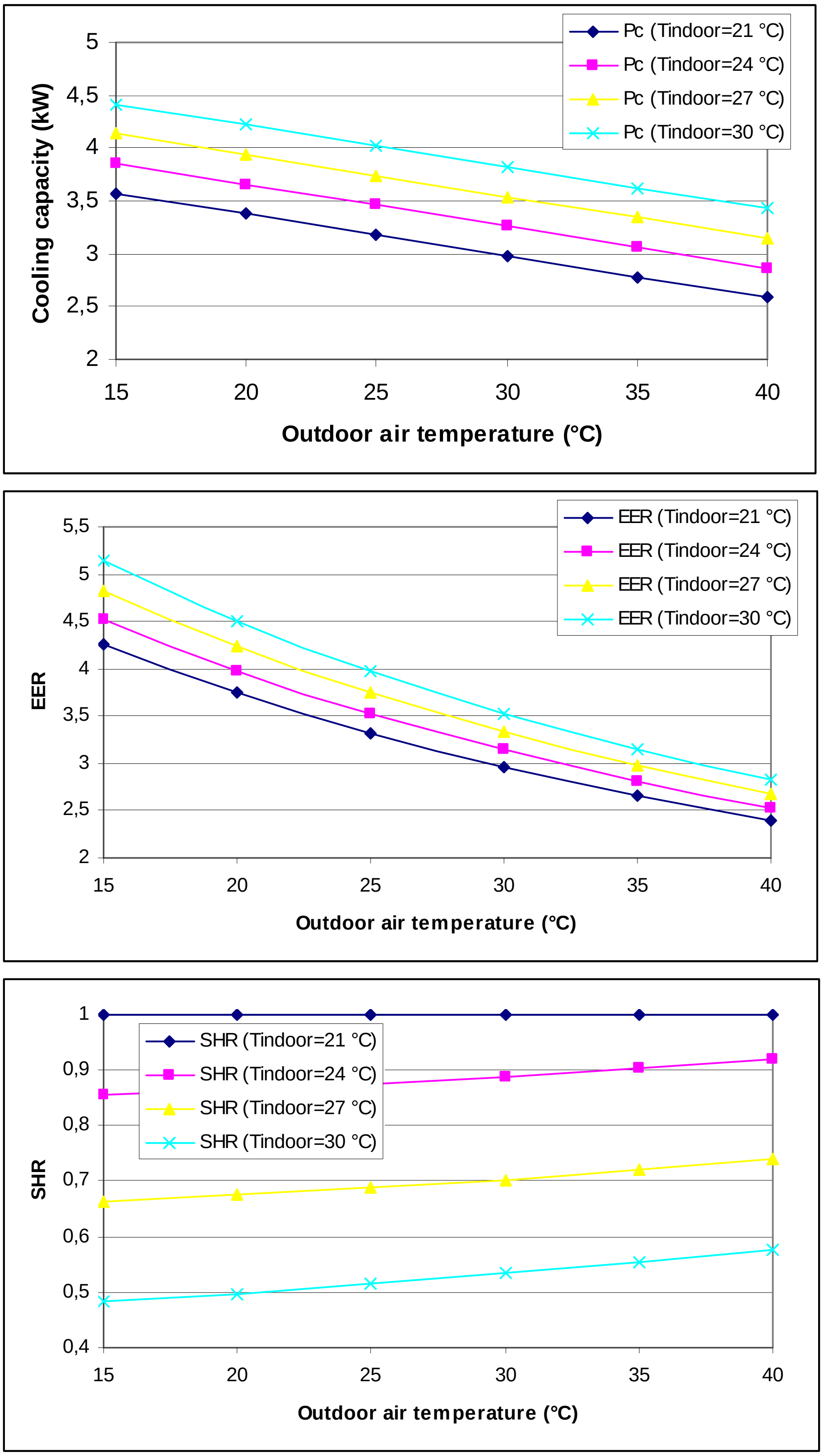


Figure 4-39: Typical variation of cooling capacity, EER and SHR for an average single split unit, source (Consoclim, 2004)

Variation of efficiency is dependent on many design parameters of the air conditioners as the refrigerant type used, the heat exchanger efficiency, the compressor performance curves -indeed, the variation of source temperatures implies variations of the compression ratio of the compressor.

In addition, for air conditioners with air as the coolant, part of the cooling capacity is sensible and part is latent: a part of the capacity corresponds to the dehumidification capacity of the cooling coil which depends on the characteristics and operating conditions of the indoor coil heat exchanger, including the temperature and humidity of the indoor air. The ratio of the sensible capacity (temperature decrease) to the total cooling capacity is called the SHR and is represented in Figure 4-39 for a constant humidity ratio of 0.5. In standard rating conditions, the SHR is close to 0.7.
Dehumidification capability is not so important for residential application, especially since European climate is rarely hot and humid as opposed to Japan or South-China climate. Nevertheless, it may be of importance that the unit dehumidifies correctly particularly in shops and offices. Highly efficient appliances may have lower dehumidifying capacities in standard conditions. But all units propose a specific dehumidification mode and some a dual mode with both dehumidification and temperature control.

Modeling results are reported in the annex 4.1.A for the tested reversible air conditioner with inverter.

Variation of heating capacity and COP with outdoor air temperature and humidity ratio

Manufacturers generally supply little information on the heating performances of air to air reversible air conditioners at varied outdoor conditions. When outdoor air temperature decreases, the pressure ratio increases. This induces a decrease in performances. Depending on the design of the heat pump and of the humidity content of the air, frost accumulation occurs starting from around 7 °C to 3 °C. Then, defrosting is needed and a consequent degradation of COP and of heating capacity occurs. The effect of these two trends is presented on the figure below.

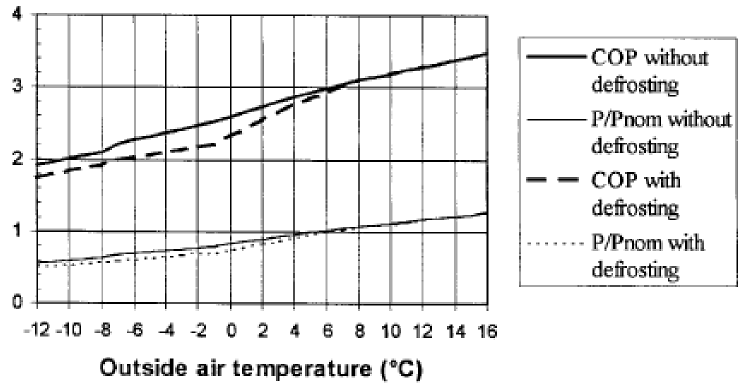


Fig. 1. Coefficient of performance at full load (COP) and ratio between real and nominal capacity at 6°C (P/P_n) as a function of outside air temperature with defrosting control on or off. The curves refer to an air to air heat pump with an indoor air temperature of 20°C.

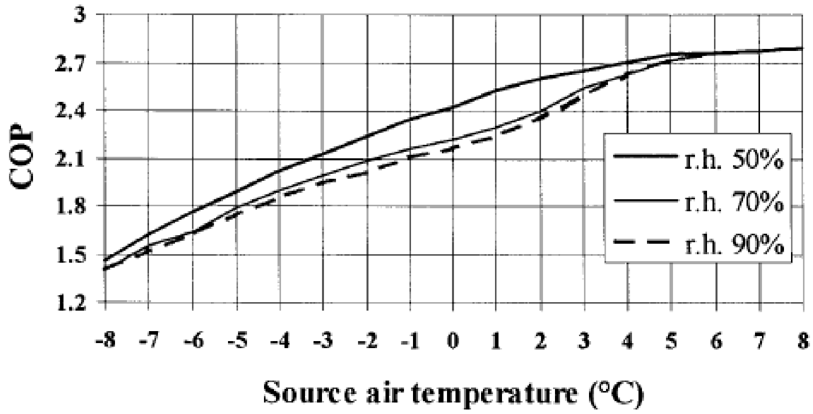
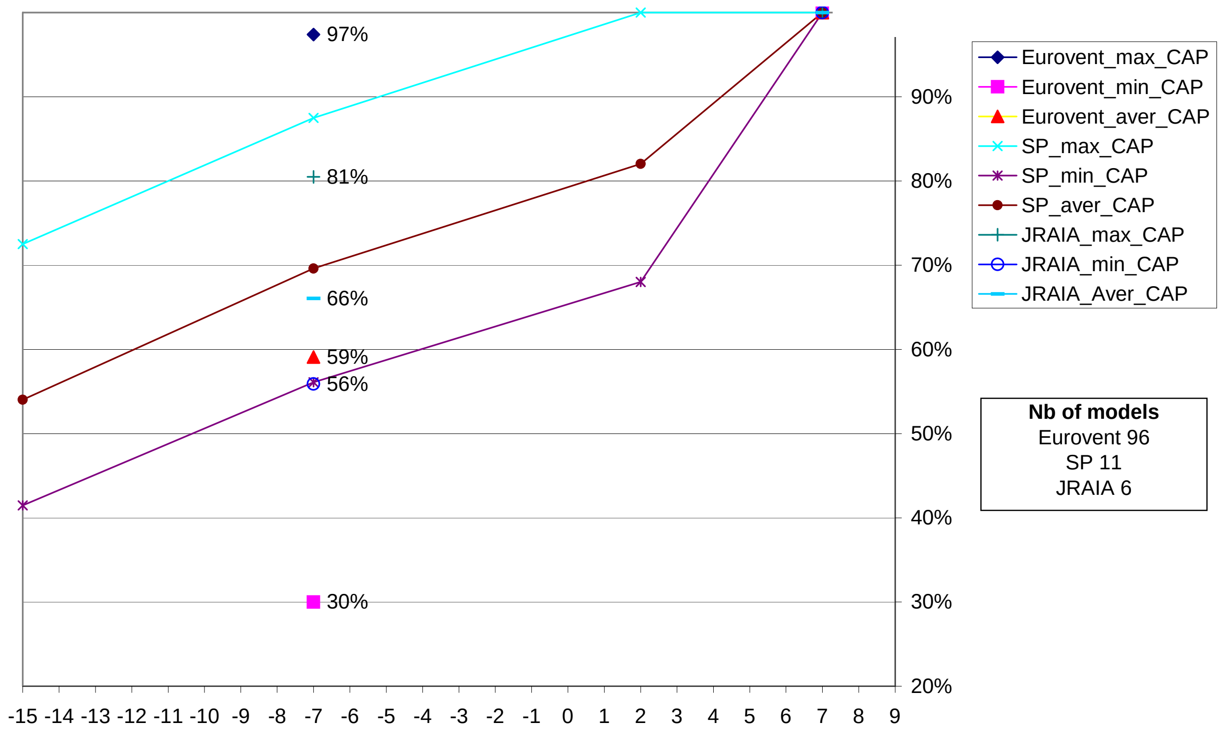


Fig. 6. COP of an air to air heat pump, in presence of optimized defrosting control, as a function of relative humidity and temperature of the source air. The sink temperature is assumed always equal to 21°C.

Figure 4-40: COP and heating capacity variation of air source heat pumps with outdoor air temperature and humidity (Schibuola, 2000)

As explained by (Schibuola, 2000), the impact of frosting / defrosting cycles will depend on the defrosting means used. In the past, fixed time interval methods were mainly used. The heat pump defrosted the outdoor coil at regular time steps (typically every 45 minutes to 1 hour) whatever the state of the coil may be. Control now normally uses on demand defrost control. The decision to launch a defrost cycle is based on the control of the temperature of the refrigerant or of the surface temperature of the outdoor heat exchanger. As explained by (Schibuola, 2000), the heat pump performance is then again more likely to be influenced by the humidity content of the air. However, no data of this influence has been identified. With the sensitivity proposed by (Schibuola, 2000), given that for temperatures below 7 °C, average winter humidity ratios (HR) are superior to 70 % for all European climates, it seems acceptable to fix the air humidity of the H1 (HR =87 %), H2 (HR =80 %) and H3 (HR =77 %) points.

Only for performances in H3 conditions (dry bulb temperature -7 °C, Wet bulb temperature – 8°C) of the ISO 5151 and EN 14511 standards have been identified. They come from 3 different sources the Eurovent database of certified products (Eurovent-tech, 2006), the recent test campaign led in the Swedish SP test laboratory (SP, 2005) and information supplied by Japanese manufacturers on air to air reversible air conditioners (with limited range of application: heating capacity inferior to 4 kW, refrigerant fluid R410A, inverter driven compressor) sold in Japan. Variation of the ratios of COP and heating capacity at H3 conditions to the COP and heating capacity at H1 conditions (dry bulb temperature 7 °C, Wet bulb temperature 6°C) is presented in the figure hereunder.



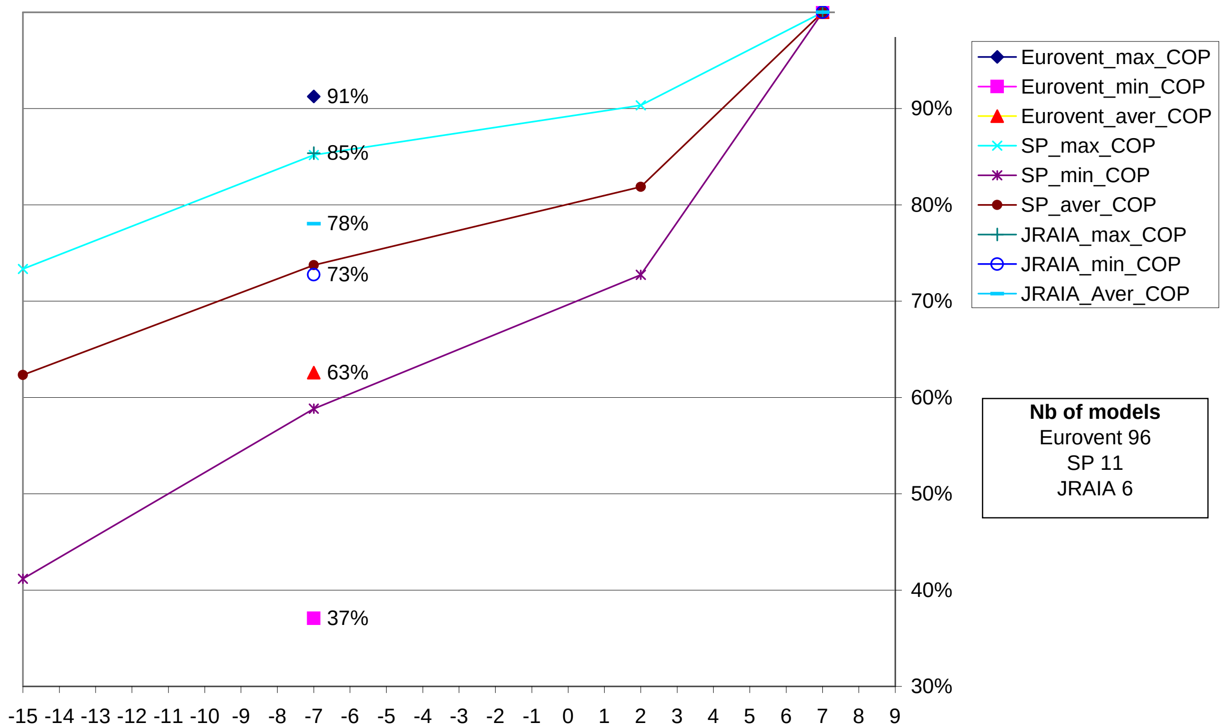


Figure 4-41: Ratio of COP and heating capacity (CAP) variation of air source reversible air conditioners in heating mode with outdoor air temperature and humidity in comparison of standard COP and heating capacity in H1 conditions; sources (Eurovent-tech, 2006), (SP, 2005) and (JRAIA, 2007). Nota: heating max COP and CAP figures are independent, e.g max curves do not refer to the same unit.

The variation in performances of the different units with temperature fall in heating mode is important. Since it is not required in the EN 14511 standard today, it is not possible to compare all units. It is certainly a possible improvement of the nowadays standard. Nevertheless, it should certainly be considered in the calculation of the energy consumption of the reversible air conditioners in the frame of this study. Default values for the evolution of the heating capacity and of the coefficient of performance could be the SP_min_COP and SP_min_CAP curves of Figure 4-41, since not too far of the average evolution of the larger set of data available (Eurovent-tech, 2006).

Variation of heating capacity and COP with indoor air temperature

As for variations of performances in the cooling mode with indoor conditions, these data are generally not supplied by manufacturers. When outdoor temperature is near the beginning of the frost / defrost range (between 3 and 7 °C), there may be coupled effects between variations of outdoor and indoor conditions. Nevertheless, these effects are not modeled even in best building simulation tools. Modeling results as reported in the annex 4.1.A for the tested reversible air conditioner with inverter are presented hereunder. The impact of the indoor temperature on the heating capacity is low but more important on the electric power input of the compressor. Final sensitivity of the COP is in the range of 1 to 2 % per °C.

Figure 4-42: Indoor unit heating capacity as a function of indoor and outdoor air temperature-heating mode

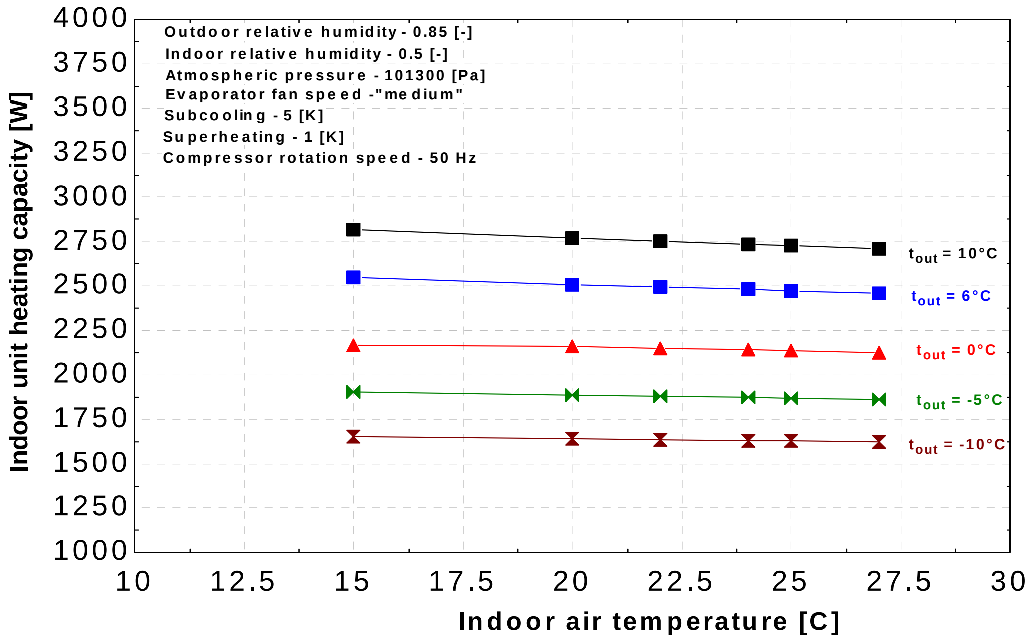
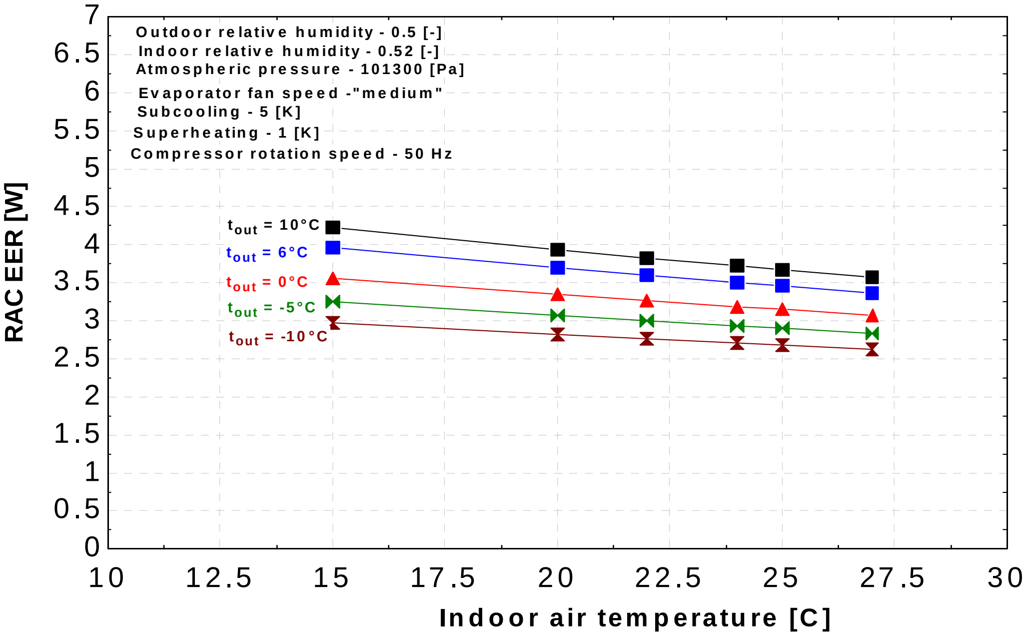


Figure 4-43: air conditioner COP as a function of indoor and outdoor air temperature-heating mode



Default behaviour laws

For energy consumption calculation of split appliances, standard laws of variations of the performances of the unit with indoor air conditions and outdoor air conditions are the ones of the model tested as reported in annex A of this task.

Concerning single duct unit, the laws of variation of performances with indoor conditions (condenser side air inlet temperature is also the indoor air temperature for these appliances) are reported below.

Note that the reference EER is lower than appears in the following graph finally, 2.3 instead of 2.6 but that what matters here are the variations as compared to rated full load performances when indoor air conditions vary.

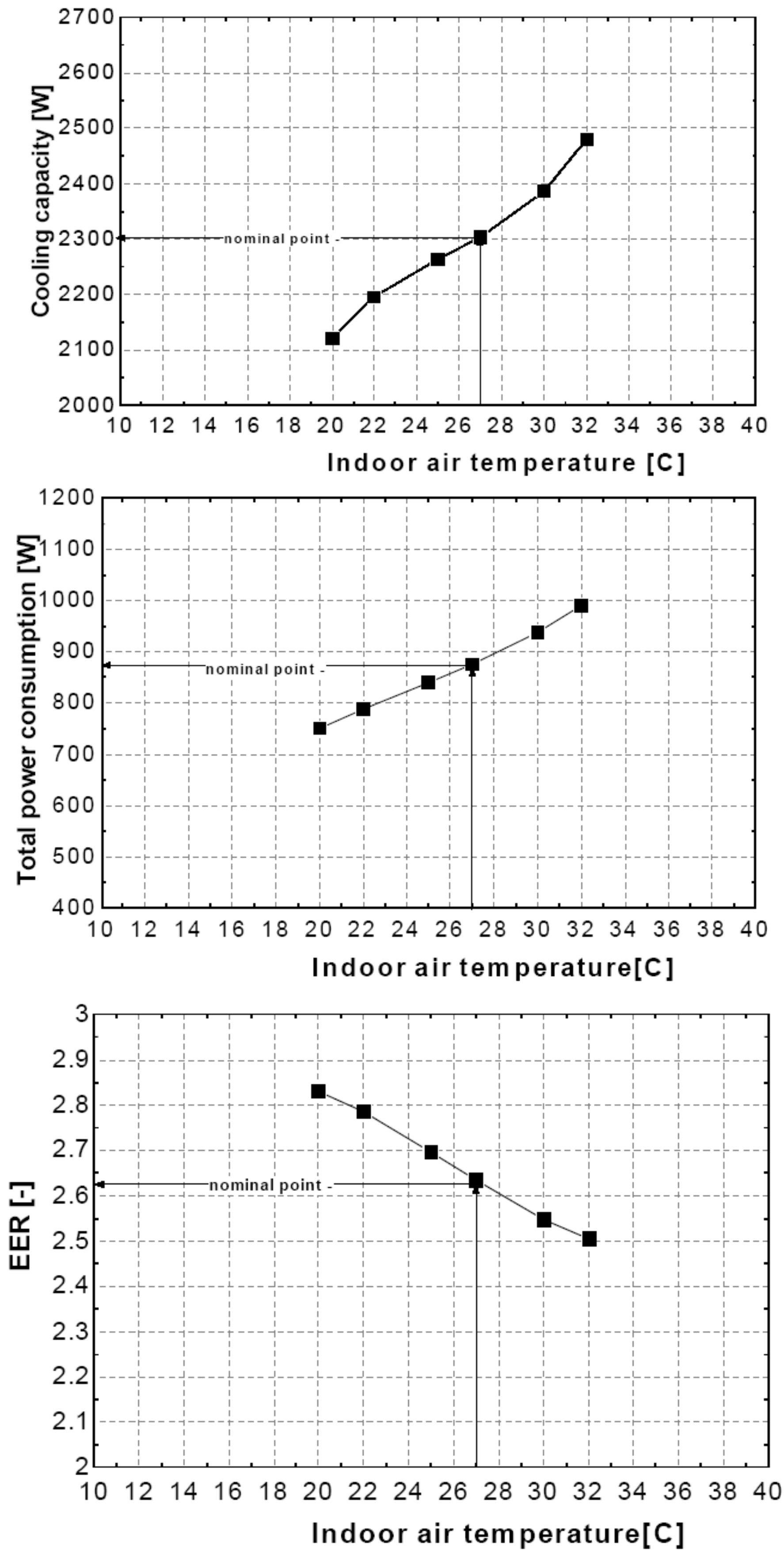


Figure 4-44: Performances variations of the single duct unit with variation of indoor air temperature at full load, relative humidity is set constant at 47 %.

4.4.1.3 Variation of performances with load ratio

The two ways of capacity control on the EU market for products in the range are ON-OFF cycles and inverter controlled compressors.

ON-OFF cycle part load efficiency degradation

PRESSURE EQUILIBRIUM

When the unit is started, energy is used to establish high and low pressure then reducing cooling capacity available for a short period of time. When unit is shut down, this energy is partly lost with pressure equalisation. As explained in (Henderson, 2000), this phenomena has been first modelled by (Parken, 1977) simply by taking into account a lag time in cooling capacity or heating capacity after compressor start-up; they referred to the normalised efficiency degradation as the part load factor, or PLF.

$$PLF = \frac{PartLoadEfficiency}{SteadyStateEfficiency}$$

On a cycle, the lag time is dependent on the air conditioner while the cycle time and relative ON and OFF times depend on the operating conditions, it is necessary to define the thermostat law of the air conditioner for a given application. A default thermostat law has been defined (Henderson, 2000) that corresponds to 3.25 cycles per hour. This can vary with the dead band control allowed by the air conditioner: set point more or less 1 °C or more or less 2 °C for instance. Henderson reports lower average maximum cycle rates of 2.5 in a study led in the USA in 1991.

(Parken, 1977) were the first to form a part load correlation. They used this concept to develop the part load degradation coefficient (Cd), that translates a linear degradation of efficiency with part load ratio.

PLF = EER part load / EER full load = 1 – Cd(1-PLR)*

The maximum number of cycles per hour directly impacts the PLF law for the same equipment with a given time constant at start-up.

Another important factor that affects this part load degradation is the design of the air conditioner. Whether the fan remains on while the compressor stops, there may remain cooling capacity without the compressor running. This will compensate partly the start-up efficiency loss. Another design solution is to use expansion valves or other means (solenoid valves) that enable to block the pressure equilibrium when the air conditioner starts. This can be done with non-bleed thermostatic expansion valves, solenoid valves or electronic expansion valves.

In the USA, default values have been defined for the Cd coefficient. For air conditioners, a default value of 0.25 is used (ARI Standard 210/240). For chillers, a default value of 0.1 is used. For air conditioners only, a procedure exists to challenge the default value. The unit is forced on-off for a period ON of 6 mn and a period OFF of 24 minutes, thus simulating approximately a 20 % part load ratio.
As reported by (Shirey, 2005), Historically these tests have been completed at dry coil conditions to make the test more manageable and repeatable. Work at the National Institute of Standards and Technology (NIST) had shown that transient measurements of humidity (or wet bulb) were difficult and that a cyclic test at dry-coil conditions yielded the same result as at wet-coil conditions. A well known default of this testing method is that forcing the fan to work after compressor stop enables to decrease the Cd coefficient while in real life, this would induce a small E

ER decrease and a large loss of cooling latent capacity (Shirey, 2005). In 2002, the US DOE (Dougherty, 2002) with cycling testing results based on two large databases (ARI and NIST) studied the possibility to modify the default value of 0.25 for units that used expansion valves or other means to stop or mitigate refrigerant migration at compressor stop. The idea was to avoid as much as possible these time consuming cyclic tests.

The different categories as reported in the table below were set up.

Equipment Category	Equalize During Off Cycle	Indoor Fan Off Delay	Equipment Examples
A	Yes	No	Cap Tube Orifice Bleed TXV
B1	No	No	Non-Bleed TXV Electronic Exp Device Liquid Line Solenoid
B2	Yes	Yes	Cap Tube Orifice Bleed TXV
C	No	Yes	Non-Bleed TXV Electronic Exp Device Liquid Line Solenoid

Table 4-8: Identified categories of different Cd values, (Dougherty, 2002)

Whether average of these groups were clearly different, but with a bias in the two sets of databases, of respective averages 0.07 (ARI) and 0.11 (NIST). The default value of Cd=0.25 has been kept until now. The table below gives for different default values by group, the percentiles that would cover the US central air conditioner Cd coefficients.

Percentile	A	B1	B2	C
99th	0.24	0.16	0.22	0.15
95th	0.22	0.14	0.14	0.12
90th	0.16	0.14	0.12	0.10
85th	0.14	0.12	0.11	0.09
80th	0.12	0.12	0.10	0.08
75th	0.12	0.11	0.10	0.07
70th	0.11	0.11	0.09	0.06
60th	0.10	0.09	0.08	0.05
50th	0.09	0.07	0.07	0.04
Sample Size	77	58	109	78

Table 4-9: Possible default Cd values for central air conditioners in the US, (Dougherty, 2002)

"STAND-BY" LOSSES

As previously mentioned, air conditioning systems may have parasitic electric consumption when the compressor is not running. This translates in an hyperbolic EER degradation when load ratio declines to 0. The equation for this simply assumes a constant supplementary power that does not decrease with load. Then, PLF can be written as:

*PLF = EER part load / EER full load = Load / (Cc * Load +(1 – Cc))*
With
Cc = 1- “Measured sleep power” / (Pe (c))

This degradation can generally be neglected for cycling except when the crankcase heater works when the compressor is OFF. The following figures illustrate 3 different values corresponding respectively to 1, 2 and 3 % of the electric power of the compressor at full load (Henderson, 2000).

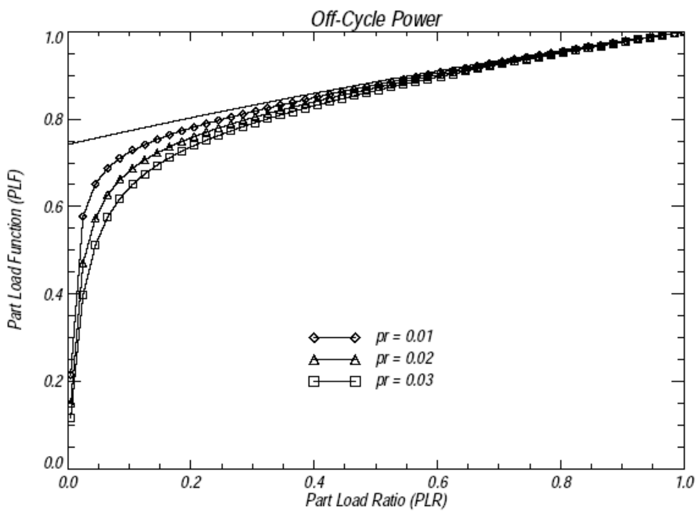


Figure 4-45: Part load degradation as a function of pr (=Cc) for 1 %, 2 % and 3 % of nominal full load compressor input, source (Henderson, 2000) with Cd = 0.25

For units with crankcase heaters, values identified for Cc in the technical analysis by component for EU compressors in the range 0.5 % to 3 %. For the time of cooling or heating operations, the yearly impact will then be dependant on the load curve applied, the sizing of the air conditioner as regards to the maximal load, the nature of the load (internal or climatic) …
The US DOE acknowledges this possible supplementary consumption and assumes it is taken into account because when the testing is made, energy consumption of the compressor is accounted for. The figure below shows the Cd linear coefficient and the real slope in case of parasitic losses. Nevertheless, the parasitic losses outside heating and cooling periods are not accounted for.

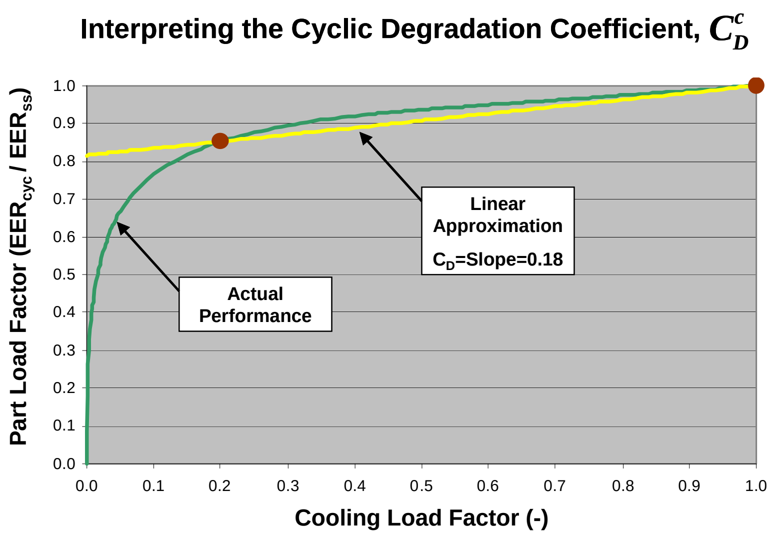


Figure 4-46: Degradation performance curve as modelled with Cd and in case of parasitic losses (Dougherty, 2002)

Another interesting point at part load conditions is the lower latent capacity of the indoor coil in cooling mode. A comprehensive study has been led (Shirey, 2005) and default models that depend on the design of the air conditioners are available.

ON-OFF cycle part load efficiency degradation: default hypothesis for base case energy computation

Final default values for energy consumption calculation are: Cd = 0.2 and Cc=0.025. This gives about 25 % degradation at 25 % load ratio. This curve is kept to represent cycling for both cooling and heating mode, for split and single duct units.

Inverter driven air conditioners: energy efficiency part load curves

Energy efficiency with inverter has been the subject of many research publication by the past because of the potential energy efficiency gains. Operating at reduced flow rate instead of cycling the compressor ON-OFF enables to take advantage of lower temperature differences between outdoor fluids and refrigerant temperatures. Compressor ratio is then decreased. According to the compressor efficiency for this new conditions, the energy efficiency at part load may be more or less improved.
Test results obtained on 11 heat pumps as published by (SP, 2005) are reported in the graph below. Units were tested at 75 % heating capacity and 50 % heating capacity at 7 °C. At 2 °C, only one point is available around 50 %. Results are presented as the ratio of COP at at part load to the COP at full load for outdoor and indoor conditions as a function of the load ratio.

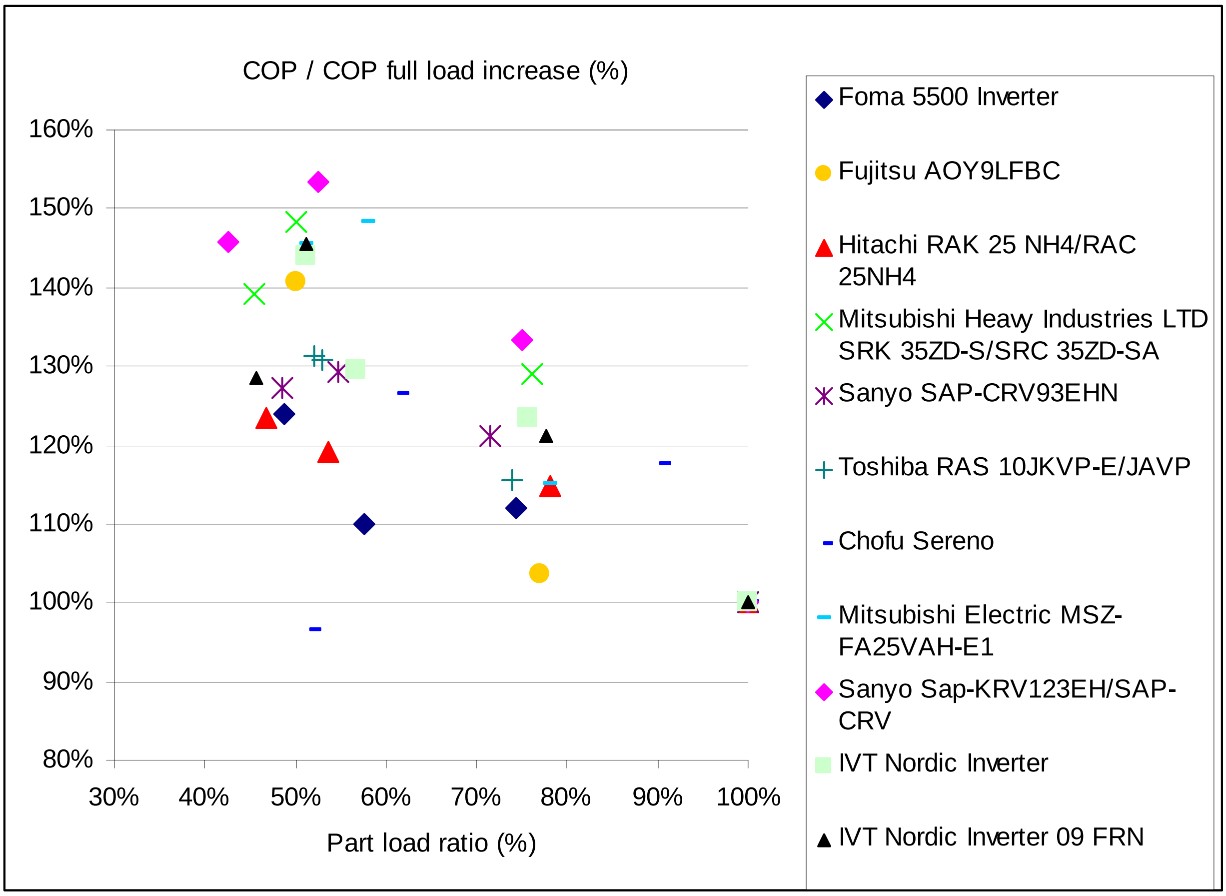


Figure 4-47: Part load performances of inverter driven air to air heat pumps (SP, 2005)

It appears that the energy efficiency generally increases with the reduction in refrigerant mass flow rate. At about 50 %, energy COP, as referred to full load COP lies between 97 % and 153 %. For different outdoor temperatures, this ratio may vary substantially, see for instance the Chofu Sereno heat pump in the figure above.

The same type of results in cooling mode has been supplied by Japanese manufacturers (JRAIA, 2007) and is presented in the table below. Results apply to a limited range of products sold in Japan (residential, below 4 kW, refrigerant R410A). The trend observed is about the same. The increase of EER is in average of 36 % at 50 % load rate.

	EER 100%	EER 50%	EER 50%/EER 100%
Air conditioner A	3.67	5.1	1.39
Air conditioner B	3.8	5.33	1.4
Air conditioner C	3.37	4.45	1.32
Air conditioner D	3.29	4.43	1.35
Average	3.53	4.83	1.36

Table 4-10: Part load performances of inverter at 50 % load ratio, (JRAIA, 2007)

Japanese manufacturers expect performances to maintain at lower load ratio, as reported in the part load performance standard JRA4046 (JRAIA, 2004).
In fact, it may exist a minimum working frequency under which the compressor will cycle ON and OFF as a fix speed compressor. In that case, the ON-OFF degradation part above should apply. This is illustrated by experimental results of (Anglesio, 2001). An illustration can be found in (Bory, 2006).

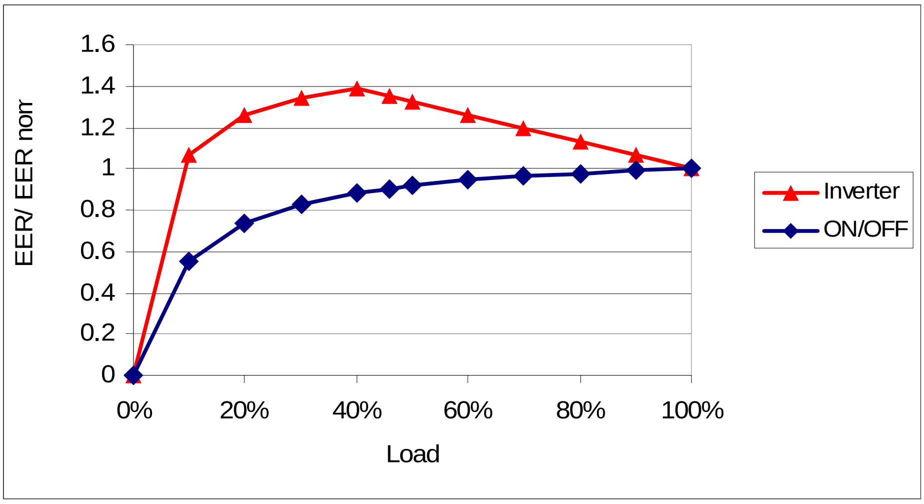


Figure 4-48: Part load performances of inverter driven air to air heat pumps (SP, 2005)

Beyond a load ratio of 40 %, the energy efficiency at part load decreases following the same equations as for compressor ON-OFF cycling.

4.4.2 Refrigerant use

This point has been studied precedently in task 3.1.2.2 in order to assess the real life efficiency decrease due to refrigerant charge leakage.

4.4.3 Water use

Water consumption is relevant for water cooled air conditioners working in open loop. According to national legislation, it can be authorized or not but precise information on this point has not been gathered. It is also relevant for units proposing evaporative cooling on the indoor coil as a supplementary cooling capacity (some single duct units) and for air cooled units that do evaporate water on their condenser side in cooling mode to reduce the refrigerant condensing temperature and thus reduce their energy consumption.

Water cooled air conditioners

Flow rates are indicated by manufacturers, either in close or open loop. Water flow rates are generally reduced in open loop, from 200 l/h/kW (kW of standard cooling capacity) for close loop circuits to 40 l/h/kW.

Water cooled air conditioners with cooling tower
For closed loop circuit, there is a loss also of water in case an open cooling tower is used. Evaluation made in (EECCAC, 2003) amounted to around 4 l/h/kW.

Evaporatively-cooled air conditioners
It is a classical feature to use the water extracted from the air of the room to decrease the dry bulb temperature of the air through the condenser. There is no additional water consumption in that case. Adding water, either by a small water tank to be filled or directly water from the network, is an option already used: DeLonghi in its technical specification indicates a 20 % increase in cooling capacity thanks to the evaporatively cooled condenser. The potential energy savings and water consumption will be analysed in task 6.

4.5 Use phase (system)

For air conditioners, the total energy consumption is the product of the cooling (and heating for reversible units) thermal energy needed multiplied by the energy efficiency. The operating conditions, like outdoor conditions of operation and average load, are a function of the climate, the sizing of the unit as compared to the cooling and heating needs and the energy efficiency of the unit for the different operating conditions.
This part on system analysis intends to supply a complete analysis of operating conditions for the different types of products. It is made of the following steps:

- Reconstitution of the heating and cooling energy thermal needs,
- Sizing cooling and/or heating capacity of equipment,

It will then be possible to calculate the final energy consumption of the products for different buildings, climates, integrating also real life conditions.

4.5.1 Heating and cooling energy need calculations

The general objective here is to calculate the heating and cooling demand that air conditioners face during their life. As shown in the use phase of the product, the annual cooling and heating requirements are to be calculated not only according to EN 14511 test standard conditions but also under off standard conditions in order to be able to calculate the annual energy use consumption.
The analysis methodology was selected to study the factors that affect air conditioner efficiency and which are mainly:

- Climate
- Heating and cooling load (related to buildings)

A key assumption for the analysis to be performed is the definition of buildings. Their thermal characteristics vary not only by climate but also by building type. The building types selected following the market analysis in task 2 are:

- Residential,
- Small office,
- Small Retail.

These three buildings types accounted for the three main representative sectors in which RAC are installed, in accordance with market analysis results. Moreover, and in order to represent building stock, buildings were categorized for old (those built prior to a specific chronology for each country) and new construction vintages.

The methodology is in three steps:
1. Define detailed building characteristics for "old" and "new" constructions.
2. Simulate annual cooling and heating thermal energy required yearly using building energy simulation software.

4.5.1.1 Building Energy Simulation

Our approach is to use building energy simulation for the analysis of energy efficiency and building loads. Such an approach requires collection of information such as detailed building geometry, a complete envelope description to incorporate construction and glazing thermal characteristics, air-conditioning systems, control methods, and hourly weather information. Building Energy simulation usually requires more than 10 weather parameters including solar radiation, temperature, humidity, wind speed, wind direction, cloudiness conditions, atmospheric pressure. Here follows the general description of the framework of the performed simulations.

4.5.1.2 Building Description

Geometry

Residence

The residence is modeled as an apartment with characteristics as defined in Table 4-11. 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 4-12. 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 4-13 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 4-11: 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^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	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 4-12: 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 4-13: 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 4-14).

Table 4-14: 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 4-15, and Table 4-16, Table 4-17, Table 4-18, 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 4-15: Internal Gains

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

Table 4-16: 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 4-17: 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 4-18: 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.

4.5.1.3 Energy Simulation Software

Detailed computer simulation has been performed using TRNSYS 16 model, which calculates building energy consumption on an hourly basis over a year (8760 hours) using local hourly weather data.

TRNSYS is a general purpose, component-based hourly simulation program, constantly included in *State-of-the-Art* reviews for building and HVAC components and system simulation and design tools, as one of the most elaborate software for whole building energy analysis, with unique technical abilities.

TRNSYS program allows the calculation of air-conditioning heating and cooling load internally in the building model by specifying a set point temperature and a maximum power, or externally from the building by modeling all of the components of the HVAC system, including split air conditioning units.

In this analysis the simulations are first performed for each building and for all considered locations, using the internal heating/cooling load calculation, in order to determine the maximum load required for the building and the annual energy requirements. The information gathered from the first set of simulations is used to size the equipment and to estimate average annual energy requirements for heating and cooling.

The simulations will then be performed with an external air conditioner component, taking under consideration performance curves that describe the performance of the equipment at off-design conditions, suitable to simulate unit’s annual performance. These curves describe in detail the cooling systems' sensible and latent cooling capacity (also heating capacity) and electric power consumption under all operating conditions.

It is worth to mention, that TRNSYS program does not need to be validated, being a calculation tool that uses standard numerical techniques for solving equations and a building model compliant with ASHRAE Standard 140 (2001) - Standard Method of Test for Evaluation of Building Energy Analysis Computer Programs. Validation is needed for the components (heater, cooling unit, energy wheel, etc.) themselves, to ensure they work properly.
The operation conditions (operations schedules and loads) are calculated from the described building types, whose energy use characteristics are calculated for their specific features. These include detailed building components, such as building geometry and orientation, shading, walls, windows, adjacent zones etc. The building operating conditions are also taken into account, by considering parameters such the occupancy levels, the interior load due to lighting and other equipment and mainly the schedules that describe weekdays and weekends variation.

4.5.1.4 Weather Data

Local full-year hourly weather data are required as inputs to TRNSYS building simulation program. The data used are in Typical Meteorological year (TMY 2) format. Files are derived from up to 18 years of hourly weather data.

The weather data is supplemented by solar radiations estimated on an hourly basis from earth-sun geometry and hourly weather elements, particularly cloud amount information.
It is important to note that the TMY2 format represents typical rather than extreme climate conditions.

We use one climatic file per country (for all EU 25), assuming that the capital or another city is the representative climatic zone.

For three climates, warm (Greece – Athens), cold (Finland – Helsinki) and moderate (France - Mâcon), the temperatures, average and minimum as well as the solar radiation and humidity ratio are represented on the following four figures.

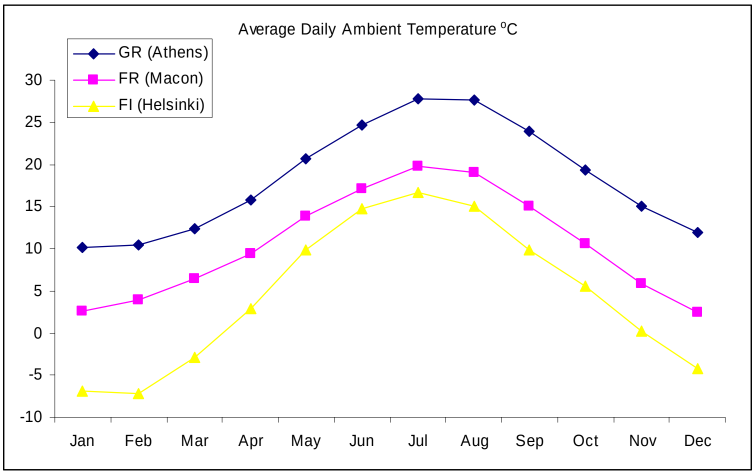


Figure 4-49: Monthly average daily ambient temperature for three climates

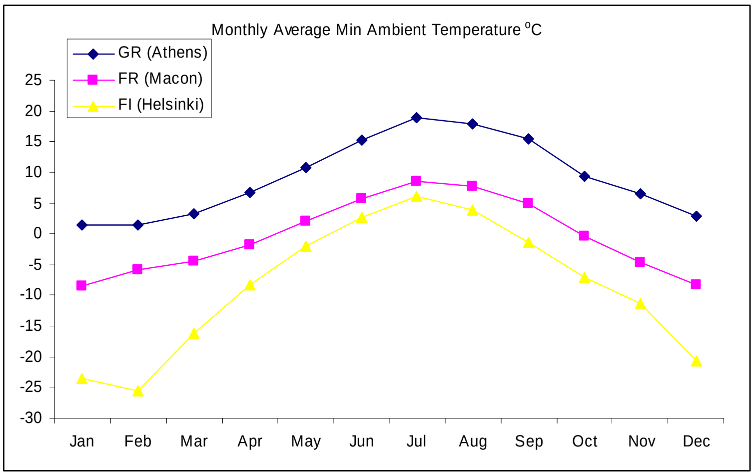


Figure 4-50: Monthly average minimum daily ambient temperature for three climates

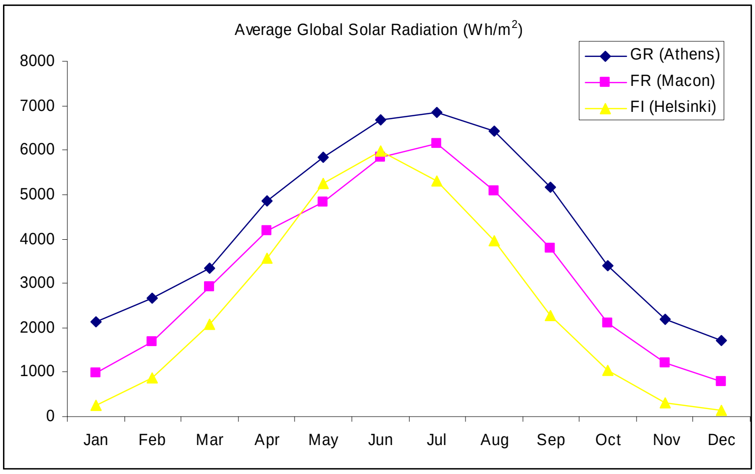


Figure 4-51: Average global solar radiation for three climates

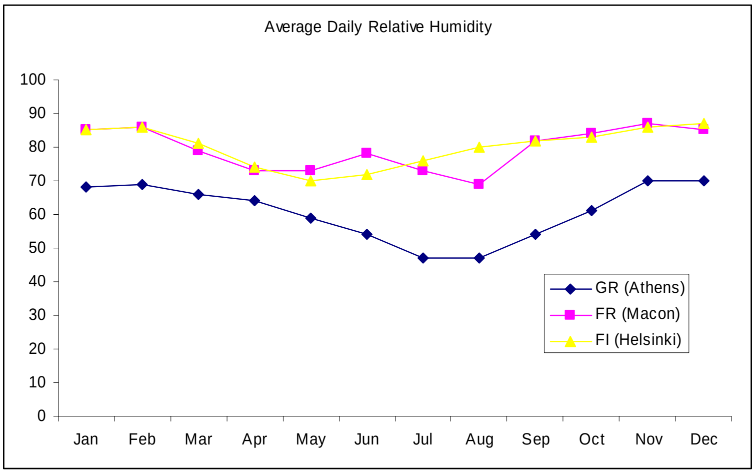


Figure 4-52: Average daily humidity for three climates

The location, longitude/latitude, altitude, cooling degree-days and climate type for the simulated climates are shown in Table 4. A city has been chosen to represent a specific country.

Table 4-19: Weather data for climates in European countries, EU 27****

Country	City	Location	Altitude (m)	Cooling Degree Days (base 10°C)	Heating Degree Days (base 10°C)	Cooling Degree Days (base 18°C)	Heating Degree Days (base 18°C)	Climate Type*
AT	Vienna	16°34N/48°7E	190	1226	1313	177	3184	5A
BE	Brussels	50°54N/4°31E	58	893	978	71	3076	5C
DK	Copenhagen	55°40N/12°18E	28	723	1456	37	3690	5A
FI	Helsinki	60°19N/24°58E	53	590	2588	36	4955	6A
FR	Macon	45°25N/9°16E	103	1567	944	349	2446	4C
DE	Bremen	53°2N/8°48E	24	842	1159	68	3305	5C
GR	Athens	37°54N/23°43E	15	3145	84	1150	1009	4A
IE	Dublin	53°25N/6°13E	82	572	760	2	3110	5C
IT	Milan***	45°25N/9°16E	103	1567	944	349	2446	4C
LU**	Nancy	48°40N/6°13E	212	943	1194	92	3212	5C
NL	Amsterdam	52°17N/4°46E	2	833	1011	46	3143	5C
PT	Lisbon	38°43N/9°8E	77	2515	29	531	966	4C
ES	Madrid***	40°27N/3°32E	582	1907	479	549	2041	4C
SE	Gothenburg	57°46N/11°52E	20	658	1705	22	3988	5A
UK	London	51°9N/0°10E	59	757	910	28	3101	5C
CY	Larnaca	34°52N/33°37E	2	3291	13	1112	754	4A
CZ	Prague	50°5N/14°16E	364	823	1606	70	3773	5A
EE	Tovarene	58°27N/26°46E	59	666	2376	48	4679	6A
HU	Budapest	47°25N/19°10E	140	1438	1294	276	3051	5A
LV	Riga	56°52N/24°7E	14	775	2231	62	4439	6A
LI**	Gdansk	54°31N/18°36E	13	644	1724	19	4019	6A
MT	Luga	35°49N/14°25E	135	3140	6	972	759	4A
PL	Warszawa	52°16N/20°58E	130	946	1745	103	3822	5C
SK	Bratislava	48°10N/17°6E	289	1363	1246	237	3040	5A
SL	Ljubljana	46°4N/14°31E	299	1233	1299	170	3166	5A

**(ASHRAE Standards 90.1-2004 and 90.2-2004 Climate Zone), ** Climate data from the nearest available meteorological station,*

**** For Italy, two climatic files are used and represent half the climate of Italy, Milan and Rome; the same applies to Spain with Madrid and Seville.*
*****Bulgarian climate is represented by the climatic data of Portugal, and Romania with Hungary.*

4.5.1.5 Analysis of heating and cooling need results

Table 4-20 to Table 4-25 show the results for the EU-25 average building types considered.
Heating set point temperatures are all 21°C, setback for heating (reduced temperature when the building or room is not occupied) is at 12°C for offices and shops and at 15 °C for residences.

As far as it concerns heating setback temperatures, we have compared results by performing sensitivity analysis, considering four alternatives: no set back, set back at 15 and constant operation at 21 °C. Considering constant heating operation at 21°C, represents an extreme assumption that can cause an % increase of 73%, 60% and 70% for residences, offices and shops respectively. These are maximum differences that occur in Cold countries. Considering set back at 15°C causes an increase of 30%, 25% and 12% for residences, offices and shop respectively. When considering 12 °C set back temperature, means 21°C during occupancy, differences are of 16%,10% and 3%.

Cooling set point was set at 25 for offices and dwellings in all climates except warm countries (see Table 4-11 to Table 4-13) where it is set at 26 °C. For shop, the cooling set point temperature is 23°C.

Table 4-20: Residential, Heating - energy consumption, maximal load and coincident temperature (heating set back – 15 °C)

	NEW CONSTRUCTION			OLD CONSTRUCTION		
	KWh/m² /year	**W/m²**	**T (°C)**	**KWh/m²/year**	**W/m²**	**T (°C)**
AT	103,84	116,30	-8,65	155,14	136,08	-10,05
BE	84,19	100,86	-6,50	126,09	118,73	-5,90
DK	112,36	116,56	-10,10	167,75	136,21	-10,10
FI	119,46	104,78	-17,40	157,20	122,52	-17,40
FR	73,99	100,95	-6,85	116,60	119,75	-7,50
DE	95,51	112,82	-8,00	143,31	131,56	-9,85
GR	14,52	79,00	0,55	17,66	63,12	0,70
IE	66,33	95,34	-2,90	103,64	106,70	-2,90
IT-Mi	93,36	113,36	-6,95	173,12	145,93	-6,65
IT-Ro	18,64	83,93	-0,55	55,07	105,52	-2,30
LU	90,65	113,83	-8,25	137,76	131,16	-8,80
NL	82,48	104,36	-5,95	125,68	118,74	-5,65
PT	4,33	54,19	5,15	18,85	84,23	4,05
ES-Ma	35,01	96,20	-5,00	91,01	133,08	-5,00
ES-Se	5,89	61,80	4,35	25,02	89,84	3,50
SE	76,84	101,06	-13,10	105,34	114,82	-13,10
UK	75,92	99,69	-3,95	117,11	112,77	-3,95
CY	2,61	43,17	4,85	15,42	71,23	5,05
CZ	76,26	89,98	-10,05	169,02	132,27	-13,60
EE	105,25	105,07	-19,40	230,35	151,65	-19,40
HU	57,23	91,08	-11,65	132,99	126,27	-11,30
LV	102,93	103,87	-17,95	224,73	146,14	-11,60
LT	78,60	87,27	-10,85	182,32	132,40	-10,50
MT	1,40	37,10	8,30	10,77	79,70	4,60
PL	84,75	93,12	-12,20	182,93	127,64	-8,50
SK	55,32	94,24	-12,05	129,42	128,03	-12,05
SI	57,57	92,87	-11,80	135,82	132,07	-11,80

Table 4-21: Residential, Cooling - energy consumption, maximal load and coincident temperature (cooling set back – NO)

	NEW CONSTRUCTION			OLD CONSTRUCTION		
	KWh/m² /year	**W/m²**	**T (°C)**	**KWh/m²/year**	**W/m²**	**T (°C)**
AT	15,71	87,30	29,15	15,31	97,42	29,15
BE	6,46	67,67	22,60	5,99	63,59	22,60
DK	4,21	52,39	19,65	3,80	57,29	24,30
FI	24,70	68,22	21,15	22,28	74,04	20,80
FR	18,40	85,18	28,00	16,85	81,12	28,00
DE	7,21	80,97	26,40	6,71	88,03	26,40
GR	65,02	123,76	33,50	67,94	167,59	32,85
IE	1,70	55,59	21,90	1,51	55,31	21,90
IT-Mi	26,01	103,25	30,60	25,80	123,49	30,60
IT-Ro	45,48	117,28	30,45	45,24	138,50	30,45
LU	9,63	74,77	27,40	9,13	84,93	27,40
NL	5,70	56,97	25,65	5,33	69,17	25,45
PT	42,58	157,65	31,15	38,54	182,73	29,75
ES-Ma	38,92	114,77	32,85	39,27	141,69	33,80
ES-Se	71,50	147,00	36,65	74,19	183,21	36,45
SE	21,55	73,48	23,55	18,60	76,85	24,70
UK	4,05	51,52	22,80	3,62	57,82	24,60
CY	71,15	118,16	32,10	78,82	142,30	33,40
CZ	8,54	58,78	23,40	5,15	58,65	23,40
EE	39,66	88,09	20,20	11,67	94,06	24,00
HU	27,82	69,53	25,40	21,36	97,04	27,10
LV	40,95	88,24	25,15	11,82	70,78	27,10
LT	28,24	57,74	25,00	1,28	53,77	23,35
MT	68,48	136,20	33,00	74,80	163,56	33,00
PL	27,24	58,62	29,30	16,37	64,99	29,30
SK	27,90	86,97	28,75	21,31	83,68	28,75
SI	19,84	67,27	27,30	13,48	70,27	27,15

Table 4-22: Office, Heating – energy need, maximal load and coincident temperature (heating set back – 12 °C)

	NEW CONSTRUCTION			OLD CONSTRUCTION		
	KWh/m^2 /year	W/m^2	T (^{o}C)	KWh/m^2/ year	W/m^2	T (^{o}C)
AT	45,23	86,72	-8,50	95,55	116,84	-8,50
BE	37,67	84,93	-5,30	80,23	106,12	-6,90
DK	50,39	99,61	-8,95	106,01	117,38	-5,85
FI	73,06	114,27	-14,15	96,20	124,39	-14,15
FR	32,52	81,51	-2,70	72,85	101,99	-3,35
DE	44,26	91,59	-8,35	92,51	125,26	-8,95
GR	12,86	86,29	0,45	27,69	107,25	0,45
IE	29,45	70,76	1,30	65,58	101,86	-3,85
IT-Mi	64,41	102,42	-2,05	113,12	130,95	-4,45
IT-Ro	16,95	84,41	0,60	36,49	99,11	-0,15
LU	40,00	86,67	-8,90	86,38	118,19	-8,45
NL	37,34	88,25	-6,30	80,04	103,13	-3,85
PT	6,78	62,50	3,80	17,86	88,68	3,80
ES-Ma	31,60	103,83	-5,60	63,91	126,94	-5,60
ES-Se	7,78	70,72	4,00	19,58	95,60	0,95
SE	49,46	95,97	-13,70	67,79	111,33	-13,60
UK	33,08	85,91	-3,75	73,45	102,86	-2,90
CY	1,62	30,89	8,05	6,90	56,73	8,05
CZ	47,71	97,12	-10,25	105,72	125,48	-11,00
EE	69,18	109,35	-16,10	150,33	141,96	-13,75
HU	36,41	83,63	-11,55	85,15	112,92	-3,55
LV	66,54	103,17	-13,25	145,95	147,39	-12,75
LT	48,60	86,99	-5,75	114,77	124,06	-11,15
MT	1,55	50,88	5,15	6,25	74,78	5,15
PL	53,40	87,04	-9,55	115,12	126,60	-7,80
SK	35,21	83,97	-4,75	82,57	115,18	-7,00
SI	35,97	80,01	-3,70	85,14	114,16	-7,75

Table 4-23: Office, Cooling - energy need, maximal load and coincident temperature (cooling set back – NO)

	NEW CONSTRUCTION			OLD CONSTRUCTION		
	KWh/m^2/ year	W/m^2	T (^{o}C)	KWh/m^2/ year	W/m^2	T (^{o}C)
AT	35,12	107,34	29,15	28,81	95,02	29,15
BE	21,95	75,02	25,40	15,17	69,49	25,50
DK	19,72	72,15	22,00	12,50	62,37	24,00
FI	39,02	79,50	26,15	31,20	77,96	25,60
FR	35,76	102,08	28,00	28,52	97,88	28,10
DE	21,08	91,21	26,40	14,53	95,97	25,20
GR	61,79	126,61	33,00	62,86	144,80	33,15
IE	10,78	68,22	22,10	4,64	58,46	22,10
IT-Mi	33,44	111,65	30,60	32,19	114,84	30,60
IT-Ro	50,22	119,96	29,75	49,12	135,84	29,95
LU	27,85	96,02	27,30	20,77	83,93	27,40
NL	20,77	83,39	24,10	13,40	70,18	24,10
PT	41,60	138,12	32,40	37,97	162,57	32,40
ES-Ma	41,02	115,65	33,30	40,17	127,08	33,80
ES-Se	65,40	160,35	36,60	67,64	190,57	35,70
SE	39,61	91,15	24,10	30,16	74,46	24,70
UK	18,60	65,27	22,10	11,67	58,49	22,80
CY	72,46	140,18	32,10	75,01	158,46	32,10
CZ	27,17	82,08	25,30	14,05	65,88	25,30
EE	38,12	85,52	25,70	20,29	77,85	20,15
HU	48,00	101,65	27,80	34,46	82,33	28,20
LV	41,15	96,73	28,30	21,97	82,22	20,75
LT	25,12	88,01	20,60	8,86	61,59	22,75
MT	67,39	147,91	32,50	67,71	165,84	33,00
PL	43,86	95,61	25,95	27,45	74,48	27,15
SK	46,88	115,41	28,75	33,50	104,01	29,00
SI	41,04	105,12	27,85	27,17	90,38	27,80

Table 4-24: SHOP, Heating - energy need, maximal load and coincident temperature (heating set back – 12 °C)

	NEW CONSTRUCTION			OLD CONSTRUCTION		
	KWh/m² /year	W/m²	T (°C)	KWh/m² /year	W/m²	T (°C)
AT	89,49	121,53	-3,90	140,49	122,74	-4,10
BE	76,92	115,30	-2,00	120,12	128,23	-6,90
DK	99,68	120,16	-1,70	157,14	140,77	-5,85
FI	101,52	121,45	-10,65	137,97	142,88	-11,90
FR	67,69	118,04	-7,00	110,07	129,29	-6,85
DE	87,22	127,66	-6,90	136,09	142,00	-6,90
GR	14,30	90,54	0,45	52,52	130,51	0,45
IE	64,06	106,43	-3,85	100,13	121,03	-2,85
IT-Mi	76,08	119,25	-7,15	182,16	168,94	-4,45
IT-Ro	17,42	92,68	0,60	63,17	124,22	3,50
LU	81,37	129,37	-8,75	129,18	140,58	-4,40
NL	76,26	117,20	-6,30	120,18	130,49	-6,30
PT	6,66	59,13	3,75	36,67	121,32	3,80
ES-Ma	32,82	107,24	-0,50	105,33	161,12	-2,60
ES-Se	6,61	65,47	3,90	37,74	127,83	0,95
SE	69,75	110,85	-9,00	97,18	145,39	-13,70
UK	70,26	123,42	-3,75	111,31	131,35	-4,10
CY	1,37	32,57	8,00	18,69	89,37	8,05
CZ	69,06	108,76	-6,70	156,22	154,92	-10,15
EE	92,86	118,26	-8,45	214,48	171,87	-10,35
HU	52,01	103,84	-3,50	123,27	137,62	-7,95
LV	90,74	118,30	-12,75	208,93	173,35	-12,75
LT	70,59	103,38	-5,15	166,35	139,01	-5,75
MT	0,95	40,08	5,15	16,76	107,27	5,15
PL	75,27	106,53	-3,10	166,52	142,77	-8,90
SK	51,10	101,88	-10,40	120,56	134,60	-4,75
SI	52,59	100,57	-4,10	125,39	131,18	-3,70

Table 4-25: SHOP, Cooling - energy need, maximal load and coincident temperature (cooling set back – NO)

	NEW CONSTRUCTION			OLD CONSTRUCTION		
	KWh/m²/ year	W/m²	T (°C)	KWh/m²/ year	W/m²	T (°C)
AT	49,83	118,83	28,65	43,08	120,74	28,65
BE	29,94	101,91	29,50	25,15	116,17	28,70
DK	25,61	89,20	27,60	20,72	90,02	27,60
FI	42,04	82,19	22,45	39,06	82,57	19,00
FR	51,08	110,63	29,35	45,21	123,27	27,35
DE	28,07	100,23	27,35	23,11	95,15	28,20
GR	115,99	148,89	33,00	104,00	159,55	37,70
IE	13,35	58,96	21,90	8,94	63,85	21,90
IT-Mi	69,92	124,96	30,05	55,81	152,78	29,95
IT-Ro	96,72	145,04	30,15	81,31	147,86	29,75
LU	40,11	100,25	28,25	34,31	122,48	28,75
NL	28,13	101,26	27,30	22,52	107,00	29,15
PT	81,59	141,26	31,50	61,25	144,79	32,10
ES-Ma	79,89	138,14	32,85	66,00	150,66	36,25
ES-Se	119,28	176,87	35,60	107,51	188,71	33,80
SE	42,54	72,45	23,20	38,34	111,22	22,05
UK	25,53	85,18	28,20	20,06	78,29	27,35
CY	130,10	162,03	35,70	120,44	164,36	32,55
CZ	32,94	77,57	28,45	21,50	75,64	26,55
EE	41,91	92,57	26,40	27,36	84,79	26,15
HU	59,26	98,73	33,55	52,54	100,27	28,00
LV	44,12	92,24	28,75	28,99	142,32	21,95
LT	28,29	71,12	23,35	13,67	65,25	23,35
MT	120,53	164,69	31,35	108,03	168,31	32,25
PL	48,50	100,68	25,60	37,87	105,35	29,25
SK	57,33	96,25	28,10	50,51	113,55	28,75
SI	50,22	103,36	31,55	41,49	107,77	31,55

There are three main factors influencing cooling and heating needs:
- the type of building
- the building age
- the appliance

The cooling and heating load differ according to the types of buildings. As shown in Figure 4-53 and Figure 4-54, cooling needs are more important in retails than in offices and more important in offices than in residences. Heating needs for residences and retails are quite similar and higher than office ones. The main factors that explain this threshold are the importance of the glazed surface, the internal loads for offices and the different set points used.

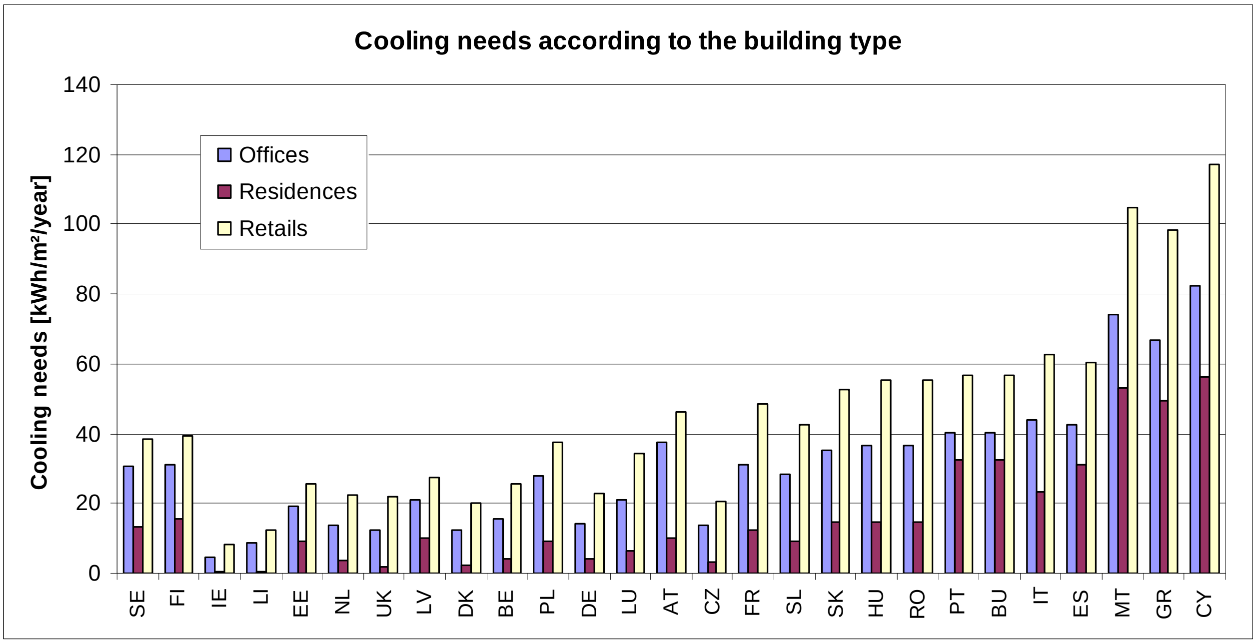


Figure 4-53: Cooling needs according to the different building types

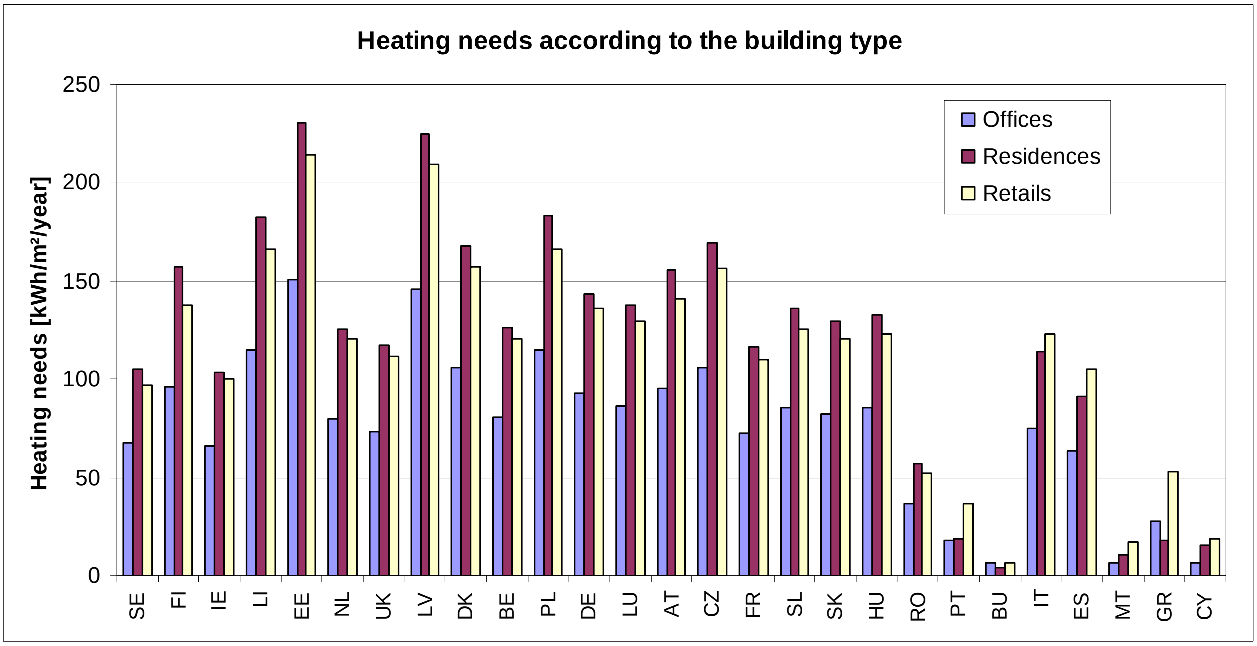


Figure 4-54: Heating needs according to the different building types

Regarding the difference between existing and new buildings, cooling needs are higher in new ones whereas heating needs are more important in existing ones. This is mainly due to a higher level of insulation in new ones along with a lower infiltration rate. A comparison of cooling needs between new and existing office buildings is presented in Figure 4-55, in this example the difference can reach 40 % but is almost null for Southern countries. Regarding heating needs, a comparison between new and existing office buildings is presented in Figure 4-56. New buildings enable a reduction of heating needs by a factor of about 2 for most of the countries.

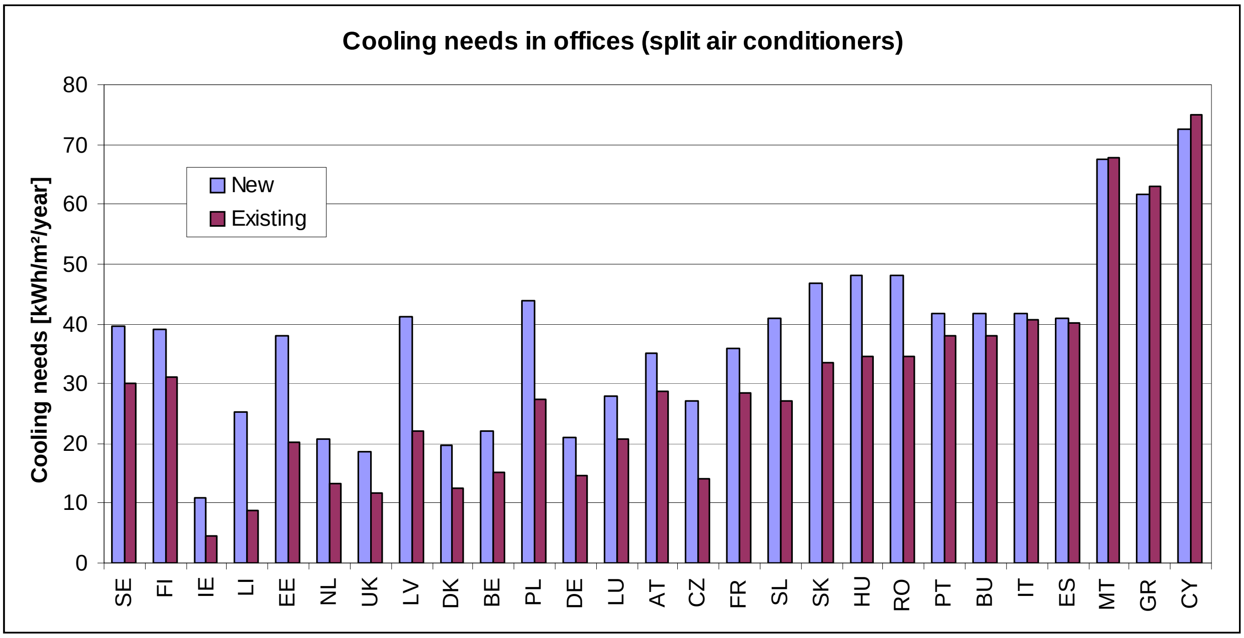


Figure 4-55: Difference between cooling needs in new and existing office buildings

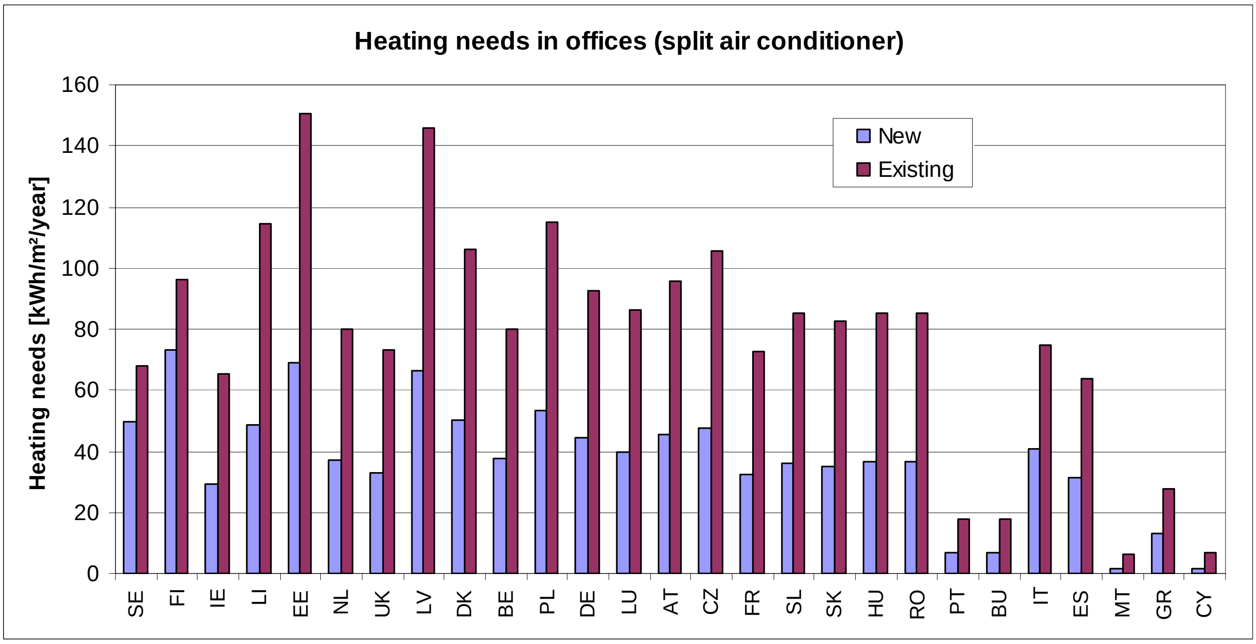


Figure 4-56: Difference between heating needs in new and existing office buildings

Single duct were also fully simulated with equal sizing than for split air conditioners. The ventilation rates of the buildings was increased of the air expelled from inside to outside by these products when cooling the indoor air. Indeed, single duct appliances and split systems differ regarding cooling needs since the first ones introduce outdoor air inside the room.
Apparently, average cooling demand increases only about 2 % as a weighted market average figure but this also linked to the fact that indoor air temperature cannot be maintained at the required set point in the room. In the commercial sector, average load increases by 30 % while it decreases of 25 % in residences and in offices. In average, the same cooling demand (energy) can be considered also for single duct air conditioners.
For instance, the differences in terms of cooling needs are plotted for existing offices (Figure 4-57). It appears that cooling needs of single duct decrease by about 30 % in Northern countries and increase by up to 25 % as compared to split equivalent installation in South European countries.

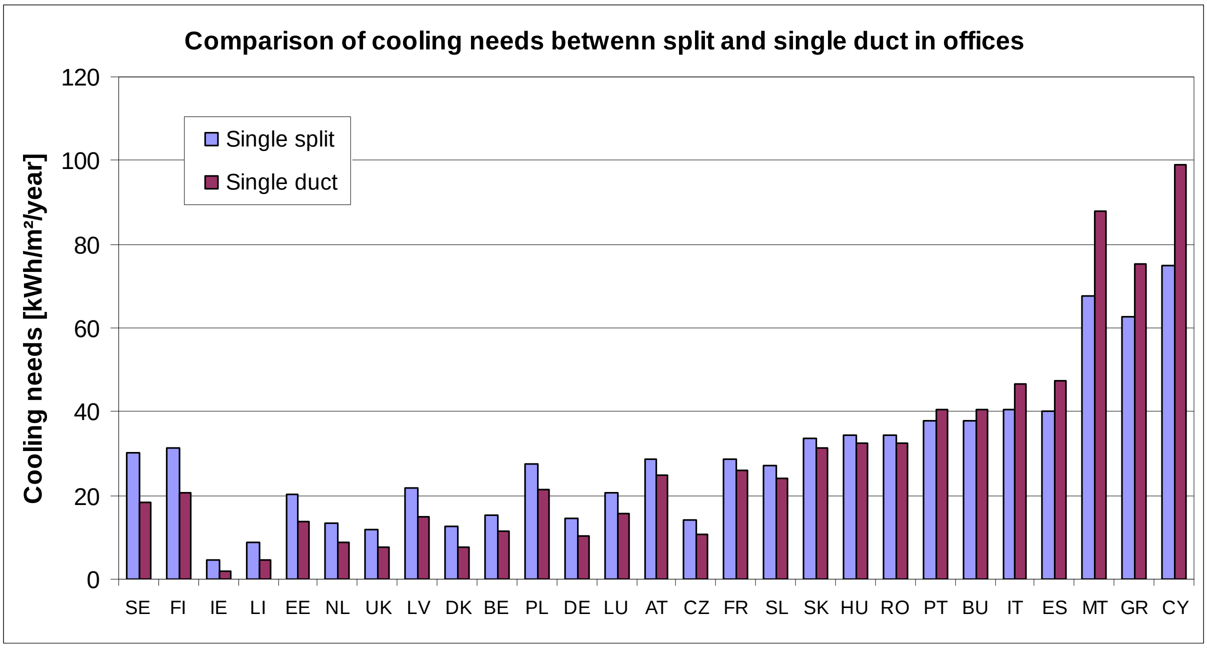


Figure 4-57: Comparison between cooling needs of a single duct air conditioner and a split air conditioner in office buildings

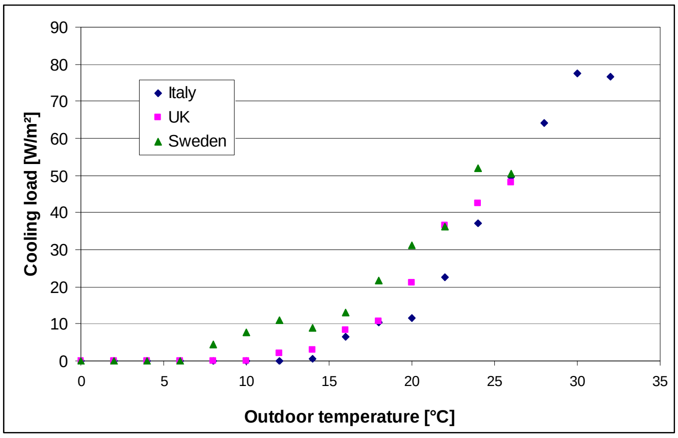
For warm weather conditions, at equal cooling capacity sizes as split air conditioners, single duct air conditioners cannot respect the cooling function as defined in task 1: “to maintain indoor air temperature at a given set point”.
To get comparable sizes, it is necessary to correct the rated cooling capacity of single duct for the infiltration of hot air. With typical unit characteristics, at 35 °C outdoor, it implies at least 60 % capacity loss as compared to EN 14511 rated capacity. This means that a unit of 2.2 kW in fact should be compared with a 0.6 kW split unit with a corrected EER inferior to 1 in T1 conditions (outdoor air 35 / recycled air 27 and 19 °C wet bulb). Only a difference of temperature outdoor / indoor of about 2 °C can be reached at 35 °C outdoor. When outdoor air temperature is lower than the set point, on the contrary, there is a free cooling effect that increases the cooling capacity and the EER as compared to rated present value. If cooling at 20 °C outdoor, the unit capacity and efficiency would increase by 25 %.

Load curves and energy weighting curves for three countries: Italy, UK and Sweden

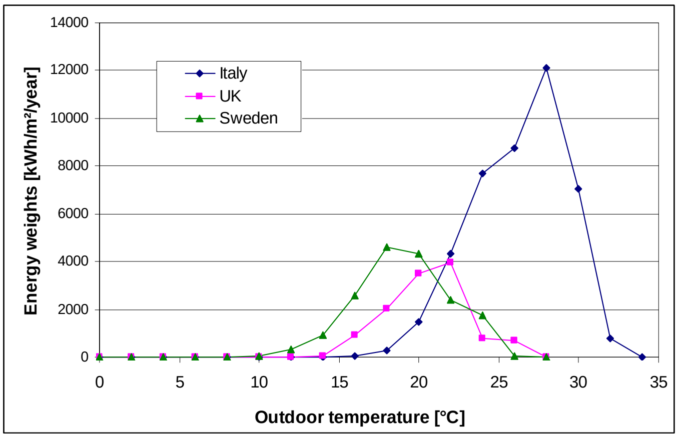
This sub section aims at giving an overview of our results in terms of heating and cooling needs by presenting the load curves and the energy distributions for three specific climates. Italy (Milan) was kept to represent warm climates whereas UK and Sweden are representative of moderate and cold climates.

All the cooling load curves from building simulations have been reduced to 19 points, one point by 2°C range of outdoor temperature. Thus, a typical load is associated to every temperature range. Furthermore, an energy weight is also associated by summing all the energy needs occurring in this range. The obtained curves are presented in Figure 4-58, Figure 4-59 and Figure 4-60.

All the heating load curves from building simulations have been reduced to 23 points, one point by 2°C range of outdoor temperature. Thus, a typical load is associated to every temperature range. Furthermore, an energy weight is also associated by summing all the energy needs occurring in this range. The obtained curves are presented in Figure 4-61, Figure 4-62 and Figure 4-63.

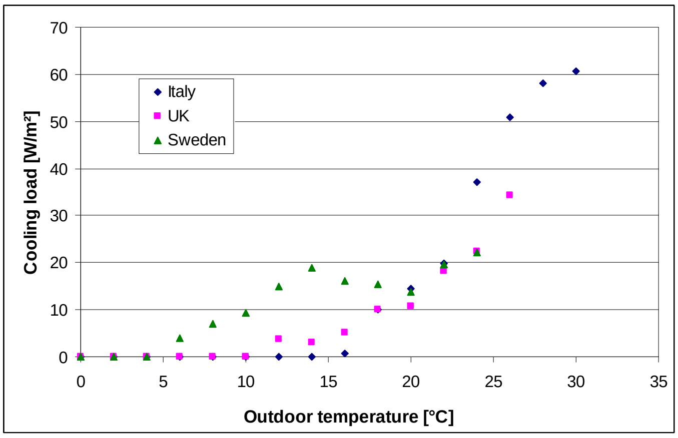


a)

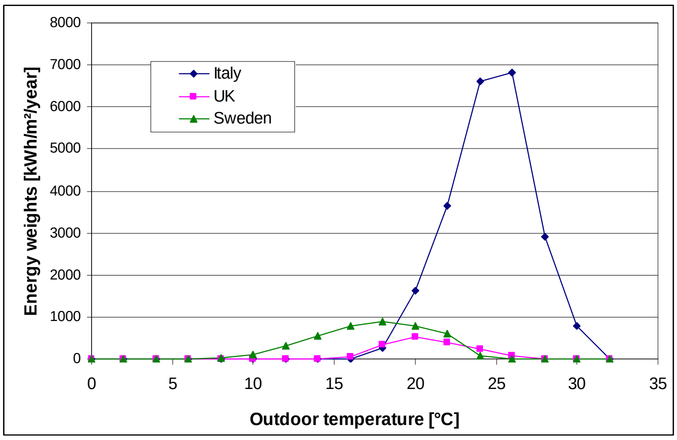


b)

Figure 4-58: a) Load curves in cooling mode for offices in the three countries – b) Weighting curves in cooling mode for offices in three countries



a)



b)

Figure 4-59: a) Load curves in cooling mode for residences in the three countries – b) Weighting curves in cooling mode for residences in three countries

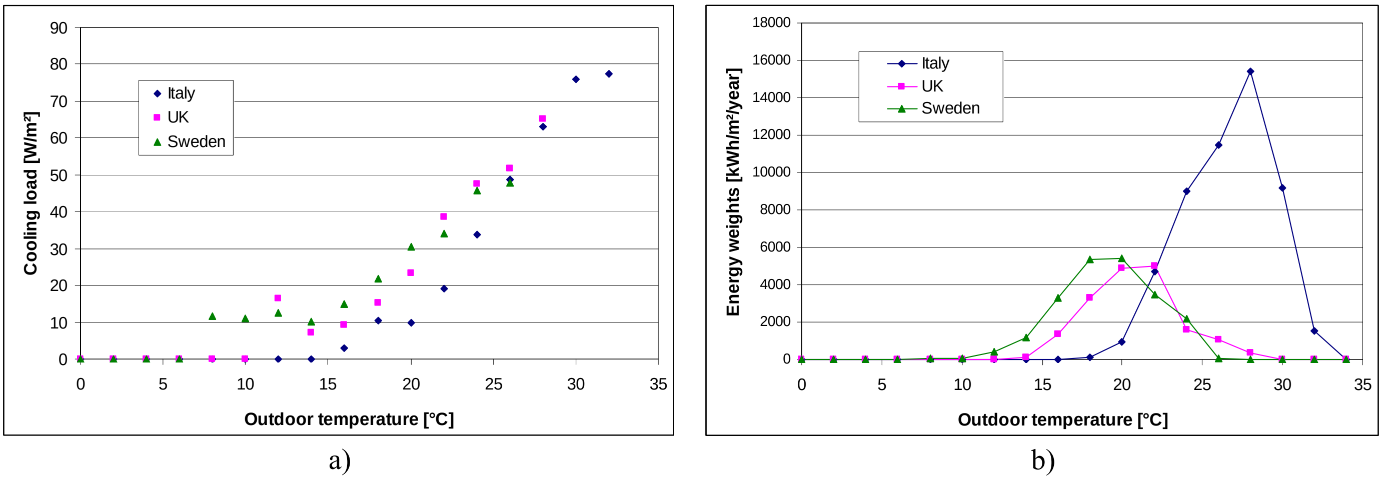


Figure 4-60: a) Load curves in cooling mode for retails in the three countries – b) Weighting curves in cooling mode for retails in three countries

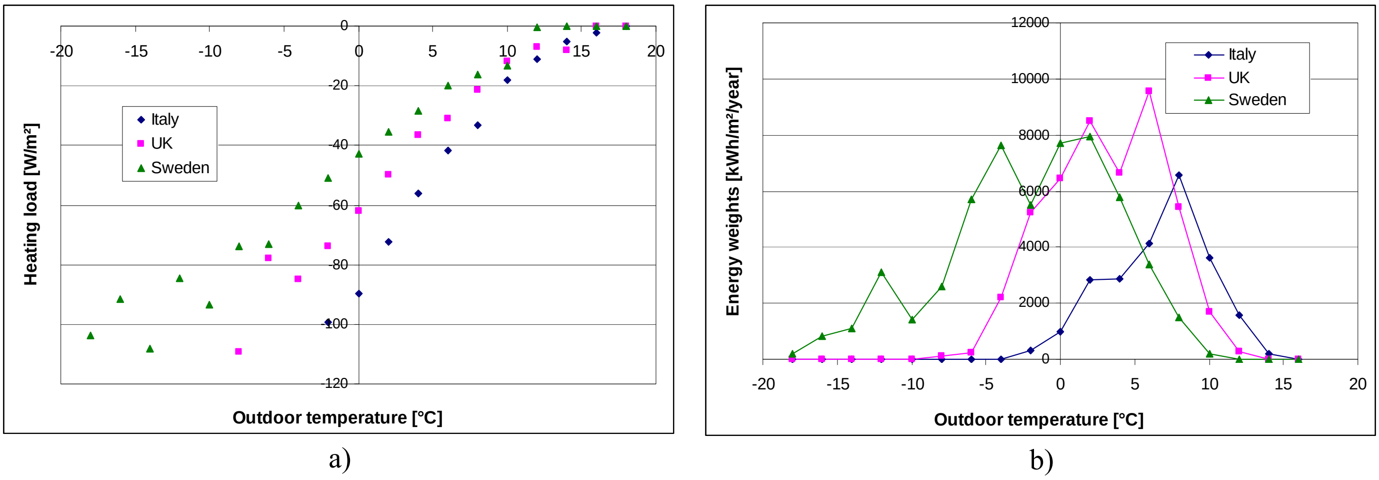


Figure 4-61: a) Load curves in heating mode for offices in the three countries – b) Weighting curves in heating mode for offices in three countries

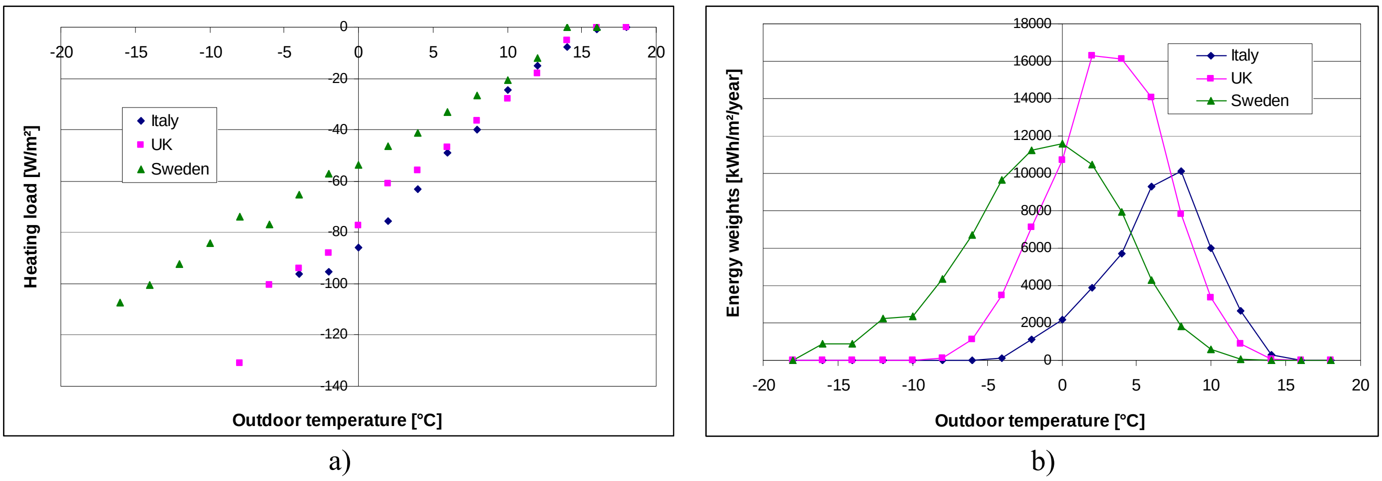
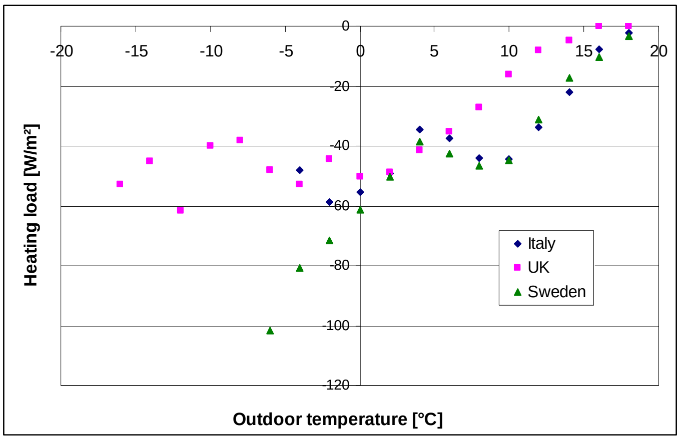
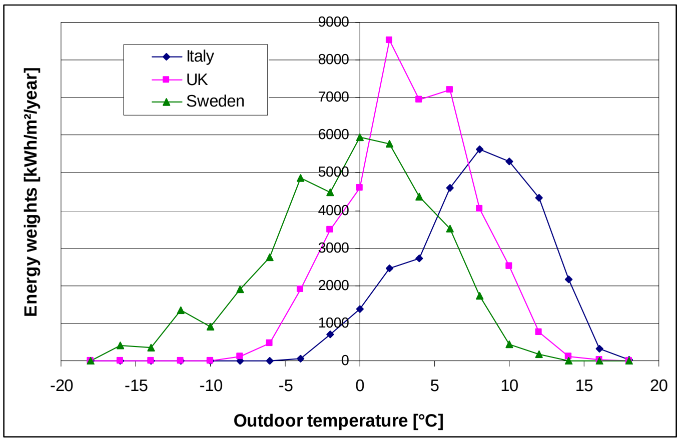


Figure 4-62: a) Load curves in heating mode for residences in the three countries – b) Weighting curves in heating mode for residences in three countries



a)



b)

Figure 4-63: a) Load curves in heating mode for retails in the three countries – b) Weighting curves in heating mode for retails in three countries

4.5.1.6 Sizing – Preliminary analysis

The systems were sized based on peak cooling and heating loads as determined by TRNSYS simulations, allowing for peak load to be met.

As European climates vary from that assumed by the standards, initial analysis was conducted using all climate zones, i.e. each European country is considered as a climate zone, and is represented by each climatic data.
A common indicator for the definition of climate zones is the heating and cooling degree-days of each country. These indicators are often used to represent climate zones for heating and cooling in the EU-25.
Variation in cooling and heating loads are not only caused by variation in climate, but also by variation in thermal building characteristics and variation in building operating conditions.
In this framework, final analysis was conducted using a number of " indicator" climate zones that reflect as well significant building characteristics.

Cooling only air conditioner

Sizing (defining the cooling capacity of the unit) is a crucial assessment when part load performances and full load performances differ largely. Perfect sizing can only be done using a dynamic simulation software.
This would have to be done in a try and correct process highly time consuming. Indeed, it would be necessary to find the correct climatic data, to model the building and its use, to perform a sensible capacity simulation with infinite available cooling capacity; then, an equipment would be chosen (with default performance curves when outdoor and indoor temperature conditions vary) and a first size identified. It would be then tested to make sure dehumidification loads are covered and corrected to get the right value.

Other intermediary methods are generally applied, simplified simulation methods with design climatic data for the location of the project, simplified thermal balance calculation methods or rules of thumb (W/m2 for a flat in Paris, an office in Stockholm ..). In that latter case, methods will vary according to each installer experience.

Maximal load in W/m2 occurs for outdoor temperature, indoor temperature and humidity ratio that are not the rating conditions. This peak value for off-design conditions is kept to size the unit. Hence, the maximal cooling capacity in simulations is kept as the value at rating conditions.

	Sizing for cooling only air conditioners [W/m²]					
	Old Offices	New Offices	Old Residences	New Residences	Old Shops	New Shops
AT	99	95	86	89	106	97
BE	69	72	58	61	85	82
DK	67	69	47	47	73	68
FI	72	81	71	73	77	73
FR	93	90	80	79	100	91
DE	91	97	79	82	97	91
GR	125	115	133	129	127	131
IE	64	70	36	51	71	69
IT	104	95	93	95	105	108
LU	90	88	70	84	97	90
NL	72	81	63	64	91	83
PT	131	121	143	135	116	120
ES	119	112	129	116	118	117
SE	77	83	66	66	80	74
UK	64	68	44	44	80	74
CY	125	124	123	125	127	132
CZ	74	93	51	64	74	74
EE	63	77	59	72	62	66
HU	96	110	80	86	95	90
LV	71	83	62	74	72	72
LI	68	89	45	63	69	73
MT	125	124	129	129	127	129
PL	65	76	39	53	66	64
SK	96	107	81	89	95	90
SL	89	99	67	76	78	80

Table 4-26: Sizes of cooling only split and moveable appliances in W/m2

Reversible air conditioner

There are substantial differences in the conditions of operation of reversible air conditioners, particularly in heating mode.

Because of large temperature differences there is a need in heating mode to consider several climatic zones. In each zones, sizing conditions of air to air heat pumps will vary. Heating zones have been designed as follows.

Since there is no widely accepted methodology, an economic optimization has been led to determine optimal sizing for all the countries for a set of simulations led by the university of Athens for 3 types of buildings (residential, office, commercial), new buildings and existing buildings.

In order to determine sizing temperatures for the different climatic areas in heating mode, a basic economic optimization has been carried out. The sizing temperature is a temperature under which the heat pump is not intended to cover all the needs, occupants using resistive heating instead. Indeed as the heating capacity increases with temperature, the lowest is the sizing temperature, the highest is the rated heating capacity and therefore the purchase cost of the heat pump. In the other hand, occupants are likely to pay less money for energy consumption with heat pump than with resistive heating. As a result, occupants have to find an optimal configuration between resistive heating and heat pump use.

The assumptions are the following ones:

It is assumed that the rated heating capacity and cooling capacity are linked with the following formula:

$$Q_h = \frac{4}{3.5} \cdot Q_c$$

Whatever the sizing temperature is, the RAC must match all the cooling needs and the rated cooling capacity is set to the maximal cooling load. Then a minimum heating capacity is calculated with the last equation.

The evolution of heating capacity according to outdoor temperature is assumed to be linear, capacity at –10 °C being 65 % of the capacity at 7°C. Heating needs that are higher then the heating capacity are matched by resistive heating.

Given a sizing temperature, it becomes possible to find out the rated heating capacity from the previous hypothesis and the heating loads and the minimum heating capacity. Thus the following values are known: rated heating and cooling capacities, loads that can be covered by the heat pump, loads that cannot be covered by the heat pump.
The economic optimization relies on the additional assumptions:

- Cost of electricity: 0.15 €/kWh
- Cost of the RAC: 180 to 300 €/kW
- Cost of a resistive heater: 50 to 60 €/kW
- COP is kept equal to 3 at 7 °C and decreases linearly – 2 %/°C
- Heating capacity decreases linearly with outdoor temperature, capacity at -10°C is 65 % of the capacity at 7 °C.
- Lifetime of the air conditioner: 12 years
- Energy consumption is not actualized (either energy price increases or cost of RAC decreases).

	Sizing temperature	**Countries**
Zone 1 - South	Perfect sizing in cooling mode	GR; PT; CY; MT; ES-Sevile; IT-Rome
Zone 2 - Oceanic	-3°C	BE ; IE ; IT; NL; ES; UK; FR
Zone 3 - Central	-7°C	AT ; DK ; DE; LU; CZ; HU ; PL ; SL ;SK ;LI
Zone 4 - North	-12°C	FI ; SE ; EE ; LV

Table 4-27: Sizing hypothesis of reversible split appliances

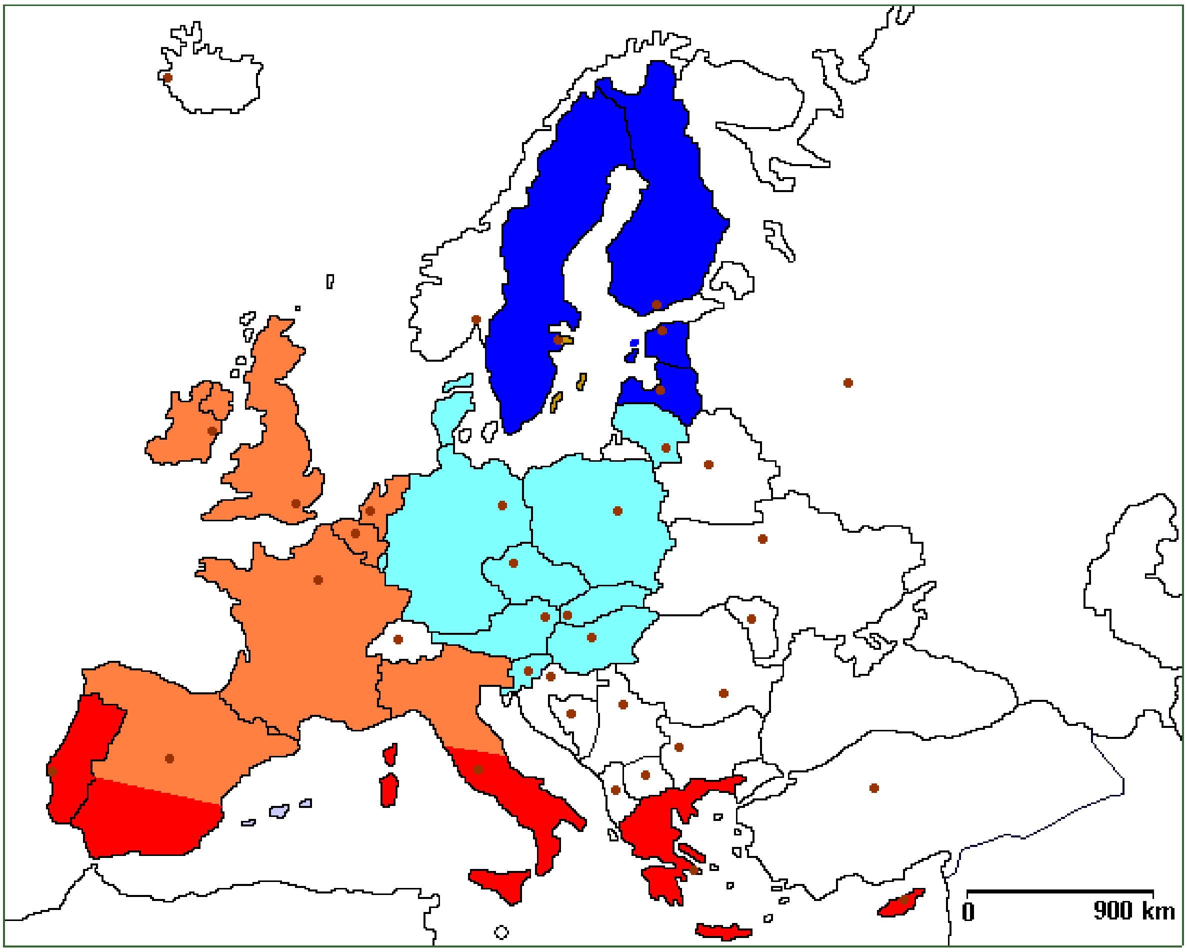


Figure 4-64: Sizing zones in Europe

Finally, rated capacities are summarized in the following table, both in heating and cooling modes:

	Rated cooling capacity [W/m²]						Rated heating capacity [W/m²]					
	Offices		Residences		Shops		Offices		Residences		Shops	
	New	Old	New	Old	New	Old	New	Old	New	Old	New	Old
AT	109	142	147	175	141	128	124	162	168	199	161	146
BE	87	111	117	138	117	135	100	127	133	158	134	154
DK	114	145	142	173	136	161	130	166	162	198	155	184
FI	158	173	163	183	175	195	181	198	186	209	200	223
FR	93	113	120	141	120	134	106	129	137	161	137	153
DE	119	149	147	176	147	167	136	170	168	201	169	191
GR	115	125	129	133	131	127	131	142	148	152	150	146
IE	87	104	111	130	121	138	99	118	127	148	138	158
IT	114	141	126	164	125	182	130	162	144	188	142	208
LU	104	135	150	214	152	167	119	155	171	245	173	191
NL	87	112	117	138	119	138	100	128	134	158	136	158
PT	121	131	135	143	120	116	139	150	154	163	137	132
ES	112	147	116	168	117	183	129	168	133	192	134	209
SE	137	152	163	183	167	189	157	174	186	209	191	216
UK	88	111	117	138	124	140	100	127	134	158	142	160
CY	124	125	125	123	132	127	142	142	142	141	151	145
CZ	111	144	126	166	135	180	127	165	143	190	154	206

EE	145	201	163	225	165	228	166	230	186	257	188	261
HU	110	141	121	162	117	163	125	161	139	185	134	186
LV	145	201	159	222	165	227	166	229	182	254	188	260
LI	104	141	125	169	119	166	119	161	143	193	136	189
MT	124	125	129	129	129	127	141	143	147	147	147	145
PL	112	145	128	169	123	170	128	166	146	193	141	195
SK	107	134	121	161	116	158	123	153	138	184	133	181
SL	100	129	121	161	113	155	114	147	138	184	130	177

Table 4-28: Sizes of reversible split appliances, in cooling and heating modes (the ratio between heating and cooling rated capacity is constant)

The results are presented hereafter for existing offices and new retails. The maximum resistive part is reached for New Retails in Finland: 34 %. However, a weighting average only gives a value of 0.25 kWh (total average heat demand is of 85.5 kWh/m²/y).

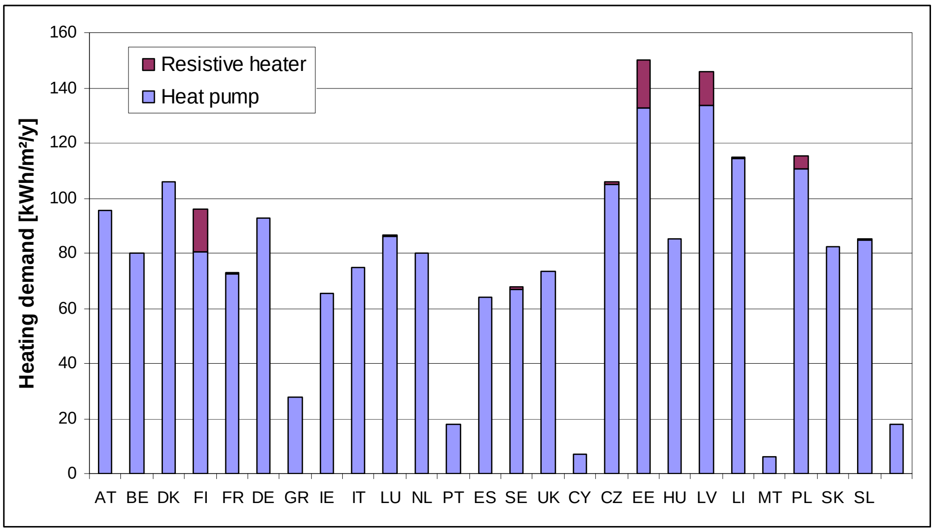


Figure 4-65: Resistive heating energy share compared to total heating demand in existing offices

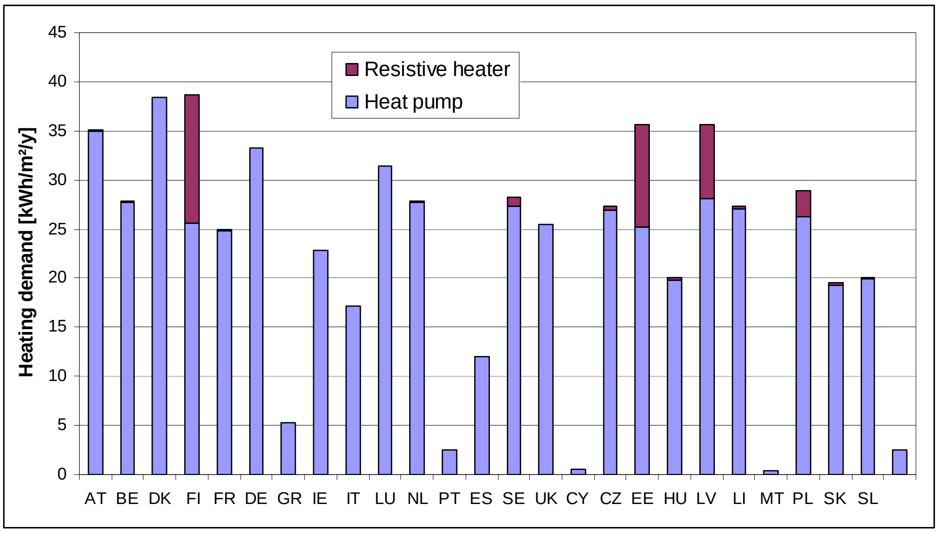


Figure 4-66: Resistive heating energy share compared to total heating demand in new retails

4.5.2 Energy consumption

Cooling season is assumed to be May 1st to September 31st. Heating season is the remaining period of the year.
For the whole Europe, sales weighted average values for EU 27 taking into account the market representation presented in Task 2 are computed hereafter.

4.5.2.1 Analysis of energy consumption results

Cooling only split systems

Energy consumptions of the 3.5kW cooling only split system range from less than 100 kWh/year to about 1000 kWh/year according to the type of building, the building age and the outdoor climate. Regarding the 7.1 kW cooling only split system, energy consumptions vary from about 200 kWh/year to 2000 kWh/year.
Parasitic electric consumptions (stand-by, thermostat-off) represent a non-negligible part of to total consumption : about 6-7% in average for retails and offices (3 % for Southern countries), 25 % in average for residences (7% for Southern countries). Figure 4-67 and Figure 4-68 are given as examples.

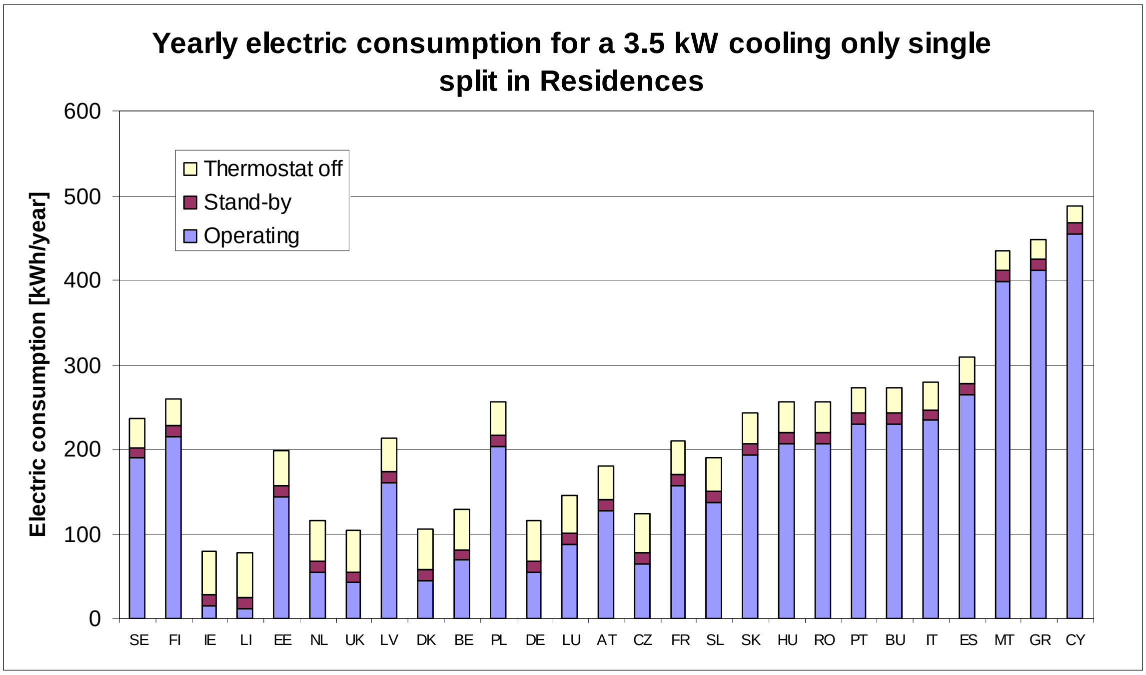


Figure 4-67: Yearly electric consumption for a 3.5 kW cooling only split system in residences

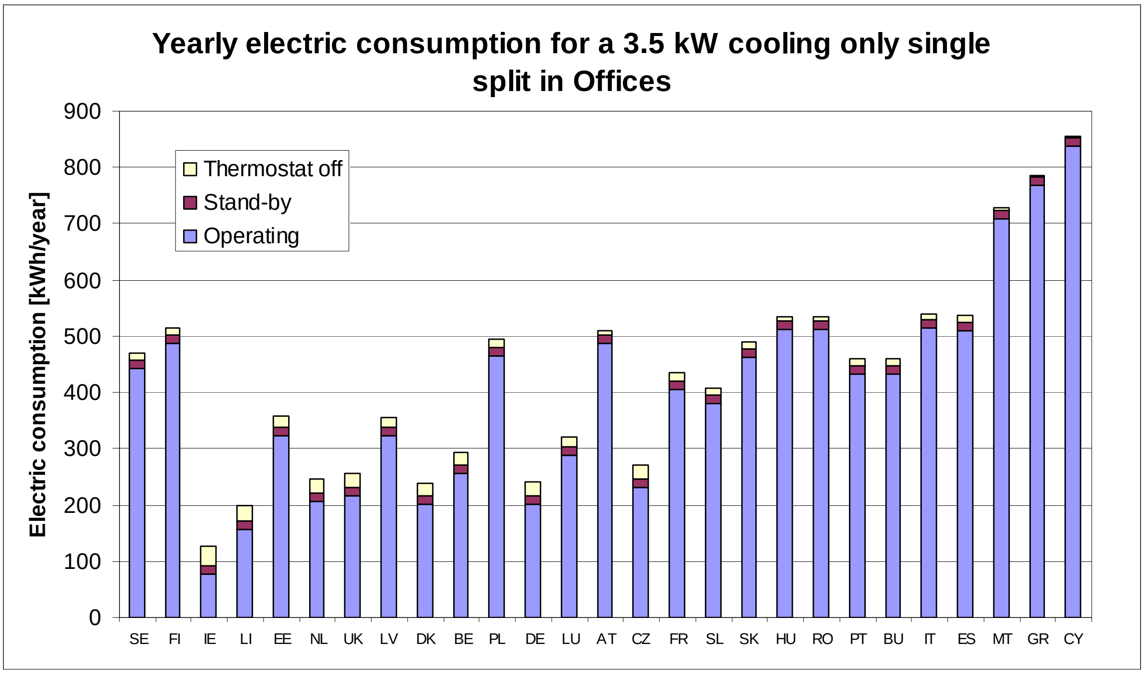


Figure 4-68: Yearly electric consumption for a 3.5 kW cooling only split system in offices

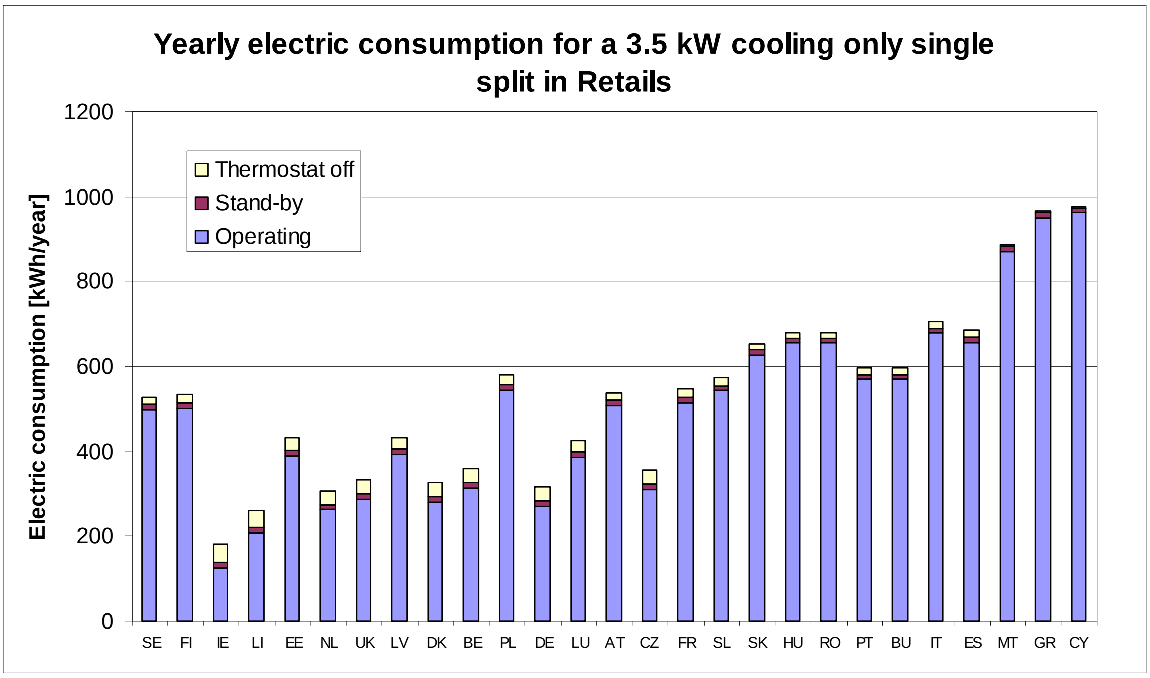


Figure 4-69: Yearly electric consumption for a 3.5 kW cooling only split system in retails

Reversible split systems

Energy consumptions of the 3.5 kW reversible split system range from less than 700 kWh/year to about 2000 kWh/year according to the type of building, the building age and the outdoor climate. Regarding the 7.1 kW reversible split system, energy consumptions vary from about 1400 kWh/year to 3800 kWh/year. In most countries (all except Portugal, Bulgaria, Malta, Greece and Cyprus), electric consumption in heating mode is more important than the consumption in cooling mode.

Parasitic electric consumptions (stand-by, thermostat-off) represent a non-negligible part of the total consumption : about 10 % in average for retails (11 % for Southern countries), 15 % in average for offices (25 % for Southern countries), 11 % in average for residences (17% for Southern countries). Figure 4-70 and Figure 4-71 are given as examples.

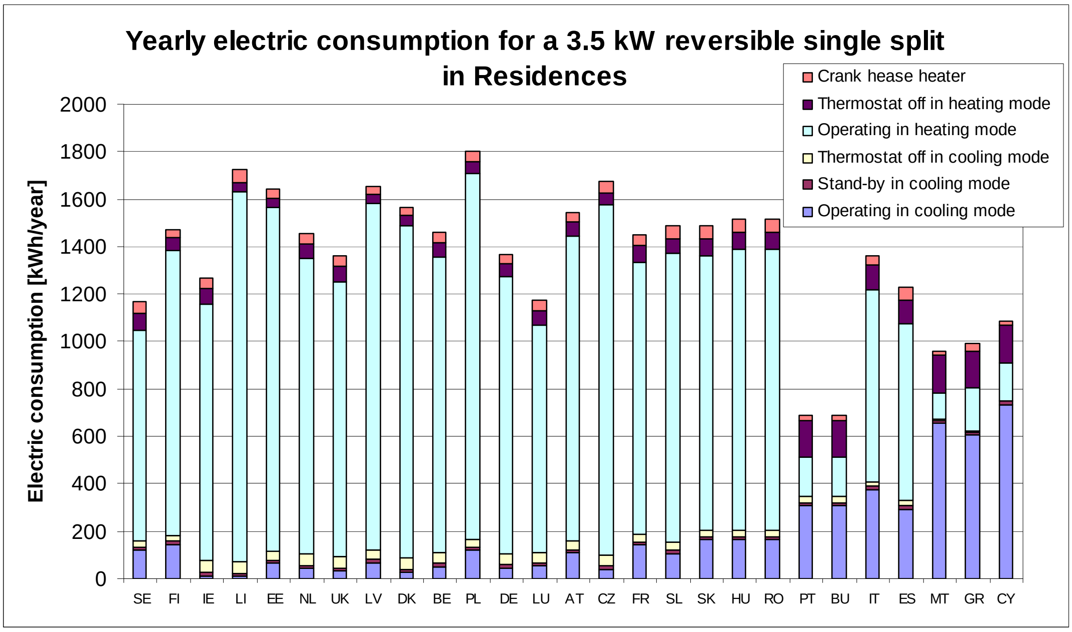


Figure 4-70: Yearly electric consumption for a 3.5 kW reversible split system in offices

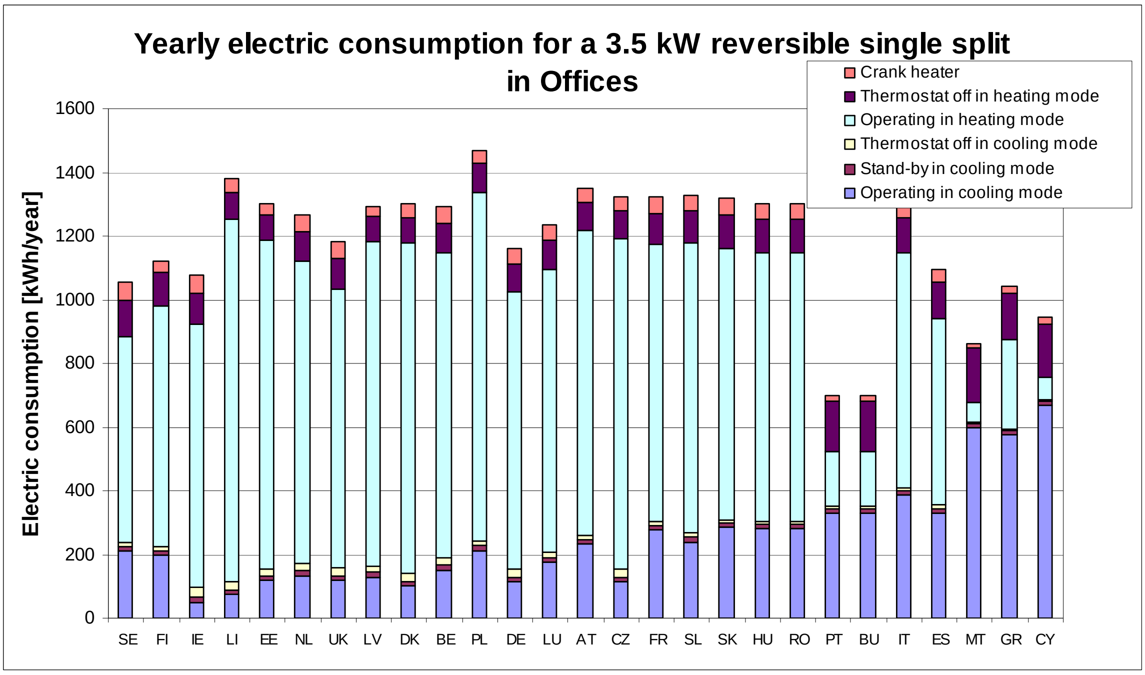


Figure 4-71: Yearly electric consumption for a 3.5 kW reversible split system in Residences

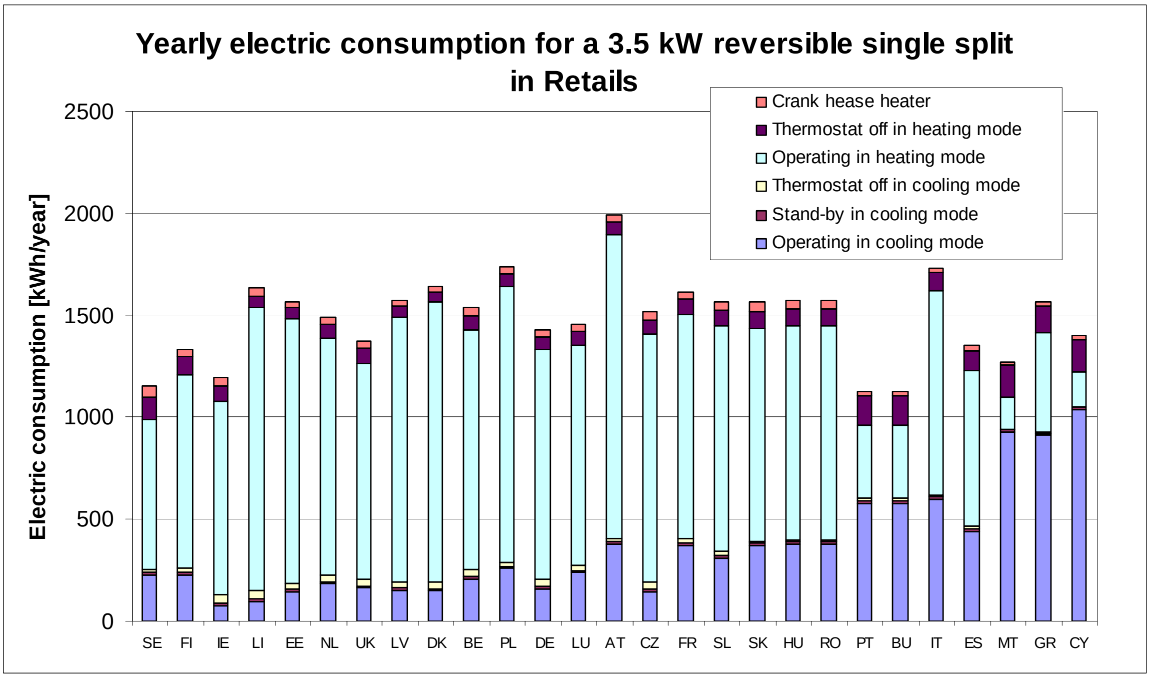


Figure 4-72: Yearly electric consumption for a 3.5 kW reversible split system in Retails

Single Duct system

Energy consumptions of the 2.2kW single duct system range from less than 150 kWh/year to about 1100 kWh/year according to the type of building, the building age and the outdoor climate.
Parasitic electric consumptions (stand-by, thermostat-off) represent a non-negligible part of to total consumption : about 17 % in average for retails (5 % for Southern countries), 20 % in average for offices (7 % for Southern countries), 45 % in average for residences (20 % for Southern countries). Figure 4-73 and Figure 4-74 are given as examples.

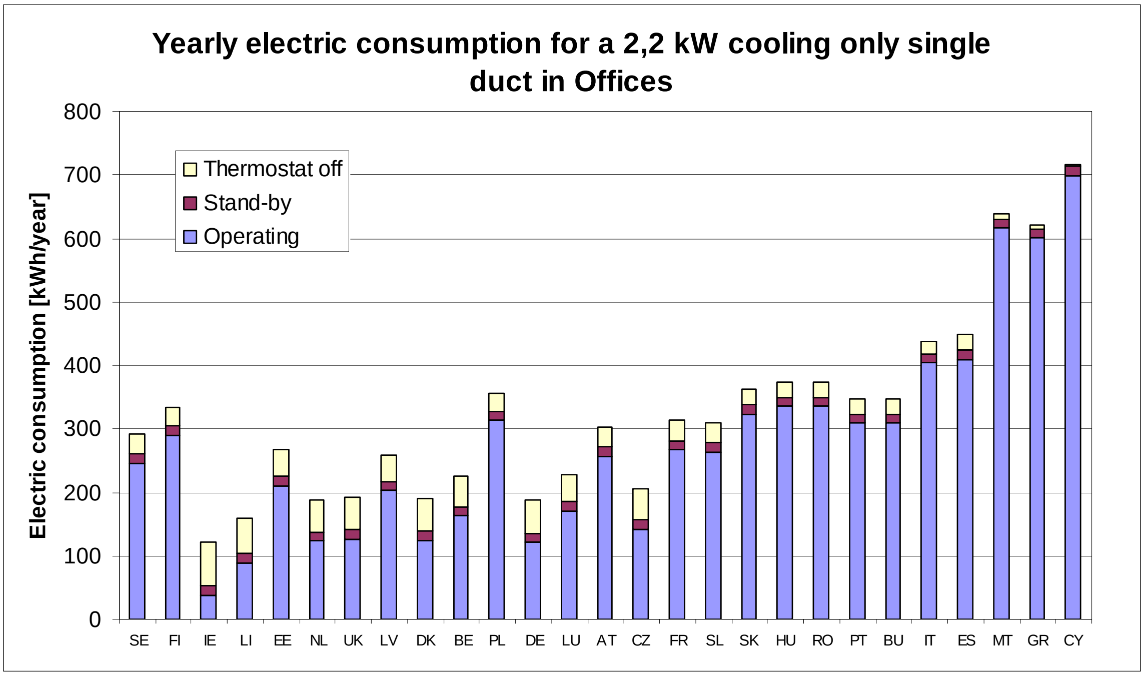


Figure 4-73: Yearly electric consumption for a 2.2 kW single duct system in Offices

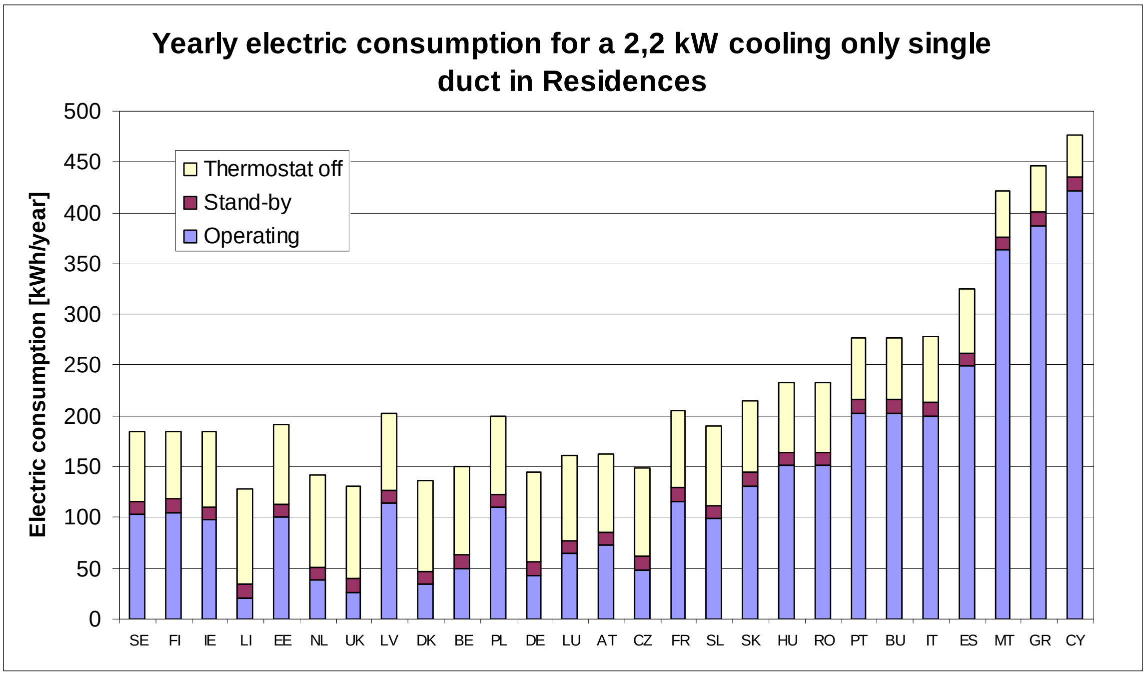


Figure 4-74: Yearly electric consumption for a 2.2 kW single duct system in Residences

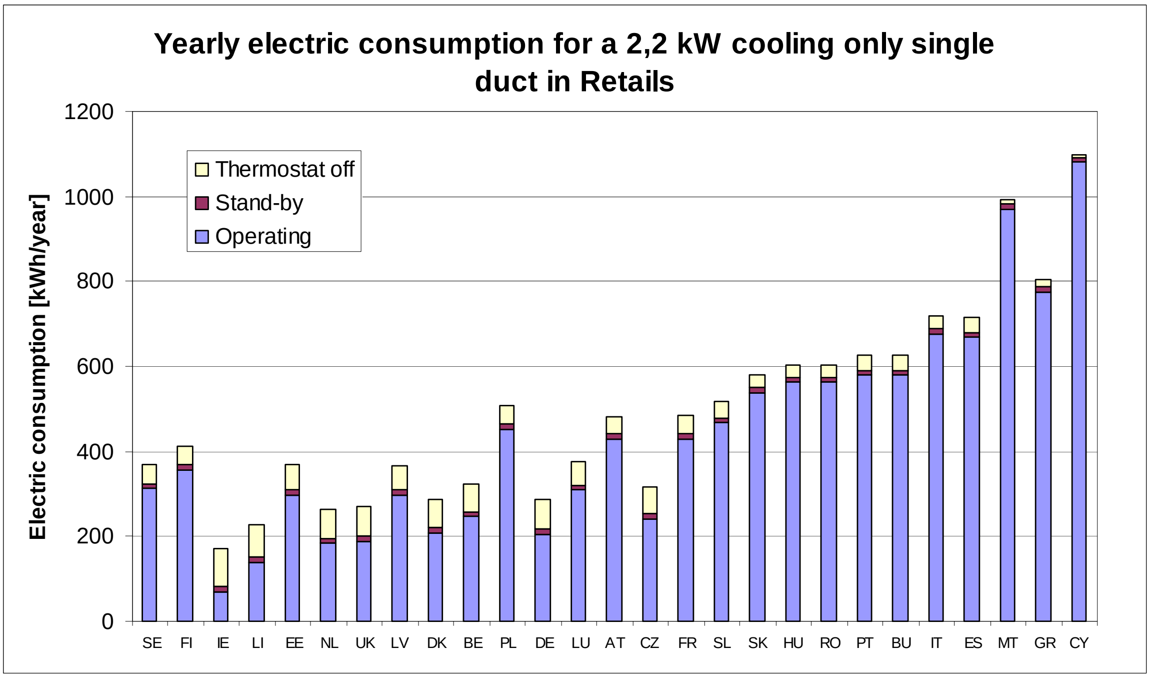


Figure 4-75: Yearly electric consumption for a 2.2 kW single duct system in Retails

4.5.2.2 Sales weighted average results for EU 27

Heating and cooling sizing load consumption EU 27

Base cases energy consumption [W/m²]	Cooling only single split [3.5 kW]	Reversible single split [3.5 kW]	Cooling only single split [7.1 kW]	Reversible single split [7.1 kW]	Single duct [2.2 kW]
Cooling sizing load	109.4	122.6	109.4	122.6	107.5
Heating sizing load	-	140.1		140.1	-

Heating and cooling energy consumption EU 27

Heating and cooling needs and energy consumption have been calculated from simulations for every country and then sales weighted with cooling capacity sales of the year 2005. This gives the average EU 27 values in the tables below.

Base cases energy consumption [kWh/m²/year]		Cooling only single split [3.5 kW]	Reversible single split [3.5 kW]	Cooling only single split [7.1 kW]	Reversible single split [7.1 kW]	Single duct [2.2 kW]
Annual electric consumption for cooling		11.8	13.1	13.6	14.5	15.2
Annual electric consumption for heating	**Thermodynamic heating**	-	33.3		34.3	
	Resistive heating [kWh/m²/year]	-	0.25	-	0.25	

Table 4-29: Average EU 27 cooling and heating energy consumption of base cases, kWh/m2/year

Differences in energy consumption of the two different sizes of split units come from differences in EER and COP of the base case units. Electric heating impact on overall heating energy consumption is low (about 1 % of electric heating by the heat pump) in average because sales are mostly in Southern countries.
The same results are expressed hereafter for the base case units (with different capacities) and by kW of cooling capacity.
- Equivalent hours of use are the ratio between annual electric consumption and rated capacity in standard conditions.
- Seasonal performance during operating hours is the ratio of heating or cooling output over the electric consumption for heating or cooling.
- Equivalent rated electric power is the ratio between rated cooling capacity and the seasonal efficiency.

Base cases energy consumption FOR ACTIVE MODE HEATING AND COOLING		Cooling only single split [3,5kW]	Reversible single split [3,5kW]	Cooling only single split [7,1kW]	Reversible single split [7,1kW]	Single duct [2,2 kW]
COOLING NEED kWh/m2/y		33	38,8	33	38,8	33,5
Annual electric consumption for cooling kWh/m2/y		11,8	13,1	13,6	14,5	15,2
Cooling mode	**Equivalent hours of use [h]**	301,6	316,5	301,6	316,5	311,6
	Seasonal efficiency	2,80	2,96	2,43	2,68	2,20
	1) KWh / unit / year	377,5	374,0	882,6	839,7	311,1
	2) Per kW cooling	107,9	106,9	124,3	118,3	141,4

HEATING NEED kWh/m2/y			85,5		85,5	
Annual electric consumption for heating	**Thermodynamic heating kWh/m2/y**		33,3		34,3	
	Resistive heating [kWh/m²/year]		0,25		0,25	
Heating mode	**Equivalent hours of use [h]**		610,3		610,3	
	Thermodynamic SCOPon		2,57		2,49	
	SCOPon		2,55		2,47	
	1) KWh / unit / year		1065,3		2226	
	2) Per kW cooling		271,4		279,6	

Table 4-30: Average EU 27 cooling and heating energy consumption per unit, and kW cooling

Additional consumptions EU 27

In order to define a final yearly performance we have to take into account parallel consumptions:

- **Active mode – thermostat on**: the compressor is on; energy consumption has been reported above.
- **Active mode - thermostat off**: The air conditioner is operational in heating or cooling mode but inside temperature is lower (higher in heating mode) than the set point. The impact of thermostat-off mode is included in the cycling low and additionally in the thermostat-off consumption corresponding to hours with no cooling or heating load while cooling (or heating) is required by the user.
- **Passive standby mode**: the air conditioner is not operational; it can be reactivated either by remote control or by timer. This mode corresponds to hours with no occupancy in the building during the cooling or heating season.
- **OFF mode**: moveable air conditioners are unplugged; split air conditioner indoor units are off but outdoor units that are hardwired still may draw current for the crankcase, depending on its control mode.

As mentioned in task 3, the crankcase heater may be controlled as a function of outdoor air temperature. Because of this control, it is necessary to include a specific post of consumption for this function that can overlap any other mode except the active mode – thermostat-on.

	Electric consumption for cooling only (<6kW) [W]	**Electric consumption for cooling only (>6kW) [W]**	**Electric consumption for reversible (<6kW) [W]**	**Electric consumption for reversible (>6kW) [W]**	**Single duct**
Active mode Thermostat-off mode	36 W Electronics: 6 W Indoor fan: 30 W	66 W Electronics: 6 W Indoor fan: 60 W	36 W Electronics: 6 W Indoor fan: 30 W	66 W Electronics: 6 W Indoor fan: 60 W	66 W Electronics: 6 W Fan: 60 W (*)
Standby-mode	Electronics: 6 W	Electronics: 6 W	Electronics: 6 W	Electronics: 6 W	Electronics: 6 W
OFF mode	6 W	6 W	< 6 W (indoor unit is switched off)	< 6 W (indoor unit is switched off)	0 W
Crankcase heater	No	No	Crank. heater: 30 W	Crank. heater: 70 W	No

(*) Generally, only one centrifugal fan is used for single duct air conditioners.

Table 4-31: Power drawn by base case air conditioners when compressor is off

Hours of operation in the different modes		Cooling only single split [3.5kW]	Reversible single split [3.5kW]	Cooling only single split [7.1kW]	Reversible single split [7.1kW]	Single duct
Cooling mode	Operating hours [h]	747.2	888.6	747.2	888.6	679.7
	Thermostat off [h]	643.9	502.5	643.9	502.5	711.4
	Stand-by hours [h]	2279.9	2279.9	2279.9	2279.9	2279.9
	Off-mode hours [h]	5089		5089		
Heating mode	Operating hours [h]		2292.2		2292.2	
	Thermostat off [h]		2796.8		2796.8	
	Stand-by hours [h]		0		0	
	Crankcase heater		1163		1163	

(*) During the heating period, it is assumed that heating is always required by the end user, whatever the set point may be.
(**) Crankcase heater is supposed to be turned off when outdoor air temperature reaches 10 °C.
(***) for cooling only split units, majority being hard wired, standby period extends to the whole year and a 6 W consumption is kept following task 3 results.

Table 4-32: Operating hours by mode

Total energy consumption EU 27

Energy consumption per unit / year KWh/y		Cooling only single split [3,5kW]	Reversible single split [3,5kW]	Cooling only single split [7,1kW]	Reversible single split [7,1kW]	Single duct [2,2 kW]
Cooling mode	Compressor on	377,5	374,0	882,6	839,7	311,1
	Thermostat off	23,2	18,1	42,5	33,2	47,0
	Stand-by	13,7	13,7	13,7	13,7	13,7
	Off mode	30,5	0,0	30,5	0,0	0,0
Heating mode	Compressor on + electric resistance		957,9		1997,5	
	Thermostat off		100,7		184,6	
	Stand-by		0		0	
	Crankcase heater		34,9		81,4	
TOTAL per unit in kWh/y		445	1499	969	3150	372

Energy consumption per kW cool / year	Cooling only single split	Reversible single split	Cooling only single split	Reversible single split	Single duct

KWh/year		[3,5kW]	[3,5kW]	[7,1kW]	[7,1kW]	[2,2 kW]
Cooling mode	Compressor on	107,9	106,9	124,3	118,3	141,4
	Thermostat off	6,6	5,2	6,0	4,7	21,3
	Stand-by	3,9	3,9	1,9	1,9	6,2
	Off mode	8,7	0,0	4,3	0,0	0,0
Heating mode	Compressor on + electric resistance		273,7		281,3	
	Thermostat off		28,8		26,0	
	Stand-by		0,0		0,0	
	Crankcase heater		10,0		11,5	
TOTAL per kW in kWh/y		127,1	428,3	136,5	443,7	169,0

Table 4-33: Average EU 27 total energy consumption per unit, and kW cooling

Operation of crankcase has been assumed to be controlled at 10 °C. With control at 0 °C, hours of operation could be divided by 10, while for 20 °C set point, it is multiplied by 2 and that without control, electric crankcase heater operations would be about 5500 hours yearly, multiplying related energy consumption by nearly a factor 4. In those conditions, its energy consumption would be about half the electric power required for heating.

Figures in the tables before should be used with caution to compare the performances of the different base cases since figures are weighted average and can translate different market shares in different countries, sectors and new or existing buildings.

SEER and SCOP calculation for the five base cases

SEER and SCOP indices have been calculated for each base case in operating mode (cooling or heating needs divided by the electric consumption in operating mode) but also by taking into account parasitic energy consumptions (cooling or heating needs divided by the total electric consumption). Results are given in Figure 4-76 and Figure 4-77. It appears that SEER indices in operating mode range from 2.2 for the Single Duct to 3.0 for reversible split of 3.5 kW. Parasitic consumptions lies between 3 % and up to 10 % for the Single Duct base case.

Regarding the SCOP indices, they are equal to 2.63 and 2.56 in operating mode and to 2.17 and 2.03 when taking into account parasitic consumptions (a drop of about 17 and 20 %). However, it must be kept in mind that this depends on how cooling and heating seasons are defined. Some of the parasitic consumptions occurring in mid-season could have been ascribed to the cooling mode instead of heating mode and included in SEER. This means that parasitic consumptions of reversible units should be considered together and not separately.

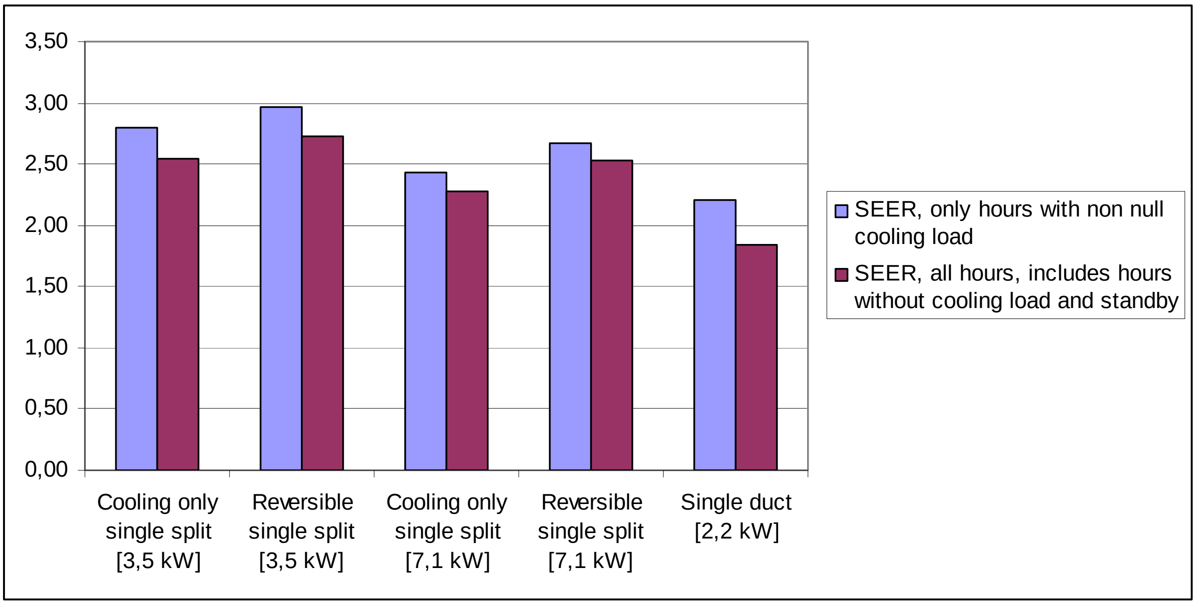


Figure 4-76: SEER for the five base cases

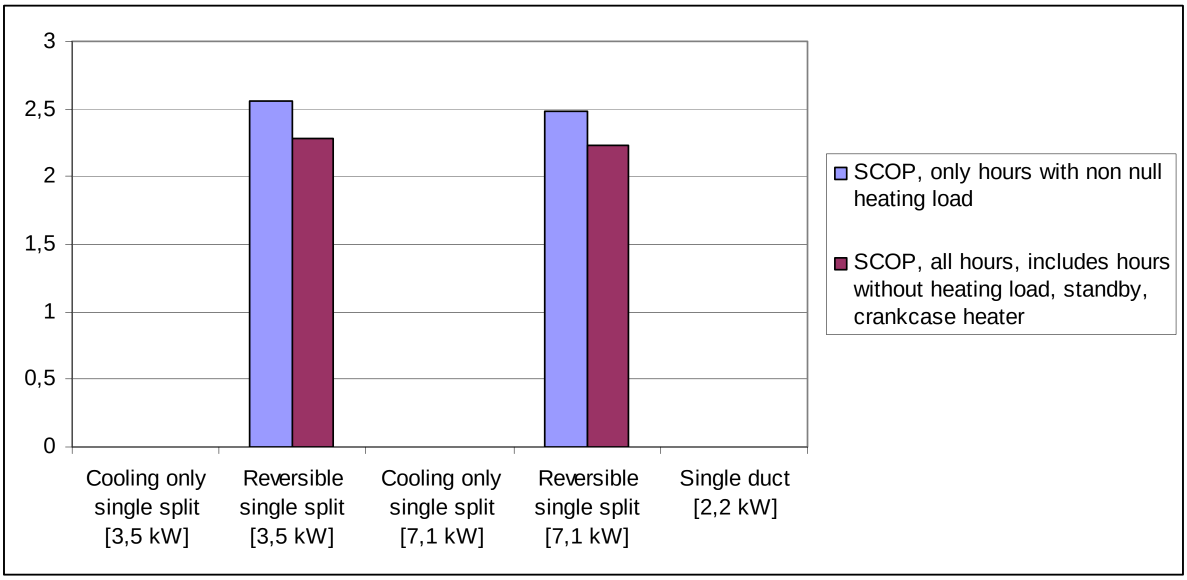


Figure 4-77: SCOP for the two reversible base cases

4.5.2.3 Seasonal performance indices

Coordination has been ensured with CEN TC 113 WG 7 working on the revision of the "technical standard" CEN/TS 14825 in order to have compatible values in this study and in the standard. More detail is available in appendix B.

We use hereafter load ratios and weighting factors as computed for sales weighted average EU 27 load curves.

In heating mode, a first set of reduction coefficients has been proposed with preliminary sizing hypothesis and temperatures of –7, -2, 2 and 7. Nevertheless, because of potential testing problems at part load and under frost conditions, it was decided to change the reference points to –7 °C, 2 °C, 7 °C and 12 °C and also sizing was moved to 2 °C in average instead of –2 °C.

Calculation of the $SEER_{on}$

$$SEER_{on} = \frac{1}{\frac{A}{EER_A} + \frac{B}{EER_B} + \frac{C}{EER_C} + \frac{D}{EER_D}}$$

Calculation of the $SCOP_{on}$

$$SCOP_{on} = \frac{1}{\frac{A}{COP_A} + \dots + \frac{H}{COP_H}}$$

$SEER_{on}$ - Europe		
Temperatures	Part Load Ratio % (1)	Weighting coefficients
35	100	9 %
30	75	30 %
25	50	37 %
20	25	24 %

(1) 100 % being the full load capacity at 35 °C

$SCOP_{on}$ - Europe		
Temperatures	Part Load Ratio % (2)	Weighting coefficients
-7	155%	4%
-7	85%	5%
-2	130%	9%
-2	70%	12%
2	100%	18%
2	50%	16%
7	55%	25%
7	30%	11%

(2) 100 % being the full load capacity at 2°C

When load is superior to the capacity of the unit (at –7 °C and/or –2 °C), the unit is supposed to operate at maximum capacity and a global heating resistance correction is then computed (see in the following paragraph).

Single duct units have the specificity to operate at equal temperature conditions at both condenser and evaporator sides. Different set points have been set for the different types of buildings (residential, office, shops) resulting in different average temperature conditions. Although in warmer climates on a hot day, set point may not be maintained properly indoor, average temperature are in general not far from the required temperatures. As a result of the repartition of the product sales in building and different climates, average operating conditions are 25.5 °C indoor with 65 % humidity ratio and 50 % load ratio. For computing a seasonal performance index, standard EN 14511 temperature conditions and 50 % part load ratio can be kept.

Resistive heating impact on the SCOP in heating active mode

Following comments made by the stakeholders, sizing hypothesis were revised in order to assess the resistive heating part to complete the SCOP index shown before. In colder climates, sizing practice of – 7 °C rather than –12 °C seems to be more common, for the Central area, -3 instead of – 7 °C and in the Oceanic zone, rather 2 than –2 °C. This led to corrected sizing values for all simulations.

Results are presented hereunder for existing offices and new retails. The maximum resistive part is reached for New Retails in Finland: 36 %. Weighted average value is about 1.1 kWh/m^2/year over 85.5 kWh/m^2/year against 0.25 before.

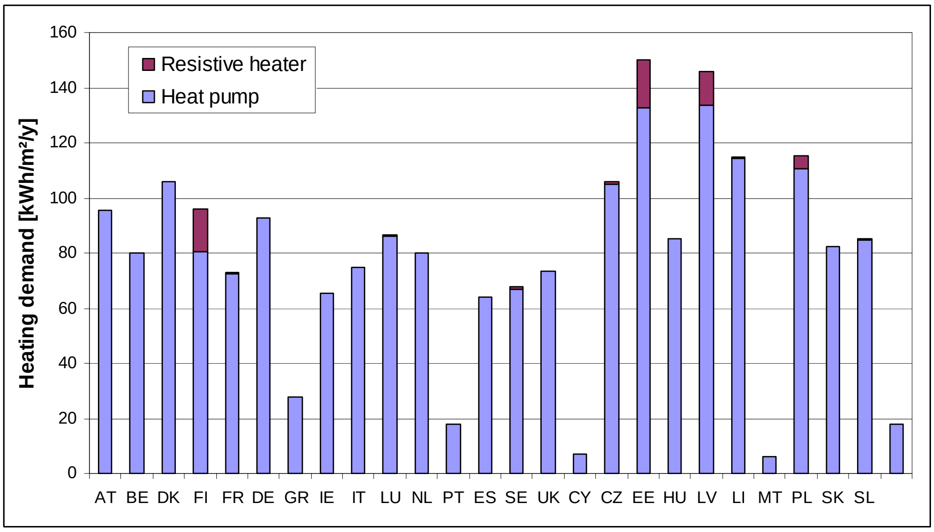


Figure 4-78: Resistive part compared to total heating demand in existing offices

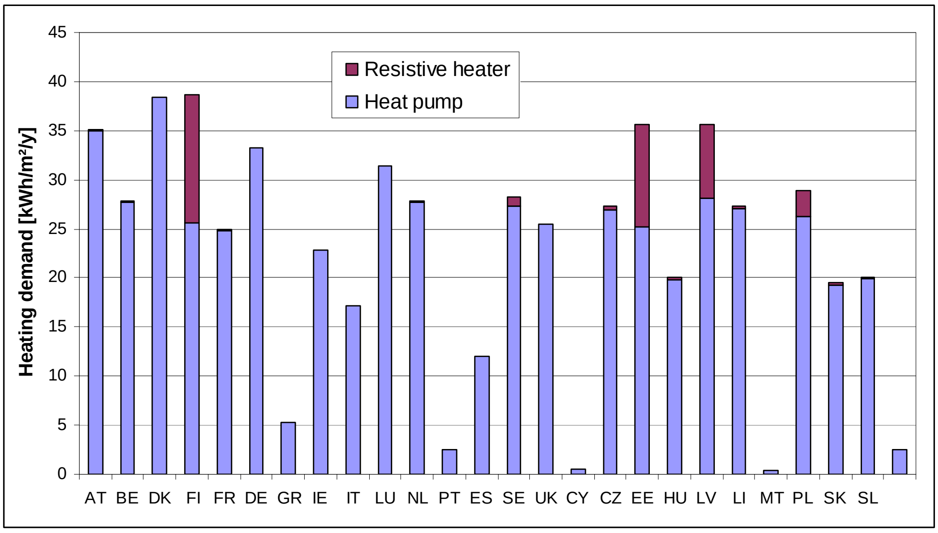


Figure 4-79: Resistive part compared to total heating demand in new retails

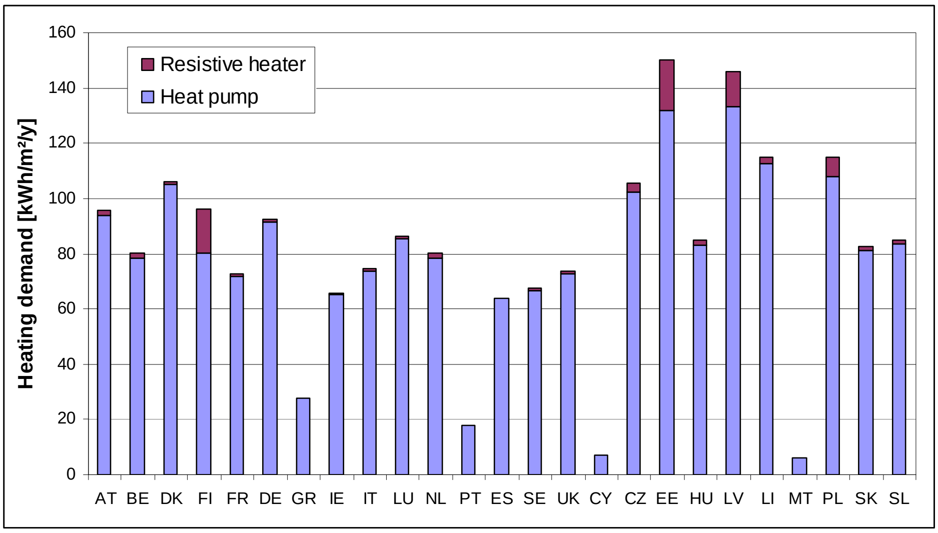


Figure 4-80: Resistive part compared to total heating demand in existing offices

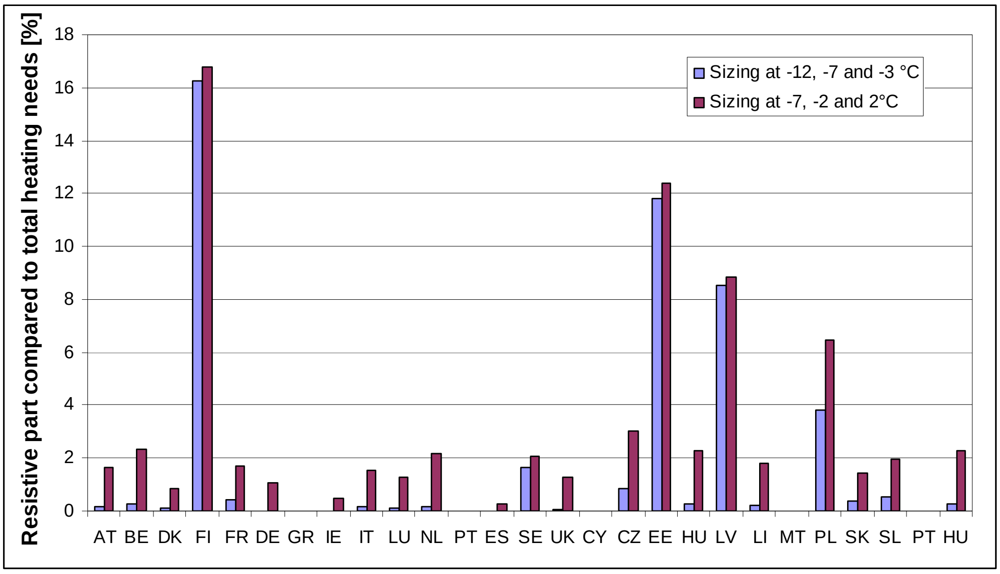


Figure 4-81: Resistive part compared to total heating demand in existing offices

What follows aims at determining the likely impact of the resistive heating on the heating SCOP and how it can be taken into account.

The share of the heating demand that must be matched by resistive heating is affected, at a given sizing condition and for a given load curve, by the decrease of the heating capacity with outdoor air temperature and the minimum outdoor air temperature until which the heat pump can work. It has been supposed until now that the base case may reach –15 °C and looses 35 % capacity at – 7 °C.

The following table gives the resistive part according to the ratio between heating capacity at –7 °C and capacity at 7 °C noted P(-7)/P(7). As an example, the corrected SCOP is given for a heat pump with SCOP without resistive heating equal to 2.7.

Capacity slope (P(-7)/P(7))	Heating demand (total)	Resistive demand	SCOP of the heat pump	SCOP of the system (HP + resistive heating)
0,75	85,5	0,70	2,7	2,66
0,7	85,5	0,86	2,7	2,65
0,65	85,5	1,03	2,7	2,65
0,6	85,5	1,21	2,7	2,64
0,55	85,5	1,42	2,7	2,63

Table 4-34: Impact of the capacity decrease between –7 °C and + 7 °C on resistive heating energy, capacity slope

The following table gives the resistive part according to the temperature under which the heat pump does no longer operate (P(-7)/P(7)=0.75). As an example, the corrected SCOP is given for a heat pump whose SCOP without resistive heating is 2.7.

Minimum temperature	Heating demand (total)	Resistive demand	SCOP of the heat pump	SCOP of the system (HP + resistive heating)
- 15	85.50	0.70	2.7	2.66
- 12	85.50	0.84	2.7	2.66
- 9	85.50	1.38	2.7	2.63
- 7	85.50	2.16	2.7	2.59

Table 4-35: Impact of the minimum temperature on resistive heating energy, minimum temperature

If we take a ratio of 0.55, it appears that the corrected SCOP differs from the first one by up to 11 %.

Minimum temperature	Heating demand (total)	Resistive demand	SCOP of the heat pump	SCOP of the system (HP + resistive heating)
-15	85.50	1.42	2.7	2.63
- 12	85.50	1.54	2.7	2.62
- 9	85.50	2.01	2.7	2.60
- 7	85.50	2.73	2.7	2.56

Table 4-36: Impact of the minimum temperature on resistive heating energy, minimum temperature

It appears that even if most of the sales are sold in southern Europe, resistive heating cannot be completely neglected. As a result, this is taken into account in the yearly consumption and in a yearly performance index.

For the new sizing performed above, the average share of resistive heating is mapped hereafter as a function of the minimum temperature of operation and of the ratio of capacity at –7 °C and + 7 °C.

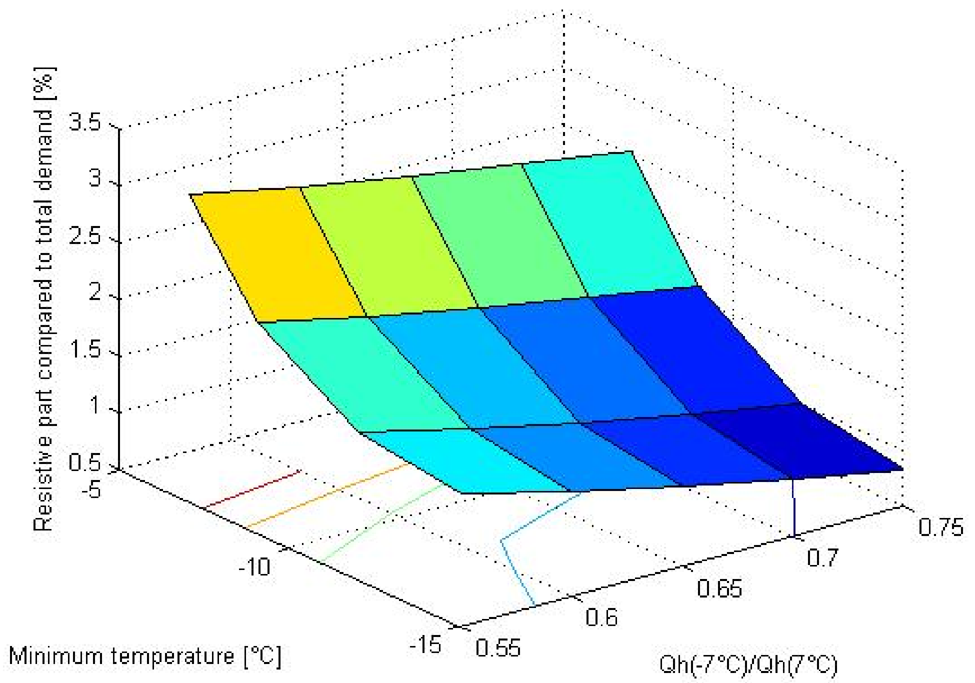


Figure 4-82: Ratio of the resistive part to the total heating demand, EU average 2010

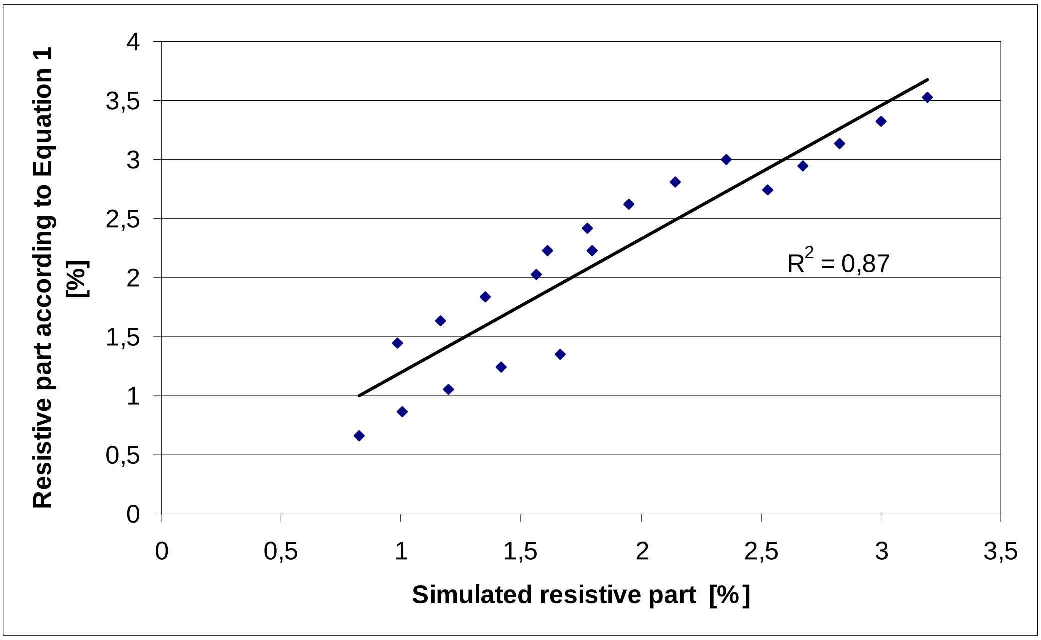


Figure 4-83: Correlation of the ratio between resistive heating to the total heating demand as a function of minimum temperature and capacity loss at – 7 °C, EU average 2010

The resistive demand compared to the total heating demand can be expressed by the following equation, with R in % positive or null:

$$R = 0.064 + 0.002.T_{min} - 0.038.r$$

With,
T_{lim}: the minimum temperature of operation of the heat pump.
r=P(-7)/P(7): ratio of the heating capacity at –7 °C to the rated capacity at 7 °C.

For the purpose of the calculation of the yearly consumption, the correction calculated may be applied either by adding the resistive heating consumption or by computing the corrected SCOP value by including R, the share of the needs covered by resistive heating, with the following formula:

$$SCOP_{on} = \frac{1}{\dfrac{1-R}{SCOP_{hp}} + R}$$

Computation of performance indices including the auxiliary power modes

Reference cooling capacity
Rated capacity as defined by the standard rating conditions given in EN 14511 and noted P_C.

Reference heating capacity
Rated capacity as measured at „A“ temperature conditions as defined in EN 14511 (standard rating or application rating conditions as applicable) and noted P_H.

Equivalent full load hours in cooling mode
Ratio between the cooling requirements of the unit to the unit standard rating capacity in cooling mode according to EN14511 (2004).

Equivalent full load hours in heating mode
Ratio between the heating requirements of the unit to the unit standard rating capacity in heating mode according to EN14511 (2004).

Thermostat-off mode
The air conditioner is operational in heating or cooling mode but inside temperature is lower (higher in heating mode) than the set point. The impact of thermostat-off mode is included in the cycling low and additionally in the thermostat-off consumption corresponding to hours with no cooling or heating load while cooling (or heating) is required by the user. Average electric power in this mode is noted P_{TO} and hours of operation in this mode are noted H_{TO}.

Passive standby mode
The air conditioner is not operational; it can be reactivated either by control device or by timer. This mode corresponds to hours with no occupancy in the building during the cooling or heating season. Average electric power in this mode is noted P_{SB} and hours of operation in this mode are noted H_{SB}.

Off mode
The air conditioner has been switched off by the user, is not operational and cannot be reactivated nor by control device or by timer. This mode corresponds to hours outside the cooling and/or heating season. Average electric power in this mode is noted P_{OFF} and hours of operation in this mode are noted H_{OFF}.

Crank heater operation
The crankcase heater operates when the compressor is off and the outdoor temperature is lower than a given value. Other parameters such as the compressor or the heat exchanger temperature may also be included into the control and have an impact on its energy consumption. Average electric power in this mode is noted P_{CK} and hours of operation in this mode are noted H_{CK}. The crankcase heater may be controlled as a function of the outdoor temperature; the set point temperature is then called T_{CK}. The values of T_{CK} and H_{CK} in the table below are used to assess the number of hours as a function of outside temperature in that case.

T_{CK} (°C)	0	10	20	35 (no control)
H_{CK} Nb of hours	120	1200	2400	5500

Table 4-37: Number of hours of crankcase operation depending on its temperature control

Linear interpolation between these values can be used to assess the hours of operation of crankcase heaters that are controlled as a function of outdoor air temperature. 1200 hours for a control at 10 °C is used in the base cases for reversible heat pumps and it is supposed there is no crankcase heater for cooling only air conditioners.

The calculation of the SEER, SCOP and APF is given by the following equations):

$$\mathrm{SEER} = \frac{H_C.P_C}{\frac{H_C.P_C}{\mathrm{SEER}_{on}} + H_{TO}P_{TO} + H_{SB}P_{SB} + H_{OFF}P_{OFF}}$$

$$\mathrm{SCOP} = \frac{H_H.P_H}{\frac{H_H.P_H}{\mathrm{SCOP}_{on}(+R)} + H_{TO}P_{TO} + H_{SB}P_{SB} + H_{CK}P_{CK} + H_{TO}P_{TO}}$$

$$\mathrm{APF} = \frac{H_C.P_C + H_H.P_H}{\frac{H_C.P_C}{\mathrm{SEER}_{on}} + \frac{H_H.P_H}{\mathrm{SCOP}_{on}} + H_{TO}P_{TO} + H_{SB}P_{SB} + H_{OFF}P_{OFF} + H_{CK}.P_{CK}}$$

Default numbers of hours to compute seasonal and annual performance factors

From cooling and heating electric consumption and average sizing conditions, it is possible to compute equivalent full load hours (at EN14511 standard rating conditions). These equivalent hours are reported in the table below by category of product.

PRODUCTS		Split CO 3.5 kW	Split R 3.5 kW	Split CO 7.1 kW	Split R 7.1 kW	SD 2.2 kW
Active modes, heating and cooling equivalent full load hours						
Cooling	h	302	316	302	316	312
Heating	h	0	612	0	612	0
Cooling mode; hours of operation in other power modes						
Off mode	h	5089	0	5089	0	5089
Standby	h	2279,9	2279,9	2279,9	2279,9	2279,9
Thermostat-off	h	643,9	502,5	643,9	502,5	711,4
Heating mode; hours of operation in other power modes						
Off mode	h	0	0	0	0	0
Standby	h	0	0	0	0	0
Thermostat-off	h	0	2796,8	0	2796,8	0
Crankcase	h	0	1163	0	1163	0
Capacity and electric power in the different modes						
Cooling capacity	kW	3,5	3,5	7,1	7,1	2,2
Heating capacity	kW		4		8,1	
Off mode	kW	0,006	0	0,006	0	0
Standby	kW	0,006	0,006	0,006	0,006	0,006
Thermostat-off	kW	0,036	0,066	0,066	0,066	0,066
Crankcase	kW	0	0,03	0	0,07	0
Seasonal performance indices						
SEERon	-	2,80	2,96	2,43	2,68	2,20
SEER	-	2,37	2,63	2,21	2,53	1,84
SCOP (thermo)	-		2,57		2,49	
R	%		0,3%		0,3%	
SCOP (with R)	-		2,56		2,48	
SCOP	-		2,08		2,19	
SEER	-	2,37	2,63	2,21	2,53	1,84
SCOP	-		2,08		2,19	
APF	-		2,22		2,29	

Table 4-38: Summary of hours of operation of the different power modes and seasonal performance indices with average EU 27 computed values

In the table below, hours are standardized in cooling and in heating mode in order to reach equivalent global performances, in terms of SEER for cooling only units and APF (annual performance factor) for reversible units).

PRODUCTS		Split CO 3.5 kW	Split R 3.5 kW	Split CO 7.1 kW	Split R 7.1 kW	SD 2.2 kW
Active modes, heating and cooling equivalent full load hours						
Cooling	h	315	315	315	315	315
Heating	h	0	610	0	610	0
Cooling mode; hours of operation in other power modes						
Off mode	h	5000	0	5000	0	0
Standby	h	2300	2300	2300	2300	2300
Thermostat-off	h	700	700	700	700	700
Heating mode; hours of operation in other power modes						
Off mode	h	0	0	0	0	0
Standby	h	0	0	0	0	0
Thermostat-off	h	0	2600	0	2600	0
Crankcase	h	0	1200	0	1200	0
Capacity and electric power in the different modes						
Cooling capacity	kW	3,5	3,5	7,1	7,1	2,2
Heating capacity	kW		4		8,1	
Off mode	kW	0,006	0	0,006	0	0
Standby	kW	0,006	0,006	0,006	0,006	0,006
Thermostat-off	kW	0,036	0,066	0,066	0,066	0,066
Crankcase	kW	0	0,03	0	0,07	0
Seasonal performance indices						
SEERon	-	2,80	2,96	2,43	2,68	2,20
SEER	-	2,38	2,55	2,21	2,50	1,85
SCOP (thermo)	-		2,57		2,49	
R	%		0,3%		0,3%	
SCOP (with R)	-		2,56		2,48	
SCOP	-		2,10		2,20	
SEER	-	2,38	2,55	2,21	2,50	1,85
SCOP	-		2,10		2,20	
APF	-		2,22		2,28	

Table 4-39: Summary of hours of operation of the different power modes and seasonal performance indices with standardized values

In what follows - otherwise it is mentioned, the indices here developed are kept for computing the required seasonal performance factors with 8 points in heating mode and the 4 point as shown here in cooling mode. For single duct air conditioners, sales weighted average conditions are kept.

4.5.2.4 Energy consumption computed from seasonal performance indices, split base cases

The following table shows the comparison of the computed values for energy consumption with the simplified indices and the results presented previously in the paragraph 4.5.2.2, in cooling mode and in heating mode for the 4 split base cases.

Cooling mode

Base case	Cooling only single split [3.5kW]	Reversible single split [3.5kW]	Cooling only single split [7.1kW]	Reversible single split [7.1kW]
Seasonal efficiency with average sizing condition at 2 °C	2.80	2.96	2.43	2.68
SEER index	2.86	3.05	2.49	2.77

Table 4-40: Seasonal cooling performance with detailed calculation and simplified index

Heating mode

Base case	Cooling only single split [3.5kW]	Reversible single split [3.5kW]	Cooling only single split [7.1kW]	Reversible single split [7.1kW]
Seasonal COP with average sizing condition at -2 °C	-	2.57	-	2.49
SCOP index With correction R And sizing at 2 °C	-	2.53	-	2.45

Table 4-41: Seasonal heating performance with detailed calculation and simplified index

4.6 End-of-life phase

Japanese and Korean manufacturers advertise they have made efforts to improve the recyclability of their units. Claimed recyclability rates are above the WEEE directive thresholds. Consequently, the default hypothesis (post WEEE conditions) as reported in the MEEuP methodology are kept in the following tasks.

Conclusion

TO BE COMPLETED

Appendix A: Testing and modelling of air conditioner performances

Manufacturers only supply performance data in standard conditions. These conditions do not always match with the real use of air conditioners. As a consequence, it has been decided to test units over a large range of conditions in order to be able to improve the assessment of their real life use. The testing method is based on both air enthalpy and refrigerant enthalpy energy balances. The measurements are used to build up an air conditioner model useful to better model the air conditioner performance in off-design conditions, in this task 4 and then to model options of energy efficiency improvement in further tasks 6 and 7. The method used is validated scientifically by calculating the uncertainty of the measurements performed and results at rating conditions are compatible with measurements with the EN 14511 standard. In addition, it has to be made clear, following the comments made by stakeholders (CECED, 2007), that this measurement is not intended to replace or modify the testing standard EN 14511 in place, and was only set up for the modelling needs of this study.

A.1) Description of the calorimetric test bench

The "indoor" test room is a volume as defined in ISO 5151, in which the desired test-conditions are maintained within the specified tolerances (0.5 K for the temperatures). The air velocity should not exceed 2.5 m/s at the inlet of the tested equipment. The "outdoor" test room has a sufficient volume (ISO 5151) to avoid any perturbation of the "normal" air circulation pattern. The distance between any wall and any equipment surface must be larger than 0.9 m, except for the floor. The manufacturer's installation instructions are respected. Two different methods are used to determine the RAC performances in cooling mode (calorimeter and air enthalpy methods).

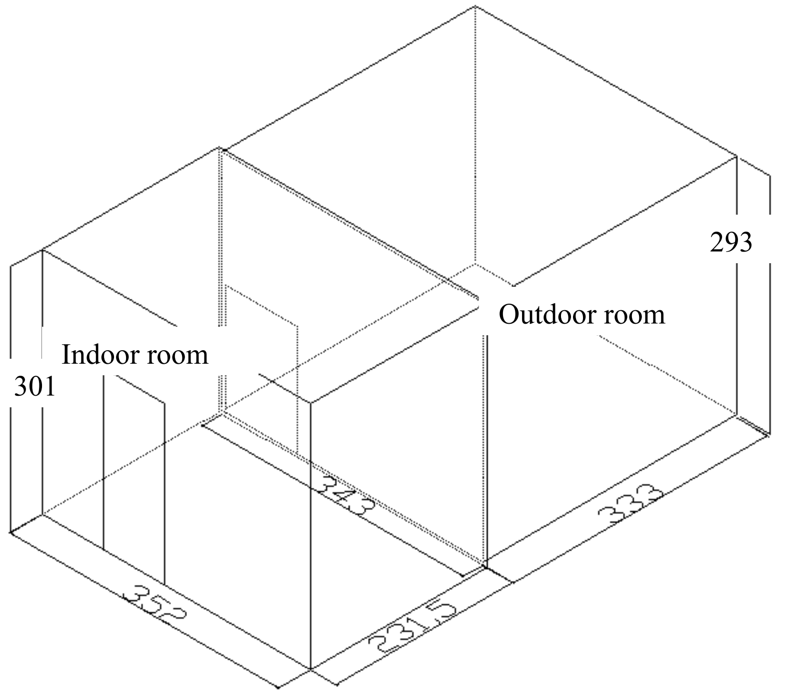
The **calorimeter** test method consists in measuring the capacity simultaneously on both sides of the equipment: indoor and outdoor. Heat and water mass balances determine the indoor unit capacity.
A 7000 W (2000 W +2000 W +3000 W) electrical heaters (Figure A.1) and a 4000 W (2000 W +2000 W) steam boiler (Figure A.1) provide the sensible and latent heat respectively. The saturated steam is superheated in order to compensate the ambient heat losses with help of 800 W super-heater. A power transducer measures the sensible heat input. The latent heat input is determined by measuring, with the help of two 10 kg force transducers, the amount of water consumed by the steam boilers.

Figure A.1: Electric heater (left) and steam boiler (right)

The outdoor unit capacity is also determined in order to check indoor side heat balance. The air is cooled by a 4000 W cooling-coil (Figure A.2), fed by the "city water" and whose air flow rate is at least two times bigger than the condenser air flow rate. The fan cooling coil power is determined from a water side energy balance. The supply and exhaust water temperatures are measured by intrusive thermocouples; the water flow rate is measured every 15 minutes with the help of a balance and continuously checked with the help of a counter.

Figure A.2: The cooling-coil of the outdoor calorimeter

The calorimeter is completely insulated. An insulated wall separates also the "indoor" and "outdoor" rooms of the calorimeter. The air temperatures are controlled in the air channels surrounding the calorimeter. The heat gains or losses of all calorimeter walls are determined with the help of heat flow meters and check thanks to a calibration of both (indoor and outdoor) parts. The general layout of the calorimeter rooms is shown in Figure A.3 and A.4 below.



Outdoor room dimensions:
- Length : 3,43 m;
- Width : 3,33 m;
- Height : 2,93 m;
- Volume: 33, 47 m³.

Indoor room dimensions:
- Length : 3.52 m;
- Width : 2.315 m;
- Height: 3.01 m;
- Volume: 24.53 m³.

Figure A.3: Calorimeters dimensions (in cm)

-

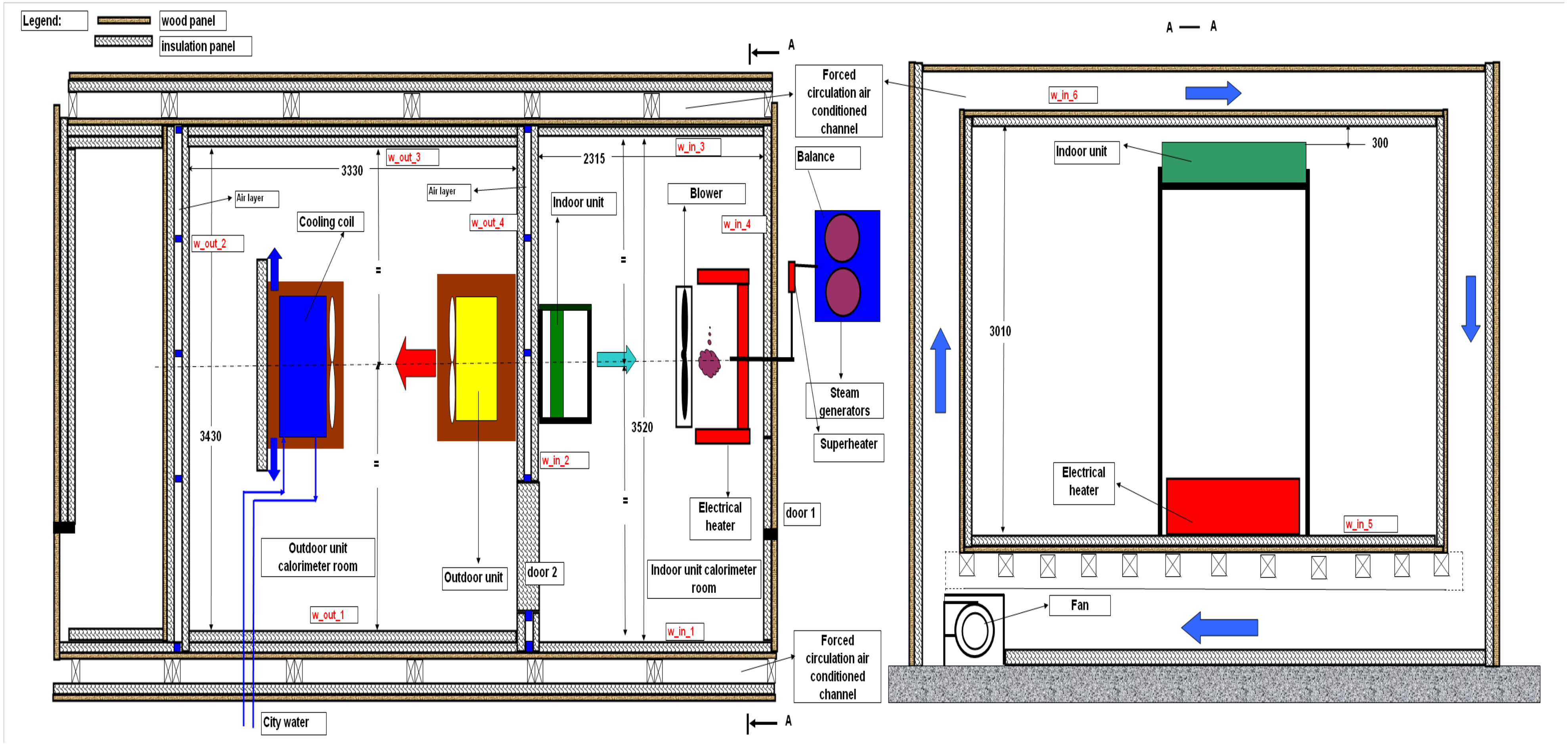


Figure A.4: General view of the calorimeter rooms

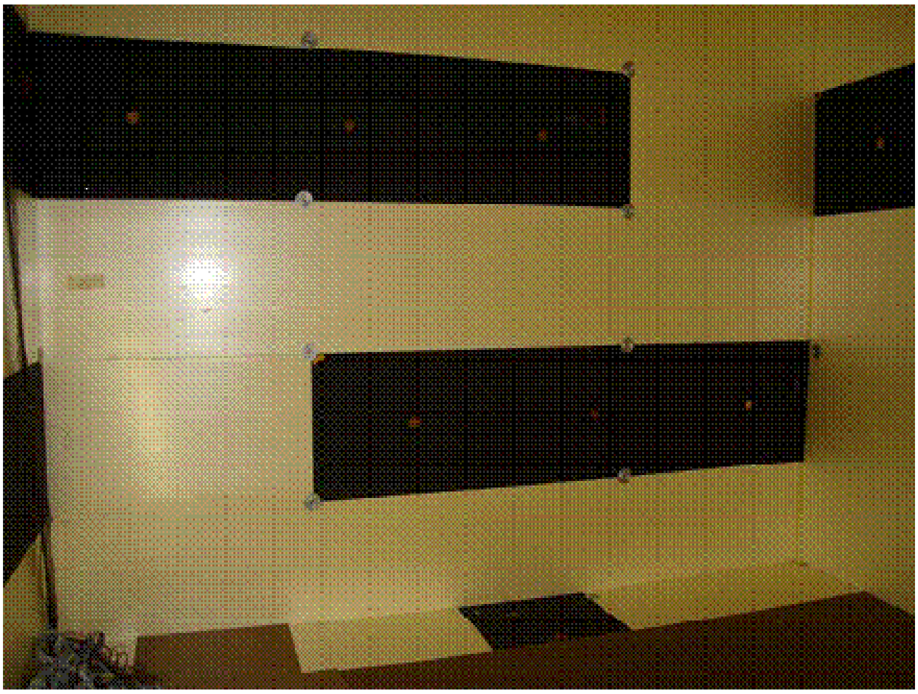


Figure A.5: View of the outdoor room

Several materials are used to build the calorimeter walls. The insulation is made with polystyrene (two different colours are used at the internal surface of this insulating layer (Figure A.5): black (in the rooms) or green (in the figure A.4) to identify the insulation parts used as heat flow meters and white for other parts), wood (magenta) and air (blue). An example is given hereafter for the left wall of indoor room on figure A.4.

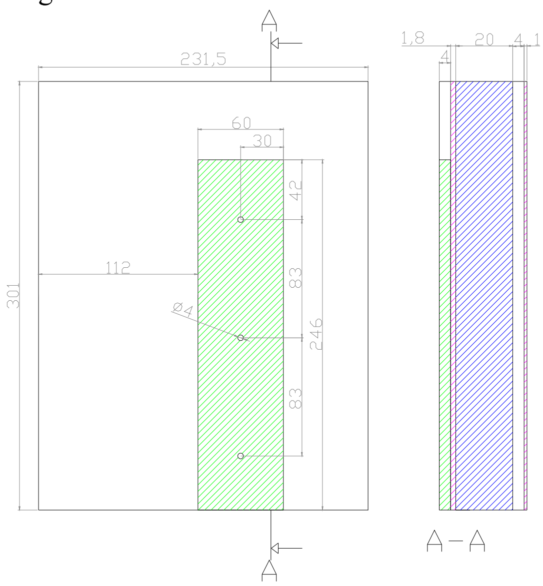


Figure A.6: Wall: w_in_1 (dimensions given in cm – Scale for the wall: 1/1 – Scale for the layers: 2/1) – dimensions and energy flow meter thermocouple position

The wall is made of five layers:
- the first one is insulation : thickness = 40 mm (A-A green and white);
- the second one is a piece of wood : thickness = 18 mm (A-A magenta);
- the third is a layer of air : thickness = 200 mm (A-A blue);
- the fourth is also a heat insulator : thickness = 40 mm (A-A white);
- the last one is again a piece of wood : thickness = 10 mm (A-A magenta).

There are two thermocouples between the heat insulator and the internal face of the wood layer.

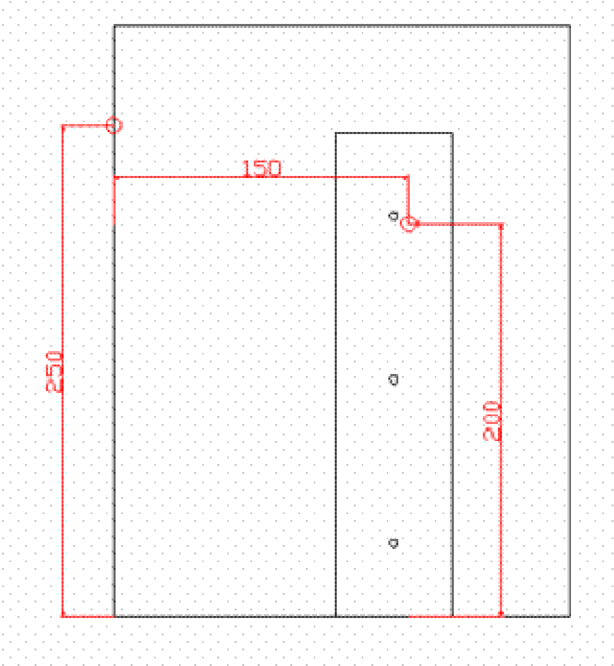


Figure A.7: Position of the thermocouples at internal face of w_in_1 wood layer

The heat gains given in Table A.1 are calculated for a supposed-to-be constant air temperature in the channels surrounding the calorimeter and for different inside air temperatures.
A same superficial heat transfer coefficient of 12 W/m^2K is applied on both faces of the wall for the calculation of its global heat transfer coefficient (U).

Description			Q_dot_amb (W)		
S	6,968 m^2		T_1 = 27°C	T_2 = 21°C	T_3 = 29°C
T_ext	22°C		24,23	-4,845	33,92
Insulator 1	δ = 0,04 m	λ = 0,035 W/m.K			
Wood 1	δ = 0,018 m	λ = 0,14 W/m.K			
Air	δ = 0,2 m	λ = 0,025 W/m.K			
Insulator 2	δ = 0,04 m	λ = 0,035 W/m.K			
Wood 2	δ = 0,01 m	λ = 0,14 W/m.K			
U ($W/m^2.K$)	0,6954				
A*U (W/K)	4,845				
Masse	89,89 kg				
cp_weighted average (J/kg.K)	1387				

Table A.1: Estimated heat gains for the wall: w_in_1

Three thermocouples are used to measure the air temperatures in the channels around the calorimeter:
- One on the left side of the calorimeter (in the middle);
- One on the right side of the calorimeter, (also in the middle);
- One above the ceiling of the calorimeter (in the middle).

The heat gains from the ambience in the two parts of the calorimeters are determined by heat balance, with the help of heat flow meters.

Dry bulb and relative humidity are measured at the supply of the indoor unit. The humidity transducer is shown in the figure below.

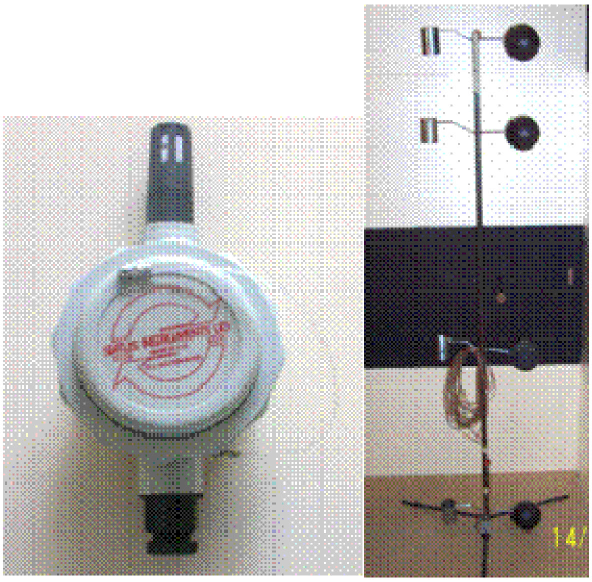


Figure A.8: Relative humidity transducer (left) and thermocouples column (right)

The air temperatures are measured inside both rooms with thermocouples fixed on two columns at four levels. The thermocouples used for air temperature measurement are protected from radiation by double cylindrical screens while other ones measuring globe temperature are located inside black spheres (Figure A.8). The layout of thermocouples on walls is shown in Figure A.9.

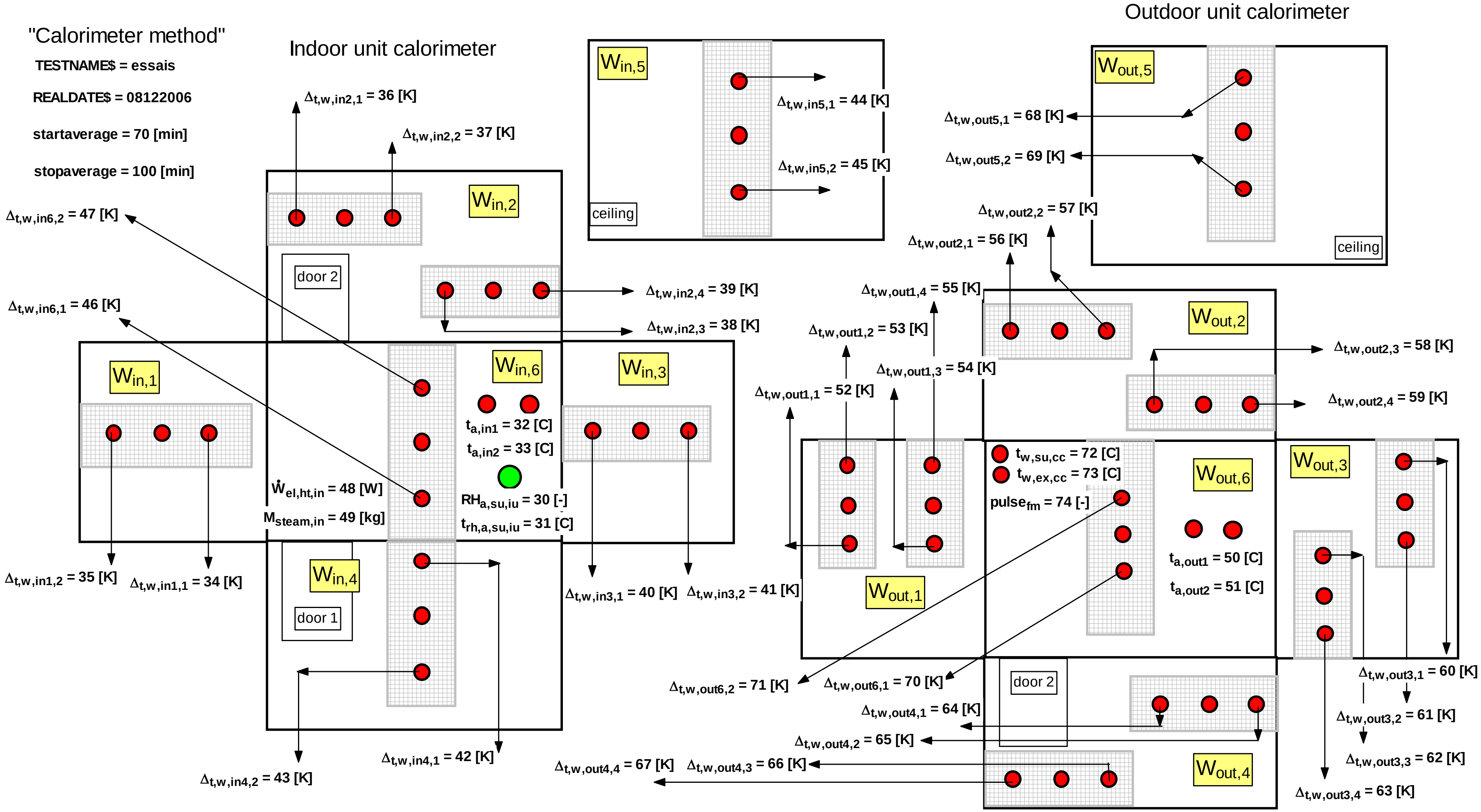


Figure A.9: Heat flow meters

The following probes are installed for dry bulb temperatures measurements:

Refrigerant side:
- 1 thermocouple at compressor supply;
- 1 thermocouple at compressor exhaust;
- 1 thermocouple at condenser supply;
- 1 thermocouple at condenser exhaust;
- 1 thermocouple at expansion valve supply (only in cooling mode);
- 1 thermocouple at evaporator supply;
- 1 thermocouple at evaporator exhaust.

The condensed water of the evaporator is collected and measured with a 10 kg force transducer. The error of the measurements is 1g by 1000g.

Complementary measurements are provided on the air-conditioning unit air side:
- 3 thermocouples at indoor unit supply;
- 3 thermocouples at indoor unit exhaust;
- 3 thermocouples at outdoor unit supply;
- 3 thermocouples at outdoor unit exhaust.

All temperatures are measured with thermocouples type T (copper-constantan). The measuring accuracy is estimated to ±0.2 K. Refrigerant temperatures are measured with contact thermocouples. A flow-meter is integrated in the refrigerant circuit (subcooled liquid line), it is a Coriolis flow meter type Micro Motion, Série Elite, ¼, inox 316L. The refrigerant pressures are measured with Keller and Druck transmitters.

The thermocouples and pressure transducers locations along the refrigerant loop in cooling mode are indicated in Figure A.10**.** In heating mode the location is the same only the name change function of heat exchangers reversibility. The variables are recorded by a 3495A IMP/PC system, with a sampling rate of 1 min. The data acquisition system measures the temperature with an accuracy better than ±0.3°C. The voltages given by the pressure and power sensors are measured with an accuracy of ±(0.01% reading + 0.01% full scale).

The thermal balance of the walls has been led and can be supplied. More generally, the complete report on the description of the test bench is available on demand, whether only main elements have been presented here.

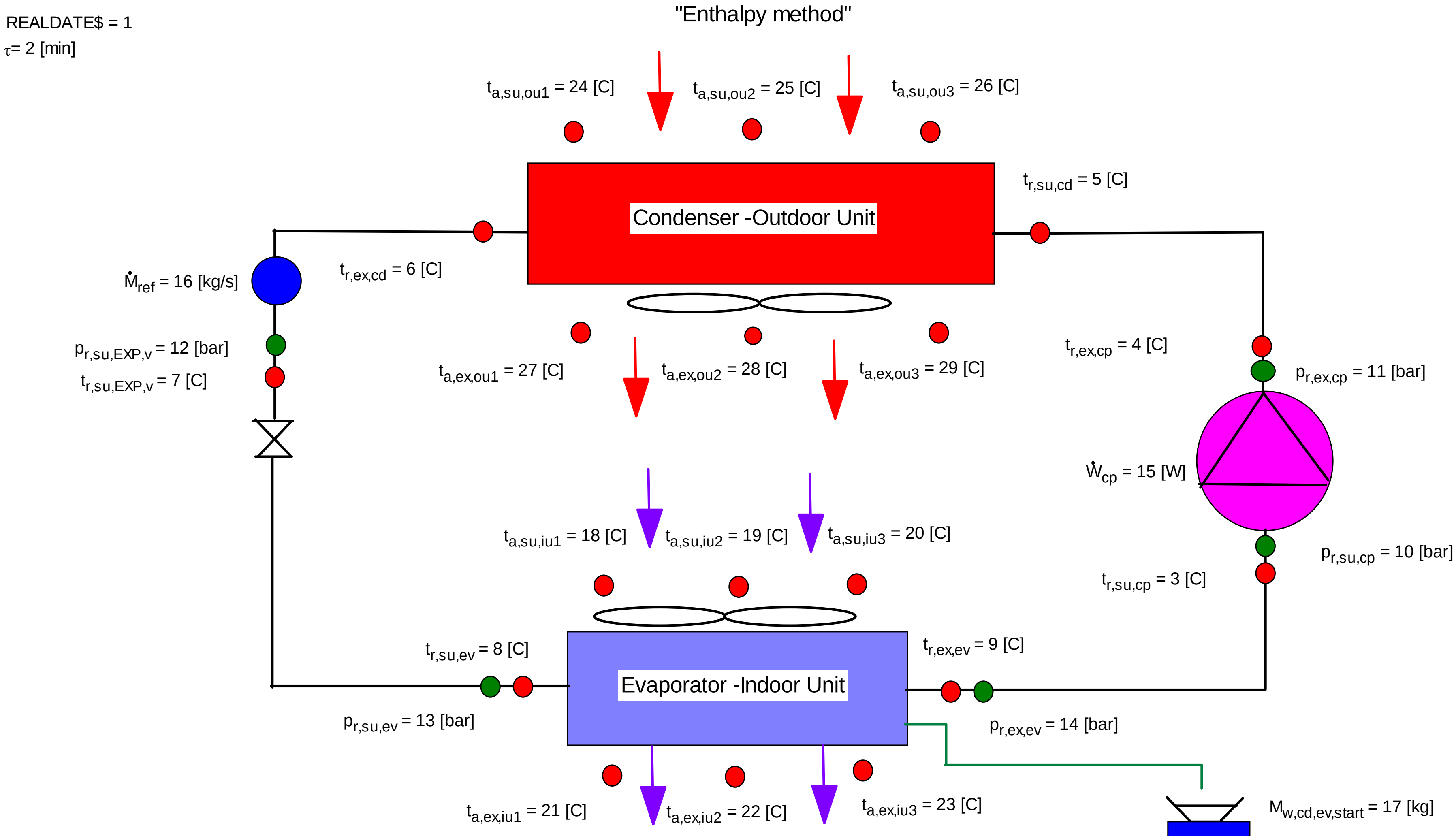


Figure A.10: Thermocouples and pressure transducers locations along the refrigerant loop – cooling mode

A.2) Testing of specific units

Testing conditions at full load

The three standard combinations of test conditions in cooling mode are presented in Table A.2.

Parameter		Standard test conditions		
		T1	T2	T3
Temperature of air entering indoor side [C]				
Dry-bulb		27	21	29
Wet-bulb		19	15	19
Temperature of air entering outdoor side [C]				
Dry-bulb		35	27	46
Test frequency		Rated frequency *		
Test voltage		Rated voltage **		
T1	Standard cooling capacity rating conditions for moderate climates			
T2	Standard cooling capacity rating conditions for cool climates			
T3	Standard cooling capacity rating conditions for hot climates			
*	Equipment with dual-rated frequencies shall be tested at each frequency			
**	The test voltage dual-rated voltage equipment shall be performed at both voltages or at the lower of the two voltages if only a single rating is published			

Table A.2: ISO 5151 test conditions – cooling mode

In order to cover a larger domain of use, six other combinations of test conditions are tested. Test conditions are indicated in the table below.

Parameter	T4	T5	T6	T7	T8	T9
T_in [C]	27	27	23	23	27	20
RH_in [-]	0.32	0,6	0,6	0,34	0,47	0,53
T_out [C]	35	35	31	31	20	20

Table A.3: Supplementary combinations of test conditions – cooling mode

For the couples of tests named **T4-T5** and **T6-T7** respectively, the indoor temperatures is kept the same, but the relative humidity is varied in order to get dry and wet regimes. In the case of **T8**, the air temperature is higher on the indoor side than on the outdoor side. For **T9**, the indoor and outdoor air temperatures are the same. In the same manner we define the three standard combinations of test conditions in heating mode (Table below).

Parameter		Standard test conditions		
		T1	T2	T3
Temperature of air entering indoor side [C]				
Dry-bulb		20	20	20
Wet-bulb		-	-	-
Temperature of air entering outdoor side [C]				
Dry-bulb		7	2	-7
Wet-bulb		6	1	-8
RH[-]		0.86	0.85	0.77
Test frequency		Rated frequency *		
Test voltage		Rated voltage **		
*	Equipment with dual-rated frequencies shall be tested at each frequency			

**	The test voltage dual-rated voltage equipment shall be performed at both voltages or at the lower of the two voltages if only a single rating is published

Table A.4: ISO 5151 test conditions – heating mode

In order to cover a larger domain of use, six other combinations of test conditions will be considered ((Table below)).

Parameter	T4	T5	T6	T7	T8	T9
T_in [C]	20	22	25	22	22	22
RH_in [-]	-	-	-	-	-	-
T_out [C]	-10	7	7	2	-7	-10
RH_out[-]	0.9	0.86	0.86	0.85	0.77	0.9

Table A.5: Supplementary combinations of test conditions – heating mode

Unit 1

The first unit tested has the main following characteristics:
- Cooling capacity (rated capacity): 2.5 kW.
- Refrigerant fluid: 410A
- COP: class A
- EER: class A
- Inverter compressor

Testing results at full load

No	Test	Indoor temperature	Indoor relative humidity	Outside temperature	Indoor unit cooling power			Evaporating Temperature	Exhaust Air Temperature - Evaporator	SHR	Outdoor unit heating power			Condensing temperature	Exhaust air temperature - Condenser
					Enthalpy Method	Calorimeter. Method	Rel diff				Enthalpy Method	Calor. Method	Rel diff		
		°C	-	°C	W	W	%	°C	°C	-	W	W	%	°C	°C
1	120207A	27,08	0,44	35,35	2225	2255	1,37	11,39	12,58	0,846	2666	2588	-2,93	44,96	39,4
2	160207A	20,76	0,5283	27,28	2786	2806	0,7357	4,186	5,561	0,703	3335	3322	-0,3897	39,56	32,17
3	260207A	28,78	0,4016	45,99	1860	1852	-0,4195	15,15	15,93	1	2371	2289	-3,46	54,11	49,79
4	130207A	27,0067	0,31	33,96	2326	2379	2,261	9,816	10,87	1	2797	2692	-3,754	44,03	38,15
5	140207A	27,396	0,61	33,73	3262	3216	-1,417	11,9	13,33	0,513	3966	3926	-0,9961	47,5	39,84
6	220207A	23,57	0,6074	30,42	2962	2906	-1,892	8,008	9,356	0,58	3565	3634	1,933	43,36	35,83
7	210207A	23,37	0,3453	31,01	2149	2209	2,785	6,944	7,994	1	2578	2572	-0,2088	40,79	35,03
8	150207A	27,46	0,4463	20,86	2370	2443	3,074	10,57	12,1	0,797	2619	2584	-1,327	30,18	24,59
9	230207A	20,88	0,5409	20,38	2397	2389	-0,3558	5,481	6,698	0,684	2748	2792	1,578	30,4	24,24
10	Dry_nom	27,046	0,2723	36,6	2528	/	/	9,975	11,17	1	3132	/	/	47,39	40,99

No	Test	Power Consumption of the RAC	Power consumption of the compressor	Compressor Effectiveness	EER	Compressor Rotation Speed**	Power Consumption of the evaporator fan	Level speed of the fan	Evaporator fictitious effectiveness	Evaporator fictitious global heat transfer coefficient	Condenser fictitious effectiveness	Condenser fictitious global heat transfer coefficient
		W	W	-	-	tr/min	W	-	-	[J/K]	-	[J/K]
1	120207A	584,1	507,1	0,691	3,8	2320	11,02	Medium	0,8372	651,2	0,3519	278,5
2	160207A	741,3	664	0,6951	3,8	3438	11,24	Medium	0,8718	600,8	0,3601	298,6
3	260207A	639,8	561,7	0,6542	2,9	1974	12,09	Medium	0,943	414,9	0,3242	238,5
4	130207A	609,2	530,8	0,7046	3,8	2522	12,47	Medium	0,9838	1526	0,3542	285,7
5	140207A	885,9	809,3	0,6879	3,7	3481	10,6	Medium	0,8661	726,1	0,3786	303,9
6	220207A	796,1	719,4	0,6816	3,7	3356	10,75	Medium	0,8686	642,7	0,3695	299
7	210207A	551,4	473,1	0,7044	3,9	2417	12,32	Medium	0,8929	743	0,3506	276,9
8	150207A	340,5	313,6	0,6299	7.0	2142	10,92	Medium	0,8201	571,2	0,3752	328,8
9	230207A	422,9	346	0,7502	5,7	2544	10,88	Medium	0,8785	581,4	0,3609	311,8
10	Dry_nom	710,5	630,7	0,7114	3,6	2731	13,81	High	1	5340	0,3374	281,6

Table A.6:Medium class room air-conditioner -Cooling mode test results

** **Note:** the compressor rotation speed is not directly measured but calculated supposing that the compressor volume is: V_s_cp_max=0.00000897 m³

No	Test	Indoor temperature	Outdoor relative humidity	Outdoor temperature	Indoor unit heating power			Condensing temperature	Air temperature at the indoor unit exhaust	Outdoor unit cooling power			Evaporating temperature	Air temperature at the outdoor unit exhaust
					Enthalpy method	Calorimeter. method	Rel diff			Enthalpy method	Calorimeter method	Rel diff		
		°C	-	°C	W	W	%	°C	°C	W	W	%	°C	°C
1	050307A	19.63	0.8663	7.32	3135	3295	5.11	45.95	42.25	2509	2548	1.57	2.62	5.3
2	070307A	19.72	0.8585	2.75	2244	2244	0	37.67	34.02	1971	2031	3.02	-3.8	0.3
3	080307B	20.46	0.7818	-6.84	2287	2287	0	39.32	35.56	1796	1778	-1.04	-10.9	-8.9
4	090307A	20.01	0.772	-9.84	2195	2195	0	38.14	34.15	1737	1637	-5.76	-15.0	-12.5
5	050307B	21.49	0.8763	7.9	2692	2748	2.1	45.28	42.02	2161	2239	3.58	3.7	5.9
6	05037C	24.94	0.8604	7.52	2800	2813	0.47	48.8	45.47	2190	2278	4.02	3.1	5.5
7	060307A	21.6	0.8667	2.23	2212	2212	0	39.45	35.92	1967	1966	-0.04	-4.01	0.04
8	080307C	22.54	0.7529	-6.42	1731	1731	0	41.46	37.52	1731	1808	4.49	-11.86	-9.2
9	080307D	22.19	0.7977	-9.73	2121	2121	0	39.79	36.39	1657	1627	-1.806	-13.54	-11.6

No	Test	RAC power consumption	Compressor power consumption	Compressor effectiveness	EER	Compressor rotation Speed	Indoor unit fan power consumption	Level speed of the fan	Condenser fictitious effectiveness	Condenser fictitious global heat transfer coefficient	Evaporator fictitious effectiveness	Evaporator fictitious global heat transfer coefficient
		W	W	-	-	tr/min	W		-	[J/K]	-	[J/K]
1	050307A	886.8	806.1	0.67	3. 6	3503	14.74	Medium	0.79	217.0	0.39	758.2
2	070307A	687.1	607.6	0.67	3.3	3154	13.47	Medium	0.79	243.0	0.33	421.3
3	080307B	783.5	702.4	0.68	2.9	3796	15.07	Medium	0.74	201.1	0.52	640.5
4	090307A	818.8	737.6	0.68	2.7	4270	15.19	Medium	0.73	202.9	0.47	532.3
5	050307B	729.7	649.8	0.68	3.7	2899	13.92	Medium	0.79	205.5	0.40	751.3
6	05037C	854.9	774.5	0.67	3.3	3157	14.33	Medium	0.76	195.9	0.40	741.4
7	060307A	734.1	654.9	0.65	3.0	3213	13.23	Medium	0.79	241.3	0.31	425.5
8	080307C	830.6	749.7	0.67	2.7	3899	14.97	Medium	0.72	189.0	0.53	470.3
9	080307C	802	720.9	0.67	2.7	3933	15.05	Medium	0.74	198.4	0.50	621.9

Table A.7: Medium class room air-conditioner: Heating mode test results

** Note: the compressor rotation speed is not directly measured, but calculated supposing that the compressor volume is: $V\_s\_cp\_max=0.00000897$ m³

Dynamic behaviour is then illustrated for both cooling and heating modes for one performance point. These tests were conducted without the intervention of the manufacturer. For the test n°2 in cooling mode (T2 conditions of ISO 5151 standard), the temperatures are maintained as near as possible to 21 and 27 °C in the indoor and outdoor rooms respectively. The humidity ratio in the indoor room is near to 53 %. The total test time is 300 min. The averaging period is chosen between minutes 220 and 280. The time evolutions of the air temperatures at supplies and exhausts of indoor and outdoor units of are shown in Figure A.11.

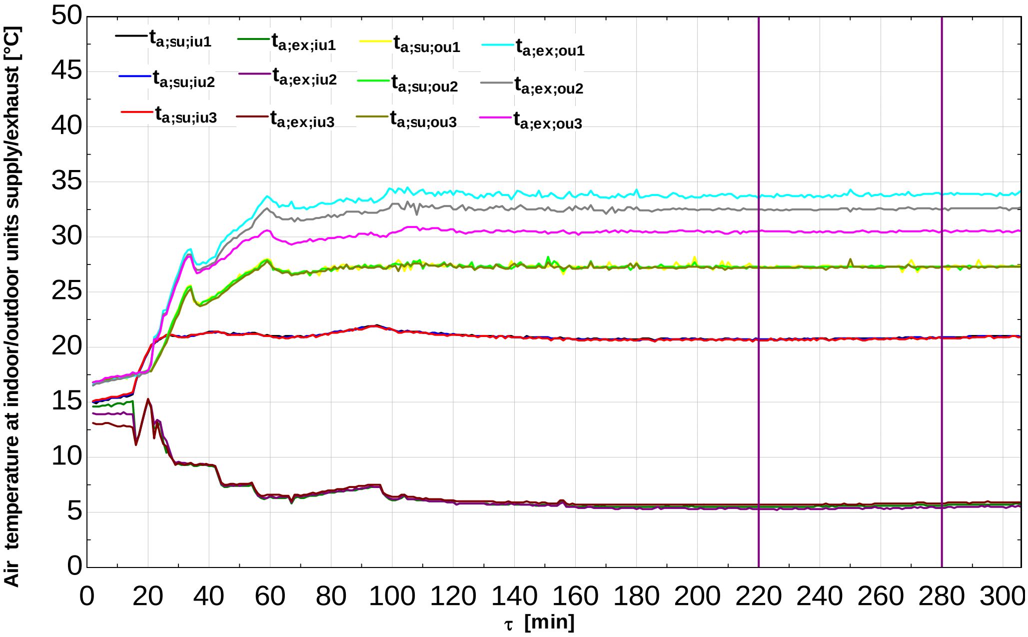


Figure A.11: Time evolutions of the air temperatures at supplies and exhausts of the indoor and outdoor units, Test n°2 cooling mode

The test n°2 in heating mode is illustrated on figure A.12 with frost and defrost cycles.

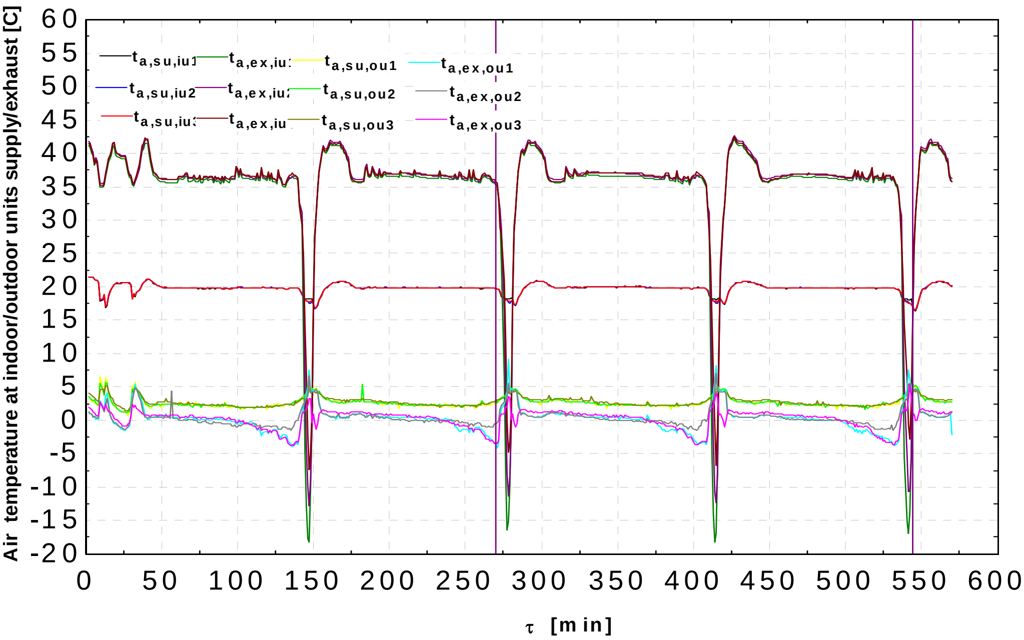


Figure A.12: Evaporator and condenser supply and exhaust air temperatures

A.3) Modeling of the performances of the units tested

Modeling strategy

To determine the performances of the air-conditioning units and also to simulate the entire system (calorimeter room and air conditioners), a model is developed with the thermodynamic software EES (Klein, 2002). The inputs of the model are the average values of measurements in the selected period when the air-conditioning system and the calorimeters are in steady-state conditions.

Modeling is done in two phases: a so-called "mother" model has been developed by fitting physical performance equations of each component detailed models.

This mother model will be then used in tasks 5, 6 and 7 to model the performances of air conditioners in cooling and heating modes under varied operating conditions.

This "mother" model is used to generate performance curves for varying air and humidity conditions. This so-called "daughter" model may be coupled with building simulations to couple the air conditioner model(s) to every detailed building simulation.

This strategy will also enable to compute the impact of options for improvement on a yearly basis with realism.

This detailed model is able to predict the performances of the air conditioner in cooling and heating mode with the following precision.

Cooling mode (full load):
- cooling power: ± 5%;
- heating power : ± 5%;
- room air conditioner power consumption: ± 5%;
- EER: ± 5%;
- SHR: ± 5%;
- air temperature at the indoor unit exhaust: -0.5 K - + 1.5 K;
- air temperature at the outdoor unit exhaust: -0.5 K - + 1.5 K.

Heating mode (full load):
- cooling power: ± 5%;
- heating power : ± 5%;
- room air conditioner power consumption: ± 5%;
- EER: ± 5%;
- air temperature at the indoor unit exhaust: -1.5 K - + 1K;
- air temperature at the outdoor unit exhaust: -2 K - + 1.5 K.

Detailed modeling

A so-called "reference" ("detailed", "mechanistic", or "mother") model is, as much as possible, based on real physics. It's an assembly of meaningful equations, describing the dominant physical phenomena, as they are understood and as they can be represented. But the model is nevertheless application-oriented: its realism is only required in a given domain of use and for relevant (input and output) variables. The reference model of a room air conditioner is built by assembling several component models described hereafter.

Heating and dry cooling coil

A same heating and dry cooling coil model is used to simulate both the condenser and the evaporator in dry regime. The coil is supposed here to behave as a fictitious semi-isothermal heat exchanger. Laminar and turbulent regimes are considered on air and "refrigerant" sides respectively. The output variables are the coil thermal power and the exhaust temperatures of both fluids (air and refrigerant). The parameters are: the nominal flow rates of both fluids and the three nominal thermal resistances (air side, metal and refrigerant side). The input variables are the supply conditions on both sides of the coil.
The main equations of this model are presented hereafter.

Air side heat balance:

$$\dot{Q}_{ev,dry} = \dot{M}_{a,ev} \cdot cp_{a,su,ev} \cdot (t_{a,su,ev} - t_{a,ex,ev,dry})$$

Refrigerant side heat balance:

$$\dot{Q}_{ev,dry} = \dot{M}_{r,ev,dry} \cdot \Delta_{h,ev,dry}$$

$$\Delta_{h,ev,dry} = h_{r,ex,ev,dry} - h_{r,su,ev,dry}$$

Heat transfer through the (fictitious) semi-isothermal-flow heat exchanger:

$$\dot{Q}_{ev,dry} = \varepsilon_{ev,dry} \cdot \dot{C}_{a,ev,dry} \cdot (t_{a,su,ev} - t_{ev,mean,dry})$$

$$\dot{C}_{a,ev,dry} = \dot{M}_{a,ev} \cdot cp_{a,su,ev}$$

$$\varepsilon_{ev,dry} = 1 - \mathbf{exp}(-NTU_{ev,dry})$$

$$NTU_{ev,dry} = \frac{AU_{ev,dry}}{\dot{C}_{a,ev,dry}}$$

Definition of a weighted average temperature on the refrigerant side of the evaporator:

$$t_{ev,mean,dry} = \frac{t_{ev,dry} \cdot (h_{r,sat,ex,ev,dry} - h_{r,sat,su,ev,dry}) + t_{r,ex,ev,dry} \cdot (h_{r,ex,ev,dry} - h_{r,sat,ex,ev,dry})}{h_{r,sat,ex,ev,dry} - h_{r,sat,su,ev,dry} + h_{r,ex,ev,dry} - h_{r,sat,ex,ev,dry}}$$

with

$$t_{ev,dry} = \frac{t_{r,su,ev,dry} + t_{r,sat,ex,ev,dry}}{2}$$

The global heat transfer coefficient of this heat exchange is defined by considering three thermal resistances in series:

$$AU_{ev,dry} = \frac{1}{R_{a,ev} + R_{r,ev,dry} + R_{m,ev}}$$

Both convective resistances are defined by reference to nominal values:

$$R_{r,ev,dry} = R_{r,ev,n} \cdot \left[\frac{\dot{M}_{r,ev,n}}{\dot{M}_{r,ev,dry}}\right]^{0.8}$$

$$R_{a,ev} = R_{a,ev,n} \cdot \left[\frac{\dot{M}_{a,ev,n}}{\dot{M}_{a,ev}}\right]^{0.6}$$

Cooling coil in dry and wet regimes

This model is based on Merckel theory (combination of latent and sensible heat transfer), with a very slight adaptation: air enthalpy is here replaced by wet bulb temperature as total heat transfer potential).

Jim Braun's hypothesis is also used (replacing partially dry-wet by completely dry or completely wet regimes).

Selected outputs are:

- Coil "emissions" (total, sensible and latent cooling power);
- Air state at coil exhaust (temperature, moisture content and relative humidity);
- Water condensate flow rate;
- Refrigerant temperature at coil exhaust.

The parameters and the input variables are the same as in dry regime.

The equations already developed are transposed to the wet regime by substituting to the air a fictitious ideal gas, whose temperature is the actual air wet bulb temperature.

The air side heat balance and heat transfer equations become:

$$\dot{Q}_{ev,wet} = \dot{M}_{a,ev} \cdot (h_{a,su,ev} - h_{a,ex,ev,wet} + (w_{a,su,ev} - w_{a,ex,ev,wet}) \cdot c_{w,ev} \cdot t_{c,ev,wet})$$

$$\dot{Q}_{ev,wet} = \varepsilon_{ev,wet} \cdot \dot{C}_{a,ev,wet} \cdot (t_{wb,su,ev} - t_{ev,mean,wet})$$

The air state at the evaporator exhaust is calculated, according to a classical ASHRAE procedure, by identifying fictitious contact effectiveness (i.e. by considering a fictitious air side isothermal surface):

$$\varepsilon_{c,ev,wet} = \frac{h_{a,ev} - h_{b,ev}}{h_{a,ev} - h_{c,ev,wet}}$$

$$\varepsilon_{c,ev,wet} = \frac{w_{a,ev} - w_{b,ev}}{w_{a,ev} - w_{c,ev,wet}}$$

$$\varepsilon_{c,ev,wet} = \frac{w_{a,ev} - w_{b,ev}}{w_{a,ev} - w_{c,ev,wet}}$$

$$NTU_{c,ev,wet} = \frac{1}{R_{a,ev} \cdot \dot{C}_{a,ev}}$$

According to Jim Braun's proposal, the (dry or wet) regime giving the highest cooling power is selected as nearest to reality:

$$\dot{Q}_{ev} = \mathbf{If}\,(\dot{Q}_{ev,dry}, \dot{Q}_{ev,wet}, \dot{Q}_{ev,wet}, \dot{Q}_{ev,dry}, \dot{Q}_{ev,dry})$$

The sensible power is the power that would be obtained with same supply and exhaust air temperatures, but without any change of air water content:

$$\dot{Q}_{ev,sens} = \dot{M}_{a,ev} \cdot cp_{a,su,ev} \cdot (t_{a,su,ev} - t_{a,ex,ev})$$

The latent power is defined by difference between total and sensible powers:

$$\dot{Q}_{ev,lat} = \dot{Q}_{ev} - \dot{Q}_{ev,sens}$$

Compressor

The model used here is well adapted to the simulation of most rotary compressors. It includes heat transfer at the supply, at the exhaust and to the ambient. The pressure drop during the suction and discharge are neglected, as well as any lubricant circulation. The compression is considered as isentropic up to the internal pressure and then at constant volume until the exhaust pressure.

The conceptual schema of the compressor is presented in **Figure 1.** The evolution of the refrigerant is decomposed into four steps:

1) Heating-up (su → su1).
2) Isentropic compression (su1 → in)
3) Compression at a fixed volume (in →ex1).
4) Cooling down (ex1→ex)

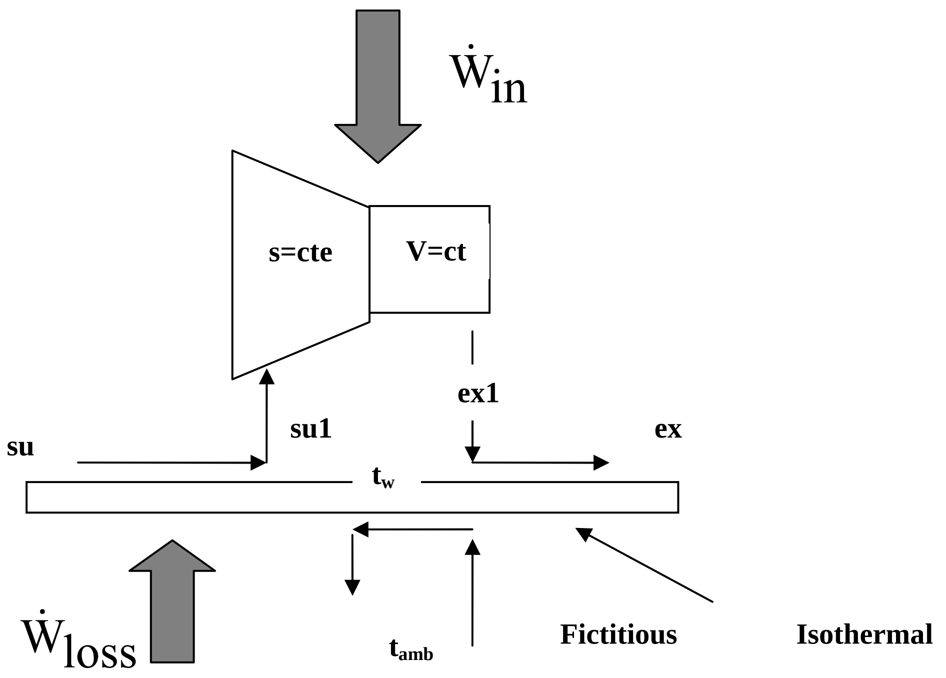


Figure A.13: Conceptual scheme of the compressor model

The evolution of the refrigerant state through the compressor is presented in Figure A.14:

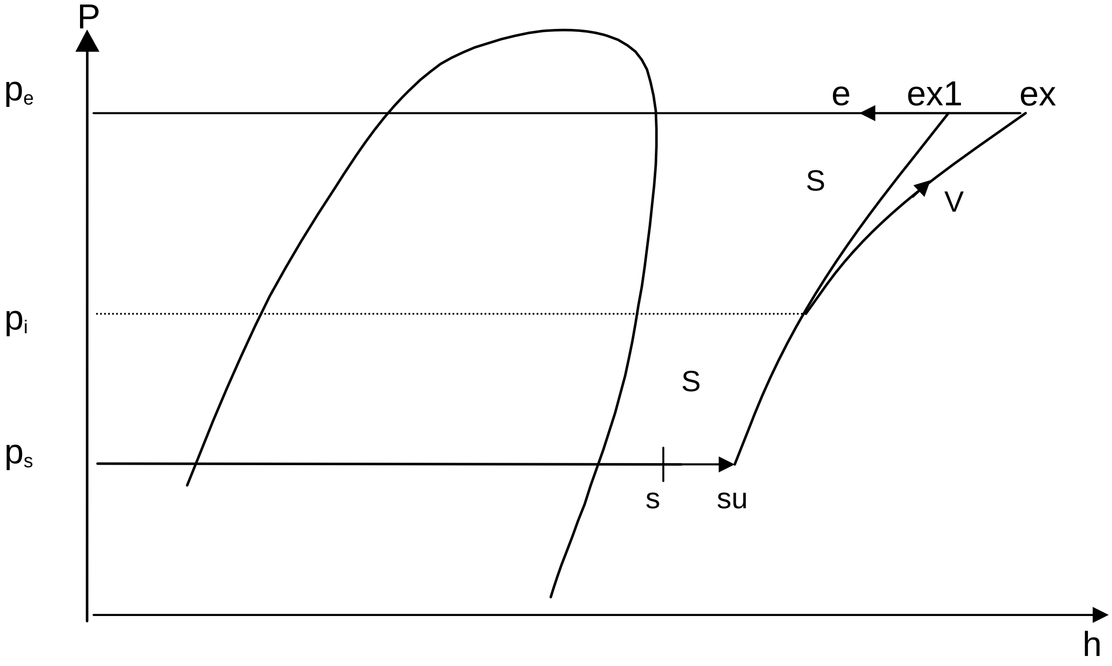


Figure A.14: Refrigerant state through the compressor

Refrigerant mass flow rate

The refrigerant mass flow rate is given by:

$$\dot{M}_{cp} = \frac{\dot{V}_{s,cp}}{v_{su1,cp}}$$

with

$$\dot{V}_{s,cp} = \dot{V}_{s,cp,max} \cdot X_{cp}$$

and with

M_dot_cp	- refrigerant mass flow rate	[kg/s];
v_ su1_cp	– refrigerant volume at the compressor supply after overheating	[m³/kg];
V_dot_s_cp	- refrigerant compressor flow rate	[m³/s];
V_dot_s_cp_max	- maximal refrigerant flow rate	[m³/s];
X_cp	– load factor (*control variable)*	[-].

Heat transfer

The different heat exchanges are represented by reference to a unique isothermal wall, which is supposed to be in contact with the refrigerant at both (supply and exhaust) sides and with the ambient. Electromechanical losses are also supposed to be directly transmitted (as equivalent heat) to this wall. The supply wall-to-refrigerant heat transfer can be described through the following equations:

$$h_{su1,cp} - h_{su,cp} = \frac{\dot{Q}_{su,cp}}{\dot{M}_{cp}}$$

$$\dot{Q}_{su,cp} = \varepsilon_{su,cp} \cdot \dot{C}_{su,cp} \cdot (t_{w,cp} - t_{su,cp})$$

$$\dot{C}_{su,cp} = \dot{M}_{cp} \cdot c_{p,su,cp}$$

$$\varepsilon_{su,cp} = 1 - \mathbf{exp}(-NTU_{su,cp})$$

$$NTU_{su,cp} = \frac{AU_{su,cp}}{\dot{C}_{su,cp}}$$

with

Q_dot_su_cp – supply heat transfer	[W];
h_su_cp – refrigerant enthalpy at the compressor supply	[J/kg];
h_su1_cp – refrigerant enthalpy after heating-up	[J/kg];
c_p_su_cp – refrigerant specific heat	[J/kg-K];
ε_su_cp - supply heat transfer effectiveness	[-];
C_dot_su_cp - thermal capacity flow rate	[W/K];
t_w_cp - fictitious wall uniform temperature	[°C];
t_su_cp – refrigerant temperature at compressor supply	[°C];
NTU_su_cp - number of transfer units	[-];
AU_su_cp – supply heat transfer coefficient	[W/K].

The same set of equations is used for the exhaust heat transfer.

The ambient-to-compressor heat transfer is given by:

$$\dot{Q}_{amb,cp} = AU_{amb,cp} \cdot (t_{amb,cp} - t_{w,cp})$$

with

t_amb_cp – "ambient" temperature (corresponding to the air temperature at condenser exhaust in the present case!)	[°C];
AU_amb_cp - fictitious ambient heat transfer coefficient	[W/K].

Wall balance

The wall balance is give by the following relationship:

$$\dot{W}_{loss,cp} - \dot{Q}_{su,cp} - \dot{Q}_{ex,cp} + \dot{Q}_{amb,cp} = 0$$

With

W_dot_loss_cp - compressor electro-mechanical loss	[W];
Q_dot_ex_cp – wall-to-refrigerant heat transfer	[W]

Exhaust conditions

The compression process is decomposed into two steps:

1) An adiabatic, reversible and therefore also isentropic compression, up to the adapted internal pressure;
2) An isochoric evolution (compression or expansion) until the exhaust pressure.

The refrigerant enthalpy after this process can be calculated as follows:

$$h_{ex1,cp} = h_{su1,cp} + w_{in,cp}$$

with

$$w_{in,cp} = w_{in1,cp} + w_{in2,cp}$$

$$w_{in1,cp} = h_{in,cp} - h_{su1,cp}$$

$$h_{in,cp} = \mathbf{h}(fluid\$, s{=}s_{in,cp}, v{=}v_{in,cp})$$

$$s_{in,cp} = s_{su1,cp}$$

$$v_{in,cp} = \frac{v_{su1,cp}}{r_{v,in,cp}}$$

$$w_{in2,cp} = v_{in,cp} \cdot (p_{ex1,cp} - p_{in,cp})$$

with:

h_ex1_cp - refrigerant enthalpy after isochoric compression	[J/kg];
w_in_cp - internal compression work	[J/kg];
w_in1_cp - isentropic work	[J/kg];
h_in_cp - refrigerant enthalpy after isentropic compression	[J/kg];
s_in_cp – corresponding entropy	[J/kg-K];
v_in_cp –corresponding volume	[m³/kg];
w_in2_cp – isochoric wor k	[J/kg];
p_in_cp – internal pressure	[Pa];
p_ex1_cp –pressure after isochoric evolution	[Pa];
r_v_in_cp – compressor internal volume ratio (*parameter to be identified*)	[-].

Prediction of the compressor power

The compressor power can be split into two terms:

$$\dot{W}_{cp} = \dot{W}_{in,cp} + \dot{W}_{loss,cp}$$

With

W_dot_in_cp – internal power	[W];
W_dot_loss_cp – electro-mechanical loss	[W]

The electro-mechanical loss can also be split into two terms:

$$\dot{W}_{loss,cp} = \dot{W}_{loss0,cp} + \alpha_{cp} \cdot \dot{W}_{in,cp}$$

With

W_dot_loos0_cp – constant electro-mechanical compressor loss [W];
α_ cp – loss factor (*parameter to by identified)* [-];

Fan(s) model

The fans are currently modelled with the help of similarity variables: flow, pressure and power factors. These variables can be correlated to each other by polynomial expressions. The main output of a fan model is the flow rate expressed here in "specific" value (in kg/s of *dry* air), as usually in air conditioning. Other outputs are: flow rate and pressure factors, exhaust air speed, total pressure difference, isentropic power and isentropic temperature increase across the fan (these two last outputs can be used as checking information). The fan is supposed to be characterised by the diameter of its impeller (scale variable), the exhaust area and the coefficients of two polynomial correlations. Supply air conditions (temperature, pressure and moisture content), rotation speed and supply-to-exhaust *static* pressure difference are taken as input variables.

The equations of this model are built on the basis of the definitions of two (flow and pressure) similarity factors:

$$\phi = \frac{\dot{V}}{A \cdot U}$$

Reference area:

$$A = \pi \cdot \frac{D^2}{4}$$

Peripheral speed:

$$U = \pi \cdot D \cdot N$$

rotation speed:

$$N = \frac{rpm}{60}$$

Pressure factor:

$$\psi = \frac{\Delta P_{total}}{P_{dynam,periph}}$$

$$\Delta P_{total} = \Delta P_{stat} + P_{dynam,ex}$$

Two dynamic pressures are considered: one at the exhaust and the other one at the periphery of the impeller:

Exhaust dynamic pressure:

$$P_{dynam,ex} = \frac{C_{ex}^2}{2 \cdot v}$$

$$C_{ex} = \frac{\dot{V}}{A_{ex}}$$

Peripheral dynamic pressure:

$$P_{dynam,periph} = \frac{U^2}{2 \cdot v}$$

The other non-dimensional variables considered are the isentropic effectiveness and the power factor:

Fan isentropic power:

$$\dot{W}_s = \dot{V} \cdot \Delta P_{total}$$

Isentropic effectiveness:

$$\varepsilon_s = \frac{\dot{W}_s}{\dot{W}_{shaft}}$$

Power factor:

$$\lambda = \frac{\phi \cdot \psi}{\varepsilon_s}$$

The three factors are inter-correlated through polynomial laws such as:

$$\phi = \alpha_0 + \alpha_1 \cdot \psi + \alpha_2 \cdot \psi^2 + \alpha_3 \cdot \psi^3$$

$$\lambda = \beta_0 + \beta_1 \cdot \psi + \beta_2 \cdot \psi^2 + \beta_3 \cdot \psi^3$$

This is an “orphan” model (not generated from a reference model), but from long time well validated and easy to tune…
Polynomials are fitted to manufacturer’s performance data. Attention is paid to the distinction between total and static pressures: manufacturers present fan performance in terms of total pressure rise, whereas the measurements are usually made in terms of static pressures.

Validation of the detailed model

The model results are shown as a function of the test results.

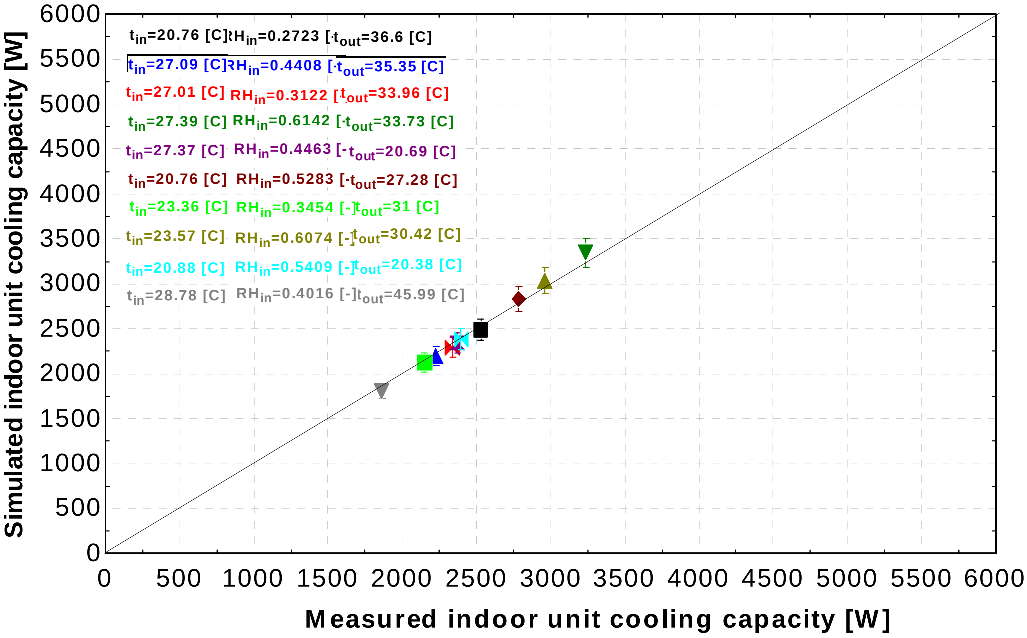


Figure A.15: Simulated indoor unit cooling power as function of measured indoor unit cooling power

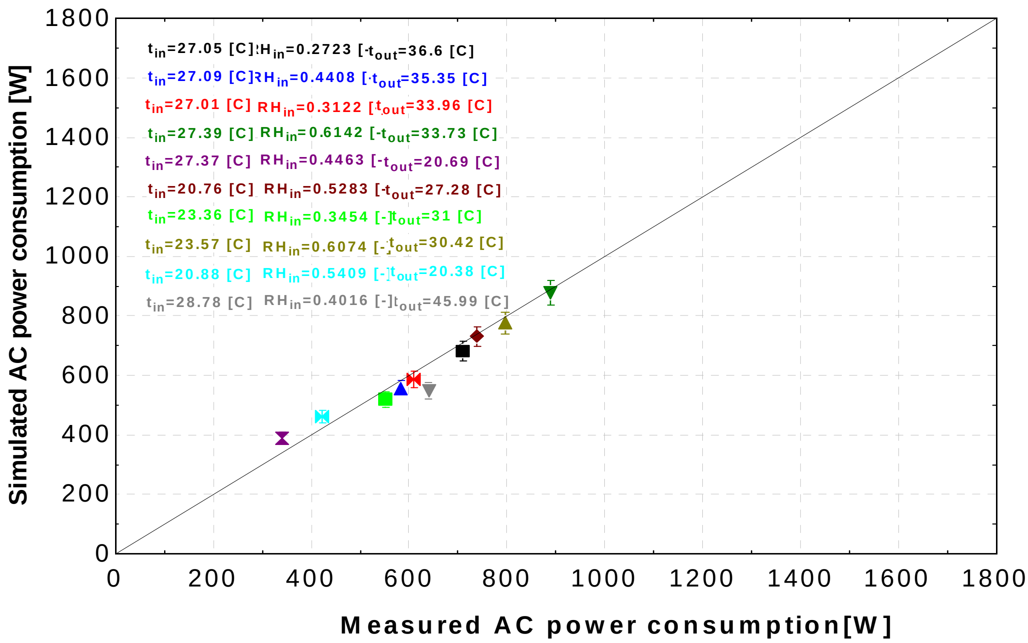


Figure A.16: Simulated room air-conditioner power consumption as function of measured room air-conditioner power consumption

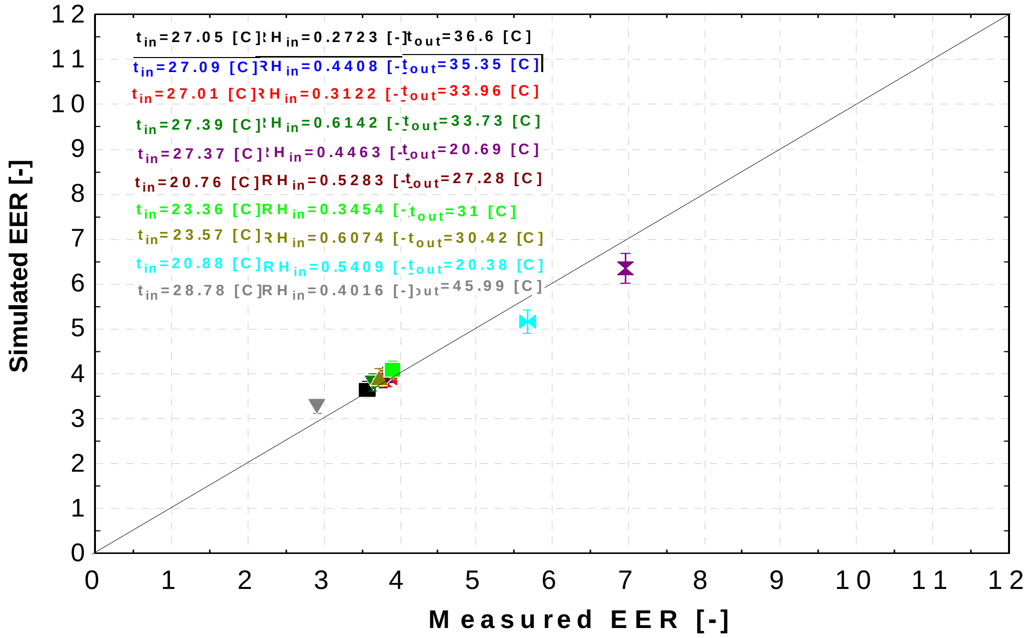


Figure A.17: Simulated room air-conditioner EER as function of measured room air-conditioner EER

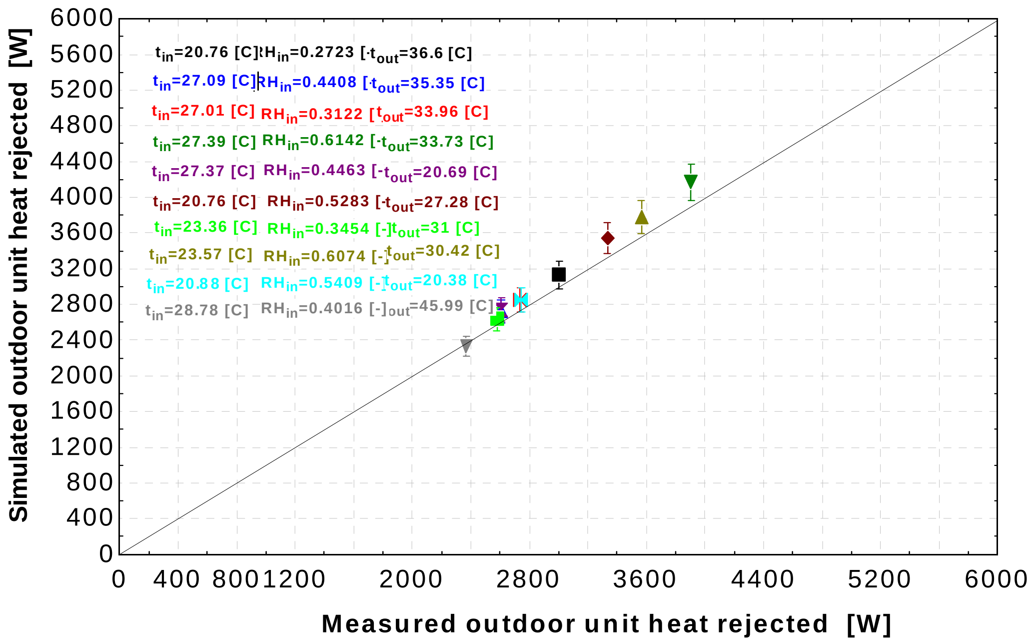


Figure A.18: Simulated outdoor unit heat rejected as function of measured outdoor unit heat rejected

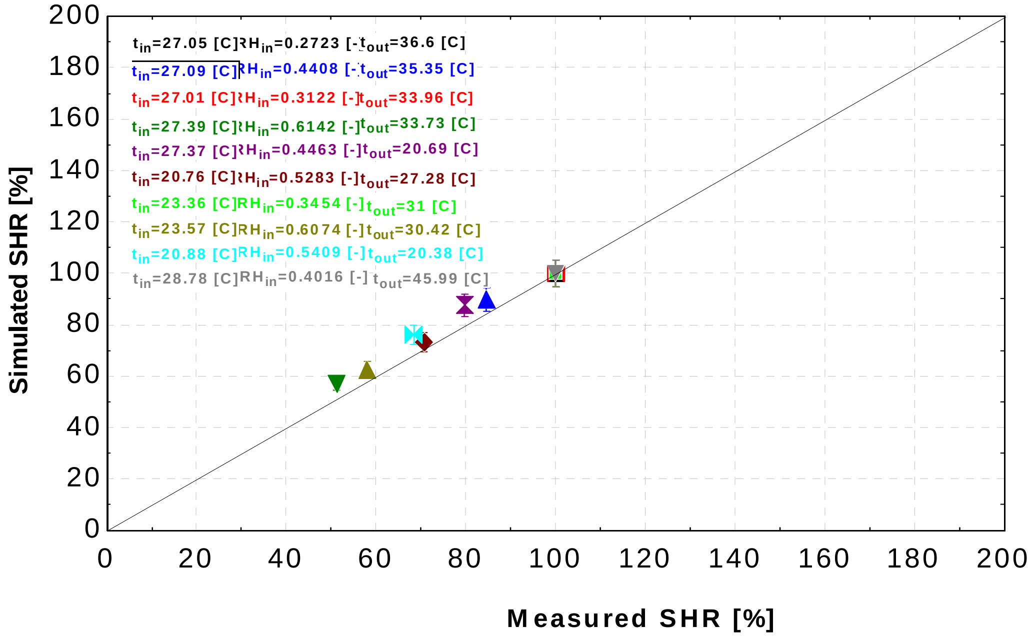


Figure A.19: Simulated indoor unit SHR as function of measured indoor unit SHR

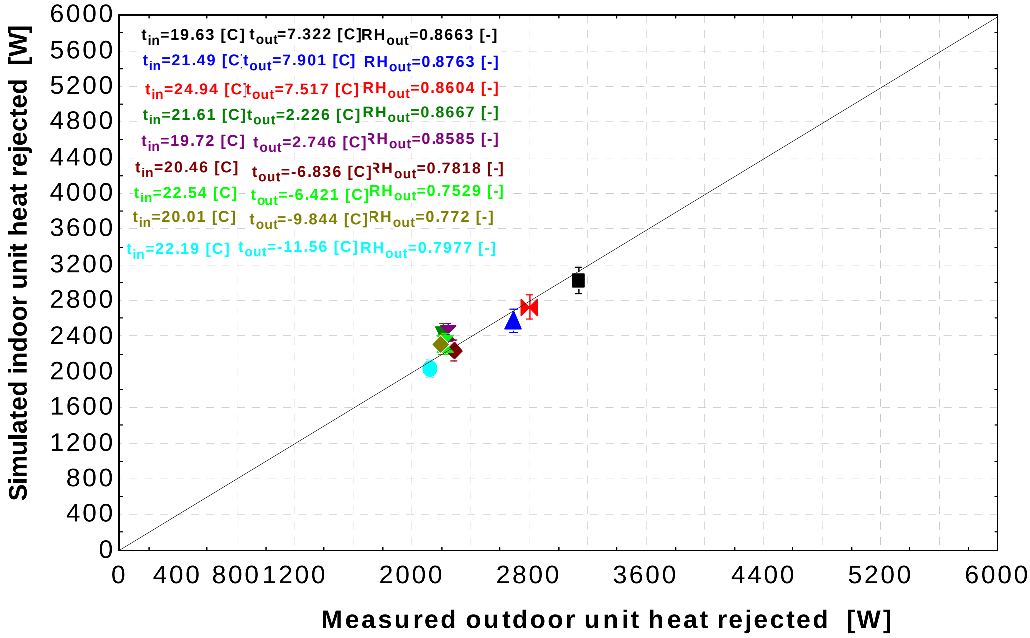


Figure A.20: Simulated indoor unit heating power as function of measured indoor unit heating power

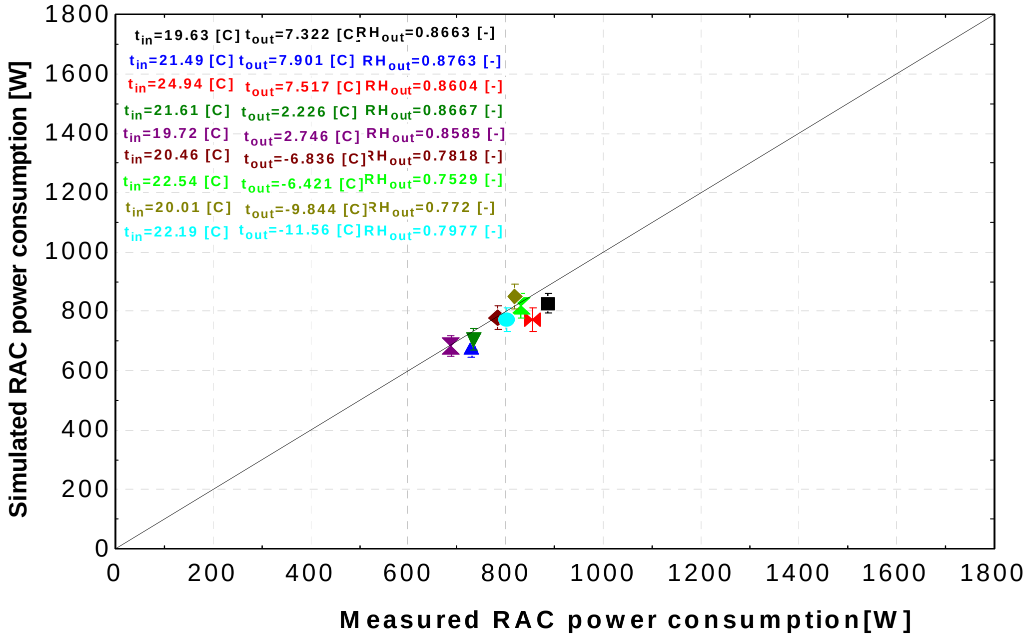


Figure A.21: Simulated room air-conditioner power consumption as function of measured room air-conditioner power consumption

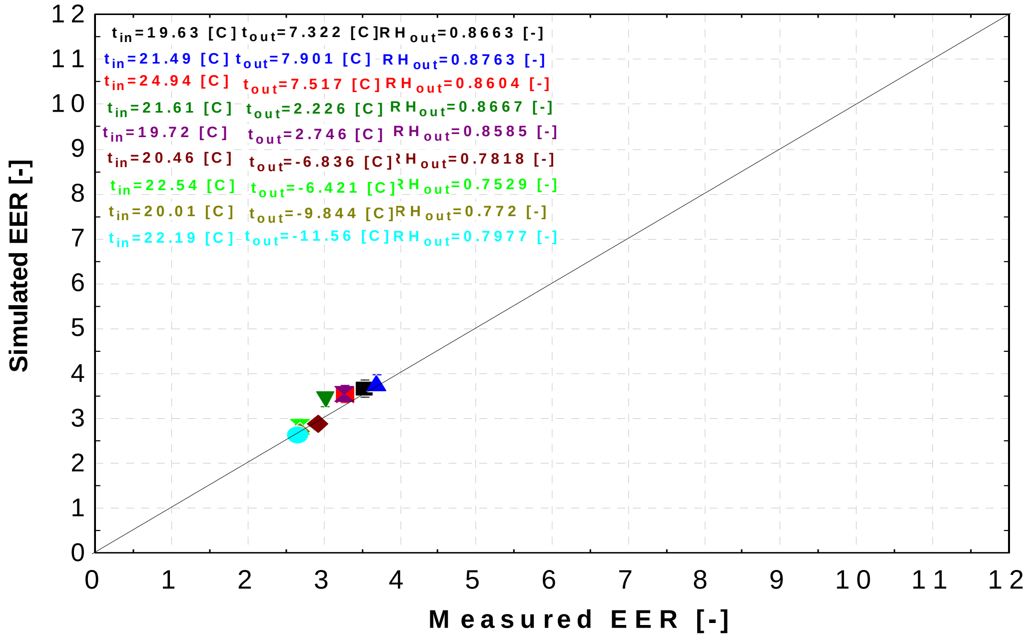


Figure A.22: Simulated room air-conditioner EER as function of measured room air-conditioner EER

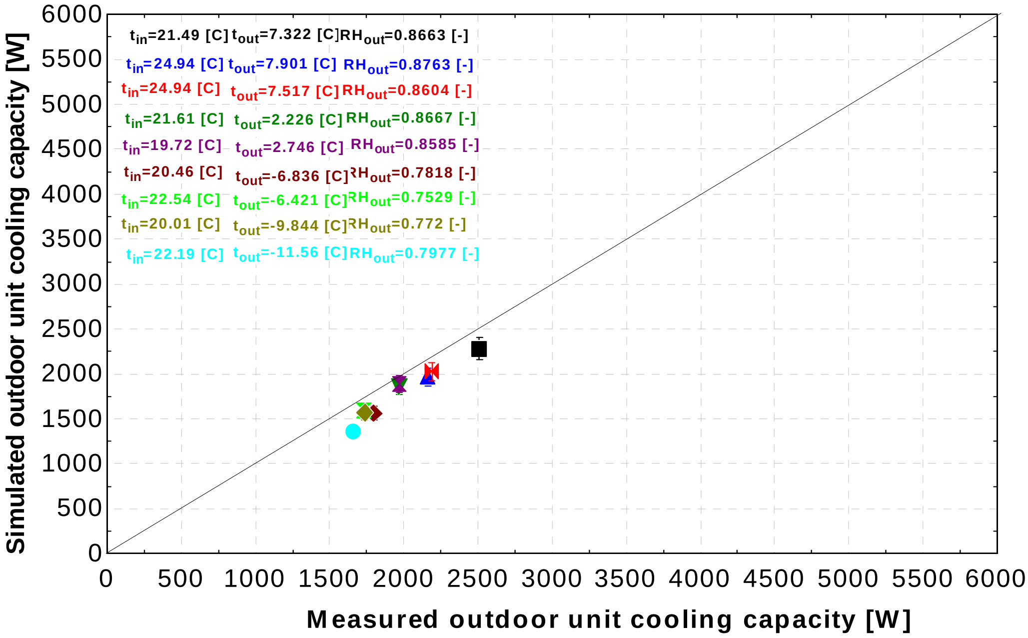


Figure A.23: Simulated outdoor unit cooling power as function of measured outdoor cooling power

Performances at full load

We present here the performance curves that enable to illustrate the variations of EER and COP with indoor and outdoor conditions. Indoor fan flow rate is fixed.

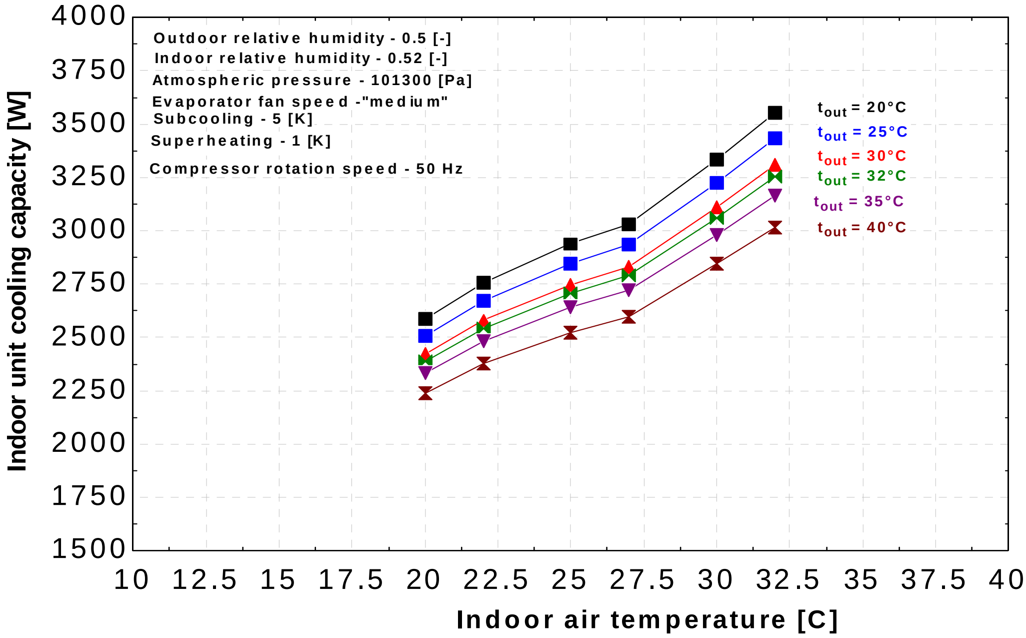


Figure A.24: Indoor unit cooling capacity as a function of indoor and outdoor air temperature-cooling mode

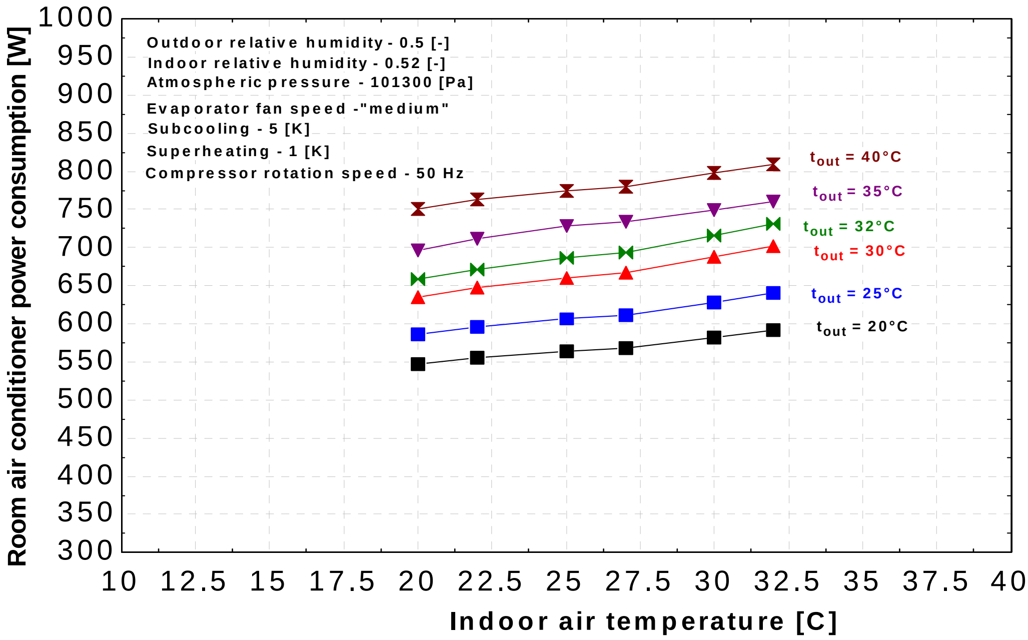


Figure A.25: air conditioner power consumption as a function of indoor and outdoor air temperature-cooling mode

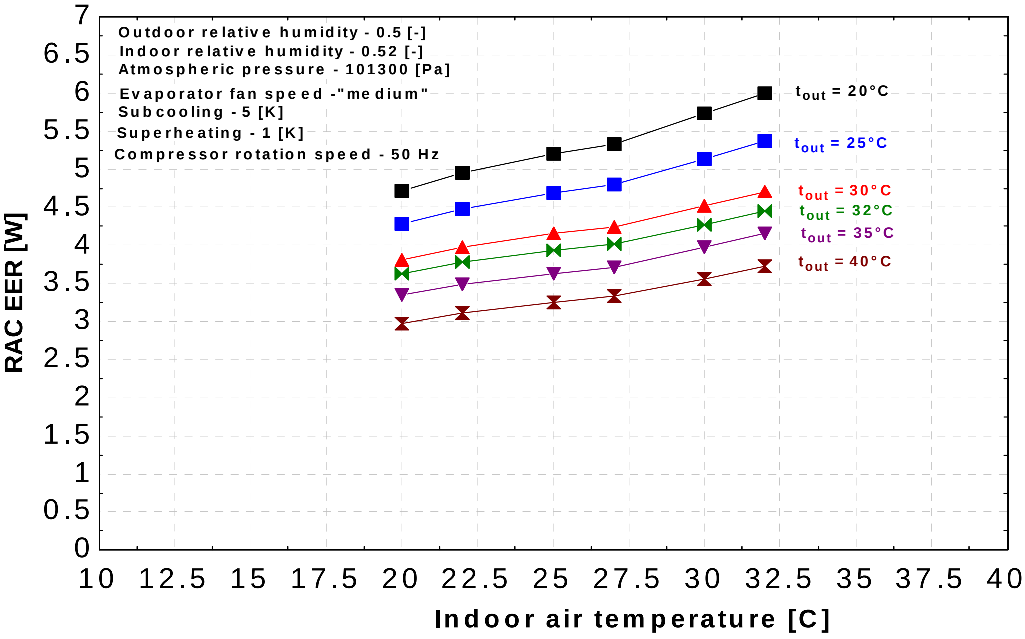


Figure A.26: air conditioner EER as a function of indoor and outdoor air temperature-cooling mode

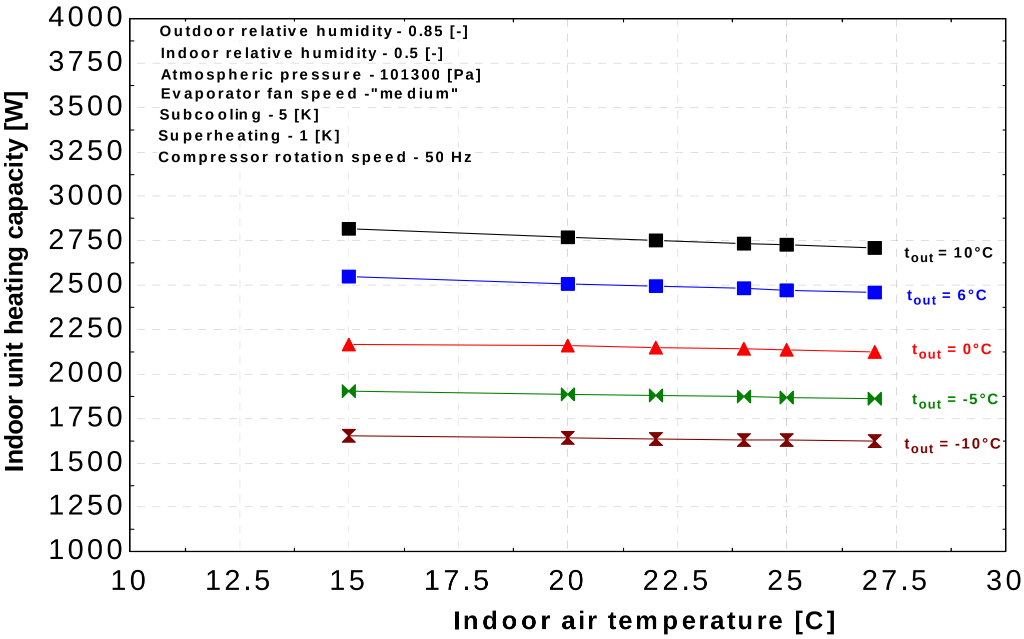


Figure A.27: Indoor unit heating capacity as a function of indoor and outdoor air temperature-heating mode

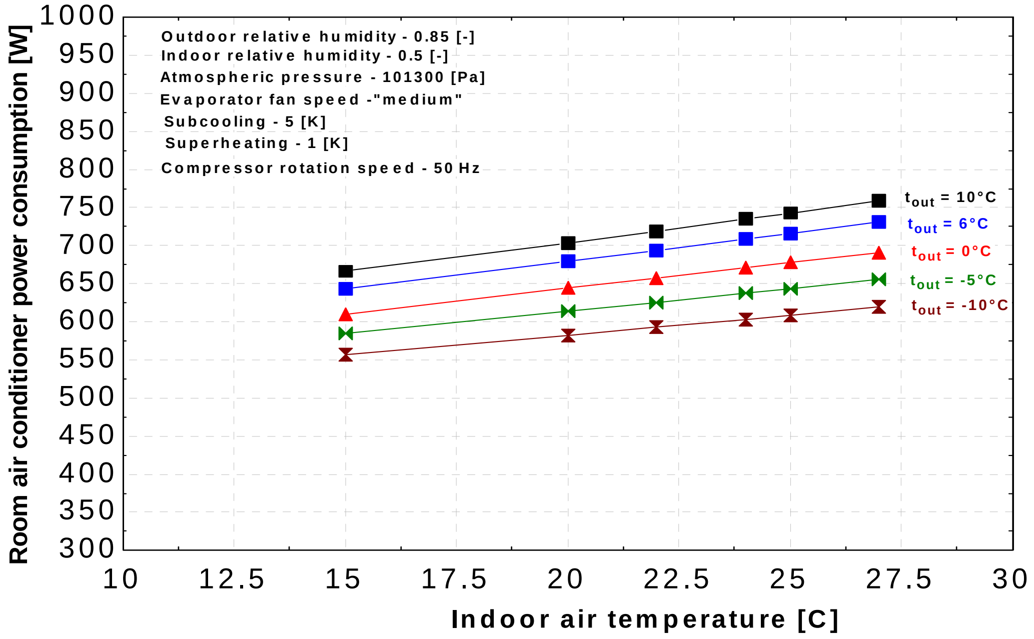


Figure A.28: air conditioner power consumption as a function of indoor and outdoor air temperature-heating mode

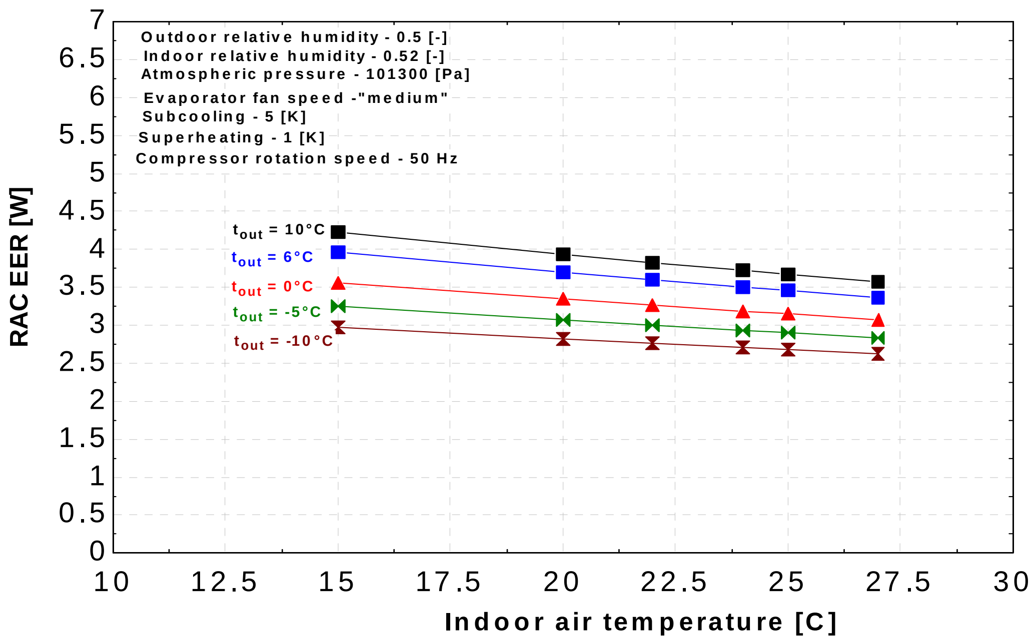


Figure A.29: air conditioner COP as a function of indoor and outdoor air temperature-heating mode

Simplified model

The simplest modeling consists in expressing the cooling full load nominal capacity and the electrical consumption as polynomial functions of two independent variables:

– The outdoor air temperature. The relative humidity doesn't affect the system performances because the outdoor heat exchanger (condenser) works in dry regime
– The indoor air temperature. This temperature must be then associated with a constant relative humidity (for example 50%).

The system performances are already predicted with the reference model. Third degree polynomials laws can be fitted on these results:

$$Q_{cooling,full,nom}=C1_{capacity}+C2_{capacity}*t_{in}+C3_{capacity}*t_{in}^2+C4_{capacity}*t_{in}^3+C5_{capacity}*t_{out}+C6_{capacity}*t_{out}^2+C7_{capacity}*t_{out}^3 +C8_{capacity}*t_{in}*t_{out}+C9_{capacity}*t_{in}*t_{out}^2+C10_{capacity}*t_{in}^2*t_{out}+C11_{capacity}*t_{in}^2*t_{out}^2$$

$$W_{cooling,full,nom}=C1_{power}+C2_{power}*t_{in}+C3_{power}*t_{in}^2+C4_{power}*t_{in}^3+C5_{power}*t_{out}+C6_{power}*t_{out}^2+C7_{power}*t_{out}^3 +C8_{power}*t_{in}*t_{out}+C9_{power}*t_{in}*t_{out}^2+C10_{power}*t_{in}^2*t_{out}+C11_{power}*t_{in}^2*t_{out}^2$$

With the help of the reference model, these laws are corrected in order to take into account the effects of the indoor relative humidity of the fan speed.

$$\dot{W}_{cooling,full} = \dot{W}_{cooling,full,nom} \cdot C_{power,relhum} \cdot C_{speed,power}$$

$$\dot{Q}_{cooling,full} = \dot{Q}_{cooling,full,nom} \cdot C_{capacity,relhum} \cdot C_{speed,capacity}$$

Relative humidity correction factors

Both correction factors (cooling capacity and power consumption ratios) can be defined as functions of a relative humidity ratio:

$$ratio_{RH,in} = \frac{1}{RH_{in}}$$

These functions are identified in the two following figures:

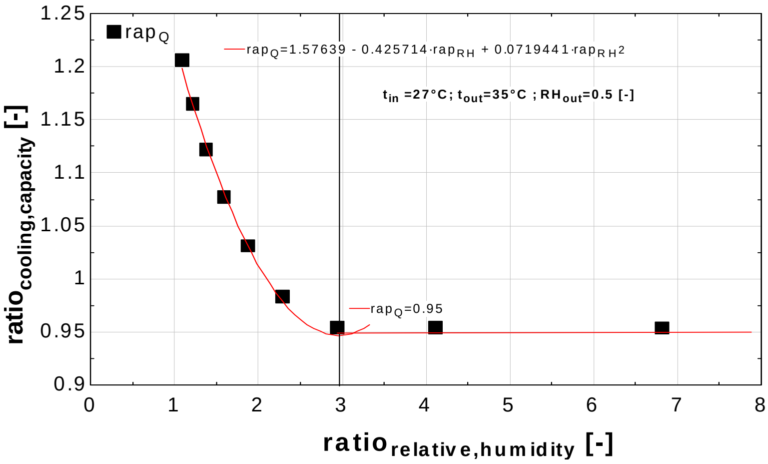


Figure A.30: Cooling capacity ratio as function of relative humidity ratio

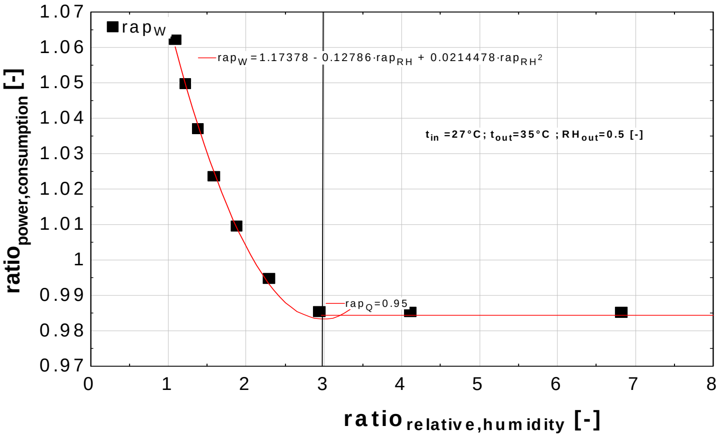


Figure A.31: Power consumption ratio as function of relative humidity ratio

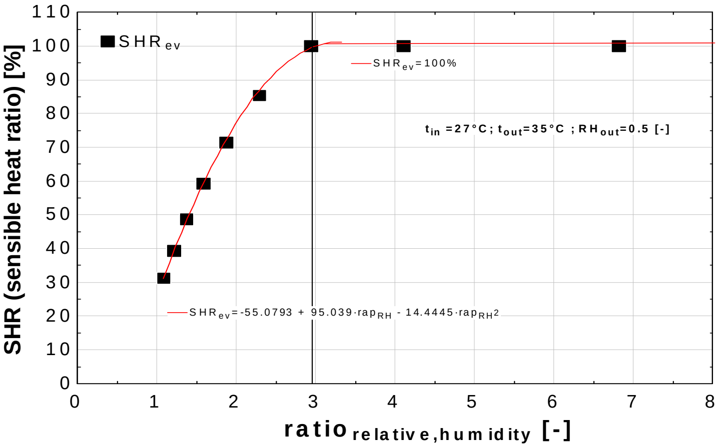


Figure A.32: Sensible heat ratio as function of relative humidity ratio

Two different zones can be distinguished and the following correction factors defined:

– from 0 to 2.9 of relative humidity ratio, where both cooling and power ratios present a (second degree) parabolic variation;

$$\mathrm{CoolingCapacityRHCorection} := 1.57639 - 0.425714 \cdot \mathrm{ratio}_{RH,in} + 0.0719441 \cdot \mathrm{ratio}_{RH,in}{}^{2}$$

$$\mathrm{PowerRHCorection} := 1.17378 - 0.12786 \cdot \mathrm{ratio}_{RH,in} + 0.0214478 \cdot \mathrm{ratio}_{RH,in}{}^{2}$$

$$\mathrm{SensibleHeatRatio} := -0.550793 + 0.95039 \cdot \mathrm{ratio}_{RH,in} - 0.144445 \cdot \mathrm{ratio}_{RH,in}{}^{2}$$

– above 2.9 of relative humidity ratio, where the cooling and power ratios are constant.

$$\mathrm{CoolingCapacityRHCorection} := 0.95$$

$$\mathrm{PowerRHCorection} := 0.985$$

$$\mathrm{SensibleHeatRatio} := 100$$

Fan speed correction factors

Constant values have been identified for the differences in fan speed values. Impact has been found quite low, cooling capacity vary between + 1 % and – 2 % while electric power varied between + 1 % and -5 % of the rated cooling capacity and air flow rate at rated air flow rate (intermediary fan speed).

Appendix B: Seasonal performance indices

B.1) Average operating conditions computed in task 4

B.1.1) Introduction

This part summarizes the average operating conditions used in task 4 under the form of reduced indices.

Both in heating and cooling mode, there are different operating modes (see part B.1.7). In parts B.1.2 to B.1.6, we focus on the active modes with main conclusions in part B.1.6 that include discussions of comments received. At the end, part B.1.7 is dedicated to the introduction of other power modes in the seasonal efficiency indices. Average operating conditions of single duct air conditioners are described in part B.1.8.

B.1.2) Load curves: generation and treatment

The dynamic simulations (performed by University of Athens within the frame of this study) of heating and cooling needs cover the EU-27 countries for 3 types of buildings (residence, office, retail), with different characteristics for new (after 2005) and existing buildings using TRNSYS 16 software.

Based on this, hourly load curves are gathered as explained hereafter to represent average European air conditioner operating conditions both in heating and cooling mode.

All the cooling load curves from building simulations have been reduced to 19 points, one point by 2°C range of outdoor temperature. Thus, a typical load is associated to every temperature range. Furthermore, an energy weight is also associated by summing all the energy needs occurring in this range.

All the heating load curves from building simulations have been reduced to two times (separating hours with setback and without) 23 points, with one point by 2°C range of outdoor temperature. Thus, two typical load curves are associated to every temperature range. Furthermore, energy weighting coefficients are also associated by summing all the energy needs occurring in the temperature ranges.

Then, the entire reduced load curves are weighted according to 2010 sales (in m² of conditioned surface) enabling to determine an average load curve and energy weights curve for all Europe. These data have been revised several times and final values are close from 2010 sales estimates as supplied by the industry.

Because of the high sales in Southern countries (Italy, Spain, Greece…), weighting coefficients and temperatures translate mainly Southern Europe conditions of operation. Nevertheless, Southern Europe means also relative important weight in countries like Spain and Italy where some regions may have relatively cold winter (Turin, Milan, Madrid). To cover correctly that important point, sales repartition have been made between Southern parts (with Seville and Rome climates) (70 %) and Northern parts (with Madrid and Turin climates) (30 %).

The balance load point is relatively low - in average 15 °C, and translates there is in general no management of building openings to introduce fresh air in summertime .

The final load curve obtained for the whole Europe in heating and cooling mode are displayed in Figure 1.

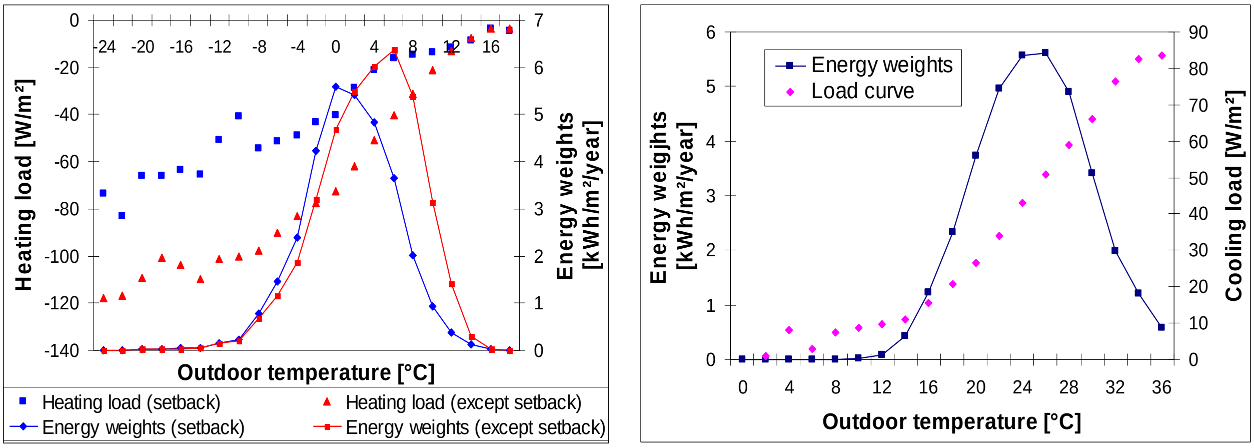


Figure 1: Heating and cooling EU average load curves.

B.1.3) Seasonal energy efficiency index in cooling mode ($SEER_{on}$)

The load curves must enable to calculate the performance of all air conditioners. Thus, the load must be divided by the sizing load, so that all the air conditioners may be compared on the same load and temperature repartition. It is assumed that the air conditioner sizing is made at 100 % for 35 °C, and temperature is not maintained above 35 °C (less than 1 % of the time). The maximal load occurs at about 35°C for Southern Europe but between 25 °C and 30 °C for others areas.

The reduction follows the same methodology as for the Eurovent ESEER following CEN TC113 request. This method, based on the IPLV approach of the ARI 550/590 standard, enables to avoid modelling the performances of the units with default values and enables to compare units' performances only under the basis of tested performances. At the same time, it does not require more testing points.

Finally, a typical load curve and associated energy weighting coefficients has been determined. It has been reduced to four points assuming the sizing is made at 100 % for 35 °C. The testing points (load, outdoor temperatures) and the weighting coefficients required for the EU weighted average SEER formula (Equation 1) are summarized in Table 1.

SEERon - Europe			
Testing points	**Part Load Ratio %**	**Outside Temperatures °C**	**Weighting coefficients**
A	100	35	9 %
B	75	30	30 %
C	50	25	37 %
D	25	20	24 %

Table 1: EU weighted average $SEER_{on}$ conditions

$$SEER_{on} = \frac{1}{\frac{A}{EER_A} + \frac{B}{EER_B} + \frac{C}{EER_C} + \frac{D}{EER_D}} \qquad \text{(Eq. 1)}$$

For single duct, SEERon indice is equal to the EER of the unit at 50 % part load capacity and in temperature testing conditions of the EN14511 standard rating conditions.

B.1.4) Seasonal energy efficiency index in heating mode ($SCOP_{on}$)

Sizing temperature

According to manufacturers, it appeared that the sizing identification methodology that led in average to size at – 2 °C was too optimistic and that average sizing was rather 2 °C than – 2 °C. Following these comments, the sizing hypothesis has been revised that leads in average to 2 °C.

Number of testing points

For the same reason as mentioned for the cooling mode, a four point index was also asked for the heating mode. This leads to a methodological problems regarding the reduction methodology used for cooling. Indeed, simulations included setback during night and unoccupied periods as the average situation in heating mode (it is the reason why two load curves appear in Figure 1) and the problem became to reduce two load curves to four points and not only one as in cooling mode. The reduction procedure and its effect are discussed hereafter, and we focus here on the 8 points index. The possibility to keep only four points is studied in part A.6.

Thus, each of the load curve (the setback one and the curve without setback) has been reduced to four points. The testing points (load, outdoor temperatures) and the weighting coefficients required for the EU weighted average SCOP formula (Equation 2) are summarized in Table 2.

$SCOP_{on}$ - Europe			
Testing points	**Temperatures**	**Part Load Ratio % (2)**	**Weighting coefficients**
A	-7	155%	4 %
B	-7	85%	5 %
C	-2	130%	9 %
D	-2	70%	12 %
E	2	100%	18 %
F	2	50%	16 %
G	7	55%	25 %
H	7	30%	11 %

Table 2: EU weighted average SCOP conditions

$$SCOP_{hp} = \frac{1}{\frac{A}{COP_A} + \ldots + \frac{H}{COP_H}} \qquad \text{(Eq. 2)}$$

Resistive heating

All heating needs cannot be covered by the unit and a share of the needs has to be covered by resistive heating. In order to propose a method that enables to compare machines, it seems necessary to take the resistive heating into account in the calculation of $SCOP_{on}$.
As explained in paragraph 4.5.2.3 of Task 4, it is assumed that the resistive heating share of the heating needs covered by electric heating only depends on the lowest operation temperature and on the ratio between the heating capacity at –7 °C and the rated capacity at 7 °C. The resistive demand compared to the total heating demand has been determined from simulations and can be expressed with Eq. 3 (positive or null). This gives an advantage to inverter driven units that can declare lower

capacity at 7 °C to increase the ratio r which leads to null R values. However, the sensitivity has been studied previously in task 4 and results are in good agreement with the complete simulations without penalizing unrealistically on-off units.

$$R = 0.064 + 0.002.T_{min} - 0.038.r \quad \text{(Eq. 3)}$$

With,
T_{min}: the minimum temperature of operation of the heat pump.
r=P(-7)/P(7): ratio of the heating capacity at –7 °C to the rated capacity at 7 °C.

Seasonal energy efficiency index of the system (heat pump + resistive heating)

The seasonal energy efficiency index of the system (heat pump plus resistive heating) can be calculated using Eq. 4.

$$SCOP_{on} = \frac{1}{\frac{1-R}{SCOP_{hp}} + R} \quad \text{(Eq. 4)}$$

with,
$SCOP_{hp}$ calculated with Eq.2 and the points given in Table 3.
R, the resistive demand compared to the total heating demand (Eq. 3)

$SCOP_{on}$ - Europe			
Testing points	**Temperatures**	**Part Load Ratio %**	**Weighting coefficients**
A	-7	Full load	4 %
B	-7	85%	5 %
C	-2	Full load	9 %
D	-2	70%	12 %
E	2	100%	18 %
F	2	50%	16 %
G	7	55%	25 %
H	7	30%	11 %

Table 3: EU weighted average SCOP conditions

B.1.5) Validation of the reduction methodology

A simple model of air conditioner is used to study the impact of the reduction process both in heating and cooling mode. As explained hereafter, a sensitivity analysis is carried out to compare the performances of the units (seasonal efficiencies) when taking the discretized complete load curves (Figure 1) and the indices with 4 or 8 points (A.3 and A.4).

Simplified air conditioner model

Heating mode

Operation within the frost range is assumed to occur when the outdoor temperature is between –7°C and Td. The heating capacity and the COP at full load evolve linearly as presented in the following figures. There is a discontinuity when the outdoor temperature is equal to Td. The performance drop (both in terms of COP and capacity) is taken into account by two coefficients that translate COP and capacity losses. It is supposed, as in ARI 210/240 standard that influence of frost can be neglected below – 7 °C.

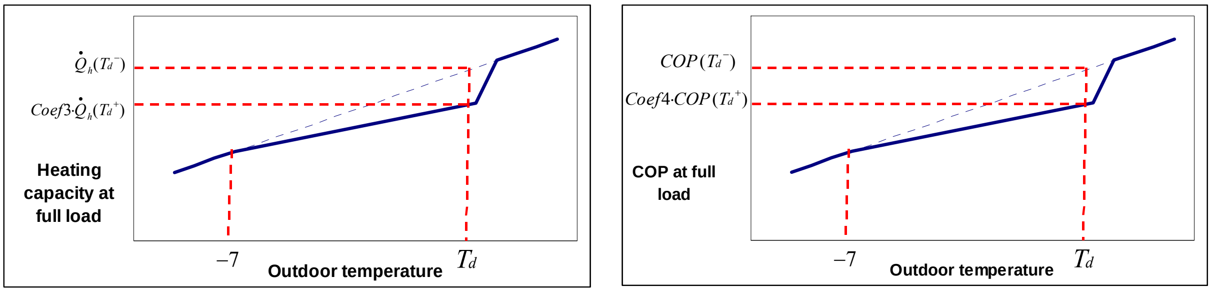


Figure 2 : evolution of heating capacity and COP according to temperature at full load

$$\dot{Q}_h(T_{out}) = \dot{Q}_h(7) \cdot (1-(7-T_{out}) \cdot Coef1) \qquad \text{if Tout} \geq \text{Td °C or Tout} \leq \text{-7°C}$$

$$COP(T_{out}) = COP(7) \cdot (1-(7-T_{out}) \cdot Coef2) \qquad \text{if Tout} \geq \text{Td °C or Tout} \leq \text{-7°C}$$

Where T_{out} is the outdoor temperature.

This enables to work out the full load capacity $\dot{Q}_h(X=1)$ and $COP(X=1)$ according to the outdoor temperature. Then the evolution of COP and capacity according to the load differs depending on whether the heat pump is equipped with an inverter compressor or an on – off one.

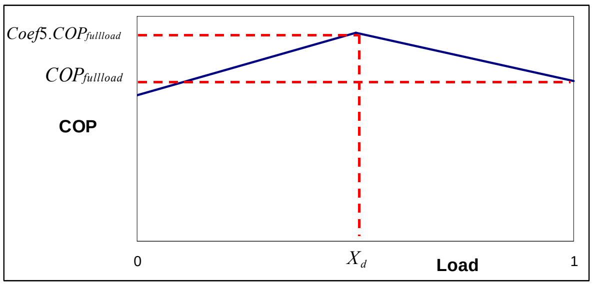


Figure 3 : Evolution of the COP according to the load

In order to study the impact of the reduction methodology on the seasonal efficiency of the units, several units are defined: their characteristics are summarized hereafter in Table 4.

Parameters	COP (7 °C)	T_d	$Coef1$	$Coef2$	$Coef3$	$Coef4$	C_D	Compressor control	$Coef5$	X_d
Unit 1	3.4	5	0.02143	0.01857	0.9	0.88	0.2	On/off	-	-
Unit 2	3.4	5	0.02143	0.01857	0.9	0.88	0.3	On/off	-	-
Unit 3	3.4	5	0.02143	0.01857	0.9	0.88	0.1	On/off	-	-
Unit 4	3.4	5	0.03143	0.02928	0.9	0.88	0.2	On/off	-	-
Unit 5	3.4	5	0.008571	0.01071	0.9	0.88	0.2	On/off	-	-
Unit 6	3.4	5	0.02143	0.0187	0.9	0.88	0.2	Inverter	0.5	1.3
Unit 7	3.4	5	0.02143	0.01857	0.9	0.88	0.2	Inverter	0.4	1.3
Unit 8	3.4	5	0.02143	0.01857	0.9	0.88	0.2	Inverter	0.6	1.3
Unit 9	3.4	5	0.02143	0.01857	0.9	0.88	0.2	Inverter	0.5	1.2
Unit 10	3.4	5	0.02143	0.01857	0.9	0.88	0.2	Inverter	0.5	1.5
Unit 11	3.4	5	0.02143	0.01857	0.9	0.88	0.3	Inverter	0.5	1.3

Unit 12	3.4	5	0.02143	0.01857	0.9	0.88	0.1	Inverter	0.5	1.3
Unit 13	3.4	5	0.03143	0.02928	0.9	0.88	0.2	Inverter	0.5	1.3
Unit 14	3.4	5	0.008571	0.01071	0.9	0.88	0.2	Inverter	0.5	1.3

Table 4: characteristics 14 inverter and on-off simplified models in heating mode

Heating capacity and COP evolution come from (SP, 2005) as reported in task 4: max, min and average slopes are used to translate different behaviours of COP and capacity with outdoor air temperature.

Cooling mode :

EER and cooling capacity are supposed to evolve linearly according to the outdoor temperature :

$$\dot{Q}_c(T_{out}) = \dot{Q}_c(35) \cdot (1 - (35 - T_{out}) \cdot Coef1)$$

$$EER(T_{out}) = EER(35) \cdot (1 - (35 - T_{out}) \cdot Coef\ 2)$$

Cycling follows the linear relationship used in the ARI 210/240 standard with the Cd coefficient.

In order to study the impact of the reduction methodology on the seasonal efficiency of the units, several units are defined: their characteristics are summarized hereafter in Table 5.

Parameters	**EER (35 °C)**	$Coef1$	$Coef2$	**C_D**	**Compressor control**	$Coef3$	X_d
Unit 1	2.9	- 0.0055	-0.009	0.2	On/off	-	-
Unit 2	2.9	- 0.0055	-0.009	0.3	On/off	-	-
Unit 3	2.9	- 0.0055	-0.009	0.1	On/off	-	-
Unit 4	2.9	- 0.003	-0.005	0.2	On/off	-	-
Unit 5	2.9	-0.011	-0.018	0.2	On/off	-	-
Unit 6	2.9	- 0.0055	-0.009	0.2	Inverter	0.5	1.3
Unit 7	2.9	- 0.0055	-0.009	0.2	Inverter	0.4	1.3
Unit 8	2.9	- 0.0055	-0.009	0.2	Inverter	0.6	1.3
Unit 9	2.9	- 0.0055	-0.009	0.2	Inverter	0.5	1.2
Unit 10	2.9	- 0.0055	-0.009	0.2	Inverter	0.5	1.5
Unit 11	2.9	- 0.0055	-0.009	0.3	Inverter	0.5	1.3
Unit 12	2.9	- 0.0055	-0.009	0.1	Inverter	0.5	1.3
Unit 13	2.9	- 0.003	-0.005	0.2	Inverter	0.5	1.3
Unit 14	2.9	-0.011	-0.018	0.2	Inverter	0.5	1.3

Table 5: characteristics 14 inverter and on-off simplified models in cooling mode

Results in cooling mode

Seasonal efficiency has been calculated for the 14 units with the 4 points index as well as with the total load curve (Figure 1). Results are given in Figure 4 and Table 6. Average difference is about 1 % for ON/OFF units and 1.4 % for inverter units, with the 4 points index underestimating slightly the performances.

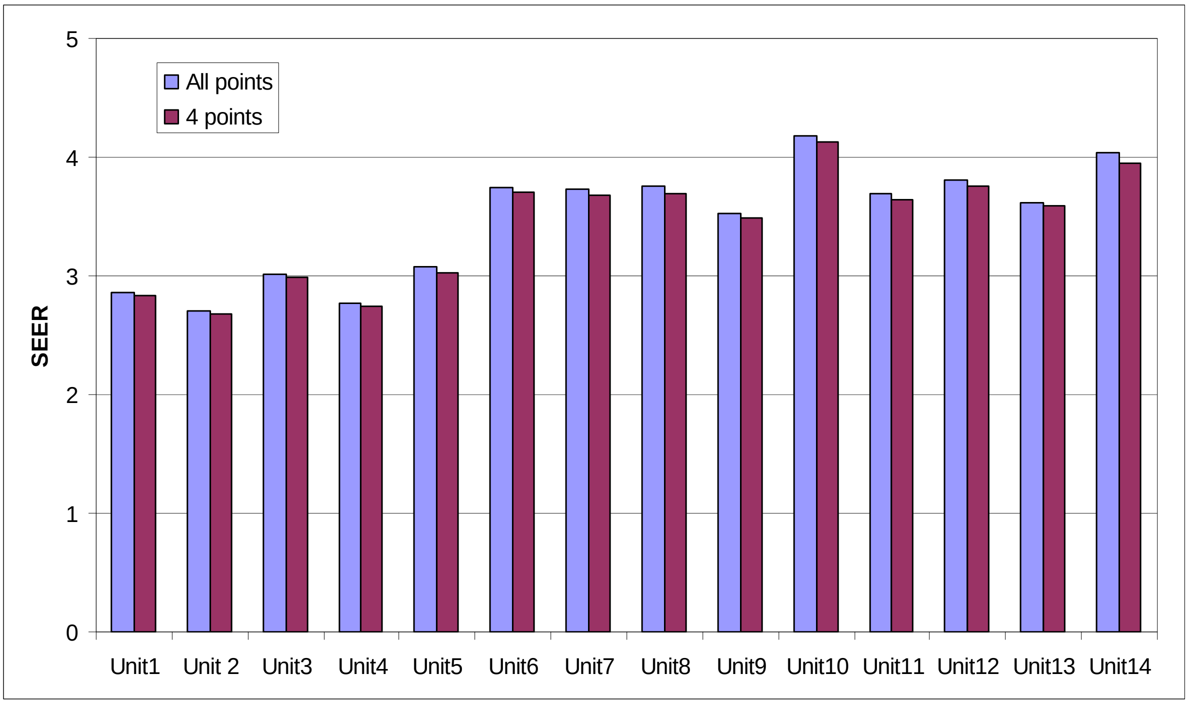


Figure 4: Comparison between the seasonal energy efficiency obtained with the total European load curve (Figure 1) and the four points index for simplified models in cooling mode

	Unit 1	Unit 2	Unit 3	Unit 4	Unit 5	Unit 6	Unit 7	Unit 8	Unit 9	Unit 10	Unit 11	Unit 12	Unit 13	Unit 14
Absolute Difference [%]	1,0	1,0	1,0	0,6	1,8	1,2	1,5	1,9	1,2	1,4	1,2	1,3	0,7	2,4

Table 6: bias induced by the reduction process for simplified models in cooling mode

Results in heating mode

Seasonal efficiency has been calculated for the 14 units with the 8 points index as well as with the total load curve (Figure 1). Results are given in Figure 5 and Table 7. Average difference is about 1.9 % for ON/OFF units and 1 % for inverter units, with the 4 points indice underestimating slightly the performances.

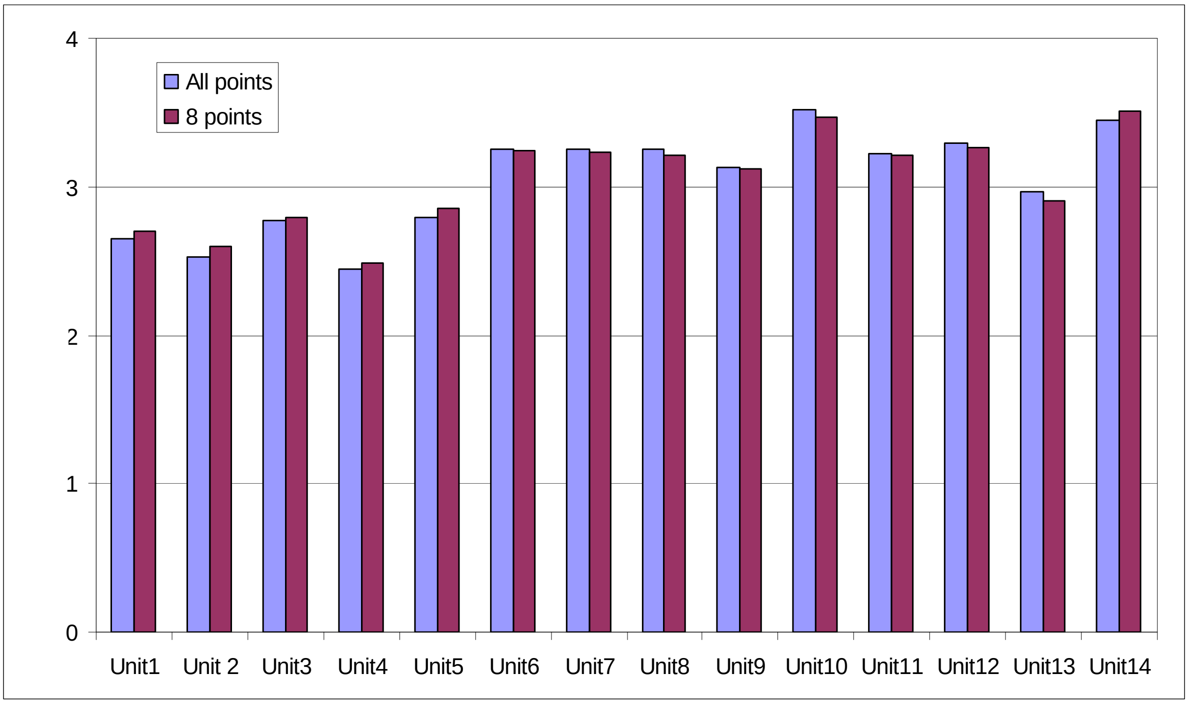


Figure 5: Comparison between the seasonal energy efficiency obtained with the total European load curve (Figure 1) and the eight points index for simplified models in heating mode

	Unit 1	Unit 2	Unit 3	Unit 4	Unit 5	Unit 6	Unit 7	Unit 8	Unit 9	Unit 10	Unit 11	Unit 12	Unit 13	Unit 14
Absolute Difference [%]	1,89	3	0,67	1,81	2,22	0,53	0,77	1,2	0,34	1,55	0,14	0,98	2,14	1,78

Table 7: bias induced by the reduction process for simplified models in heating mode

B.1.6) Equivalent formulation with 4 points indices

Cooling mode

The 4 points index obtained in cooling mode ($SEER_{on}$) is suitable to assess performance of air conditioners since it requires only four testing points and is very close to the energy efficiency ratio obtained when considering the total load curve (B.1.5).

However, a seasonal performance index in cooling mode already exists in Europe for chillers. It could get complicated to have two different indices, even with different names, to be explained to the customer. On this basis, manufacturers advised to get only one which is the present proposal in the CEN draft standard.

In order to evaluate the impact of adopting the existing standard instead of the conditions identified in the present analysis, a comparison between the Eurovent ESEER index and the index proposed in this study (B.1.3) has been carried out by calculating both indices for the 14 units described in part A.5. The results are shown in Figure 6 and given in Table 8 for the 14 units. In average, the proposed index is about 1.1 % higher than the ESEER for on/off units and 0.56 % lower for inverter units.

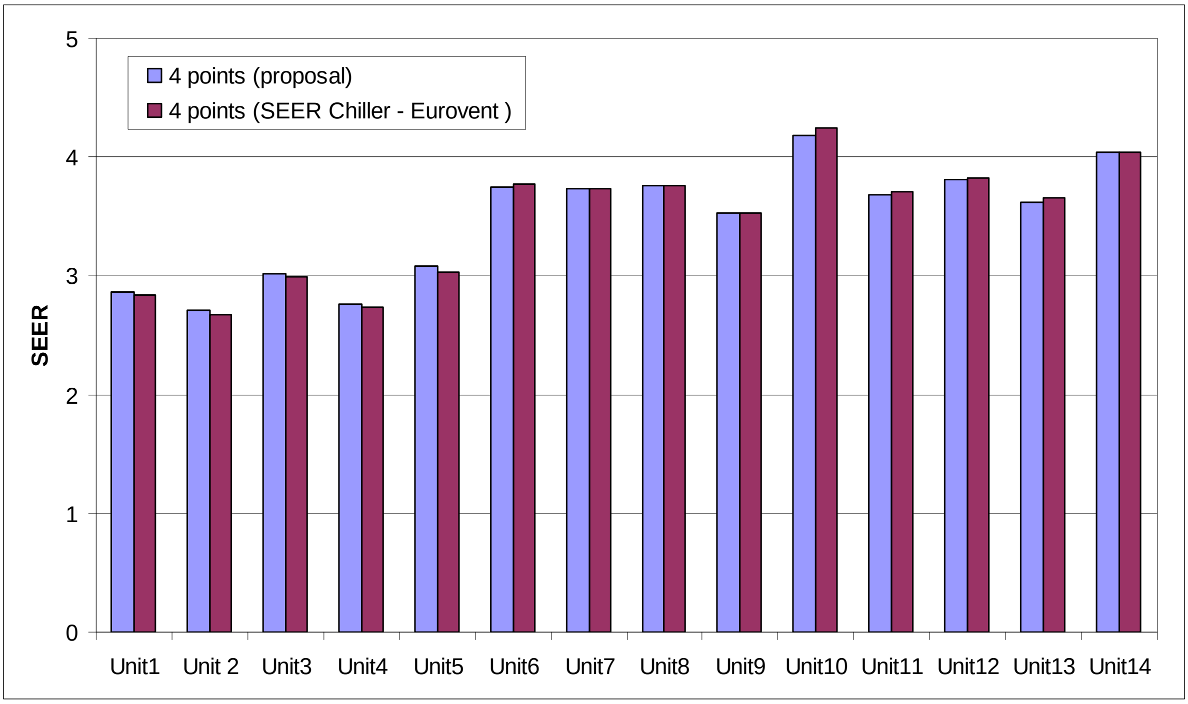


Figure 6: Comparison between the Eurovent ESEER and the four points index obtained within this study for the 14 simplified models

	Unit 1	Unit 2	Unit 3	Unit 4	Unit 5	Unit 6	Unit 7	Unit 8	Unit 9	Unit 10	Unit 11	Unit 12	Unit 13	Unit 14
Absolute Difference [%]	1,1	1,3	0,9	0,8	1,6	0,6	0,2	0,0	0,2	1,4	0,6	0,6	1	0,1

Table 8: difference between the Eurovent ESEER and the four points index obtained within this study for the 14 simplified models

It thus appears that the average operating conditions for air to air conditioners give seasonal cooling performance indices that are close to the ones obtained with the reference Eurovent index being used for chillers.

Heating mode

Manufacturers asked to reduce the total load curve to four points in order to avoid too much testing points and to avoid testing appliances in frost conditions and at part load capacity. Consequently, they imposed the temperatures (that are the temperatures already in the ISO 5151 standard in heating mode except the 12 °C dB (10 °C wB) that is to be added).

In order to reduce the 8 points curve to 4 points, the points of the setback curve are reported on the no setback curve. Weighting coefficients of lower loads are corrected by the load ratio, at equal energy weight. Results are presented in Table 9.

SCOP		
Temperatures	**Part Load Ratio %**	**Weighting coefficients**
-7	Full load	18 %
2	100	46 %
7	60	27 %
12	30	9 %

Table 9: 4 points index in heating mode

As can be seen below (Figure 7 and Table 10), the reduction generates important bias since it increases the performance in heating mode of 5.45 % in average for on/off units whereas it decreases the performance of 5.73 % in average for inverter units. This is in good agreement with the method used to report setback points: the weight on part load decreased to the benefit of ON/OFF units.

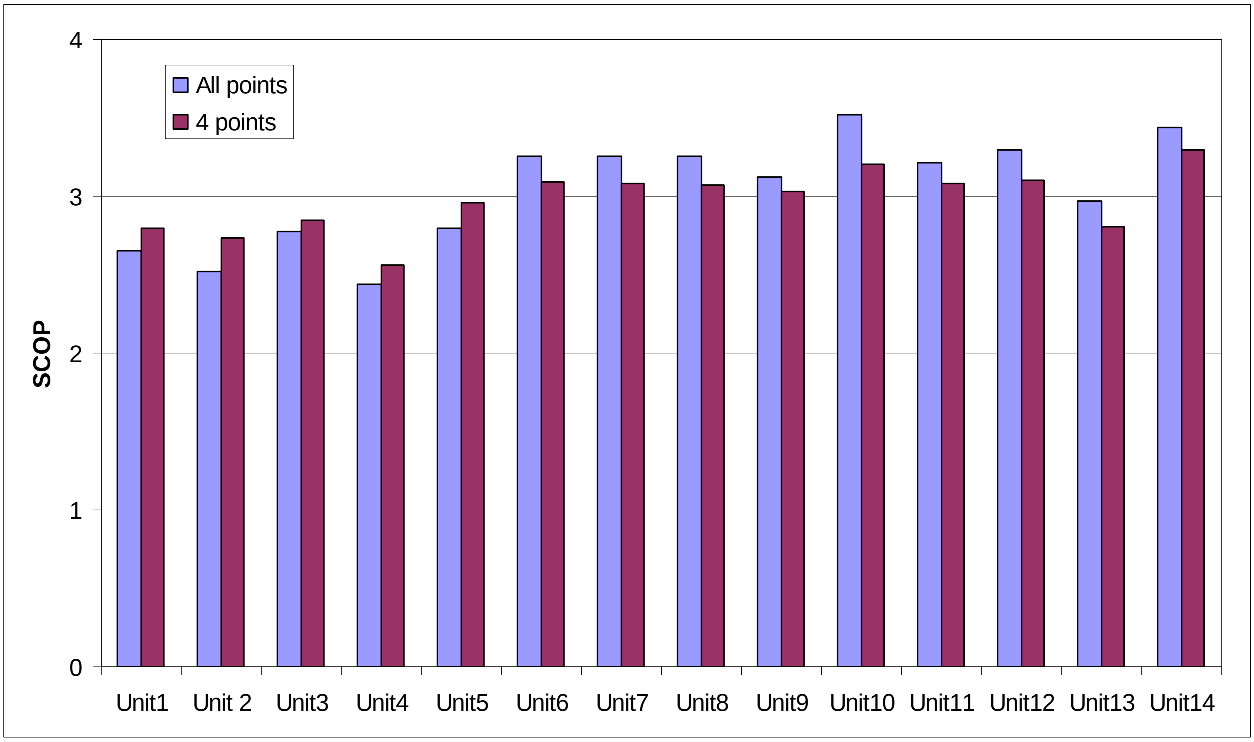


Figure 7: Comparison between the seasonal energy efficiency obtained with the total European load curve (Figure 1) and the four points index for simplified models in heating mode

	Unit 1	Unit 2	Unit 3	Unit 4	Unit 5	Unit 6	Unit 7	Unit 8	Unit 9	Unit 10	Unit 11	Unit 12	Unit 13	Unit 14
Absolute Difference [%]	5,45	8,26	2,69	4,92	5,94	5,38	5,72	5,99	3,23	10,07	4,56	6,24	5,87	4,55

Table 10: bias induced by the reduction process for simplified models in heating mode

B.1.7) Correction of indices to include other power modes

Reference cooling capacity
Rated capacity as defined by the standard rating conditions given in EN 14511 and noted P_C.

Reference heating capacity
Rated capacity as measured at „A“ temperature conditions as defined in EN 14511 (standard rating or application rating conditions as applicable) and noted P_H.

Equivalent full load hours in cooling mode
Ratio between the cooling requirements of the unit to the unit standard rating capacity in cooling mode according to EN14511 (2004).

Equivalent full load hours in heating mode
Ratio between the heating requirements of the unit to the unit standard rating capacity in heating mode according to EN14511 (2004).

Thermostat-off mode
The air conditioner is operational in heating or cooling mode but inside temperature is lower (higher in heating mode) than the set point. The impact of thermostat-off mode is included in the cycling low and additionally in the thermostat-off consumption corresponding to hours with no cooling or heating load while cooling (or heating) is required by the user. Average electric power in this mode is noted P_{TO} and hours of operation in this mode are noted H_{TO}.

Passive standby mode
The air conditioner is not operational; it can be reactivated either by control device or by timer. This mode corresponds to hours with no occupancy in the building during the cooling or heating season. Average electric power in this mode is noted P_{SB} and hours of operation in this mode are noted H_{SB}.

Off mode
The air conditioner has been switched off by the user, is not operational and cannot be reactivated nor by control device or by timer. This mode corresponds to hours outside the cooling and/or heating season. Average electric power in this mode is noted P_{OFF} and hours of operation in this mode are noted H_{OFF}.

Crank heater operation
The crankcase heater operates when the compressor is off and the outdoor temperature is lower than a given value. Other parameters such as the compressor or the heat exchanger temperature may also be included into the control and have an impact on its energy consumption. Average electric power in this mode is noted P_{CK} and hours of operation in this mode are noted H_{CK}.

The calculation of the SEER, SCOP and APF is given by the following equations (5 to 7):

$$SEER = \frac{H_C.P_C}{\frac{H_C.P_C}{SEER_{on}} + H_{TO}P_{TO} + H_{SB}P_{SB} + H_{OFF}P_{OFF}} \quad \text{[Eq.5]}$$

$$SCOP = \frac{H_H.P_H}{\frac{H_H.P_H}{SCOP_{on}(+R)} + H_{TO}P_{TO} + H_{SB}P_{SB} + H_{CK}P_{CK} + H_{TO}P_{TO}} \quad \text{[Eq.6]}$$

$$APF = \frac{H_C.P_C + H_H.P_H}{\frac{H_C.P_C}{SEER_{on}} + \frac{H_H.P_H}{SCOP_{on}} + H_{TO}P_{TO} + H_{SB}P_{SB} + H_{OFF}P_{OFF} + H_{CK}.P_{CK}} \quad [Eq.7]$$

Hours to be used depends whether the unit is an air conditioner, a reversible heat pump or a heating only heat pump and are gathered in table 12.

PRODUCTS		Air conditioner	Reversible heat pump	Heating only heat pump
Cooling, H_C	h	315	315	0
Heating, H_H	h	0	610	610
Off mode, H_{OFF}	h	5000	0	3600
Standby, H_{SB}	h	2300	2300 (SCOP: 0 // SEER: 2300)	0
Thermostat-off, H_{TO}	h	700	3300 (SCOP: 2600 // SEER: 700)	2600
Crankcase, H_{CK}	h	See Table 13	See Table 13	See Table 13

Table 12: Time spent in the different power modes to be used to calculate APF, SEER and SCOP indices

NOTA: Operation of crankcase has been assumed to be controlled at 10 °C in heating mode. With control at 0 °C, hours of operation could be divided by 10, while for 20 °C set point, it is multiplied by 2 and that without control, electric crankcase heater operations would be about 5500 hours yearly, multiplying related energy consumption by nearly a factor 4.
Linear interpolation between these values can be used to assess the hours of operation of crankcase heaters that are controlled as a function of outdoor air temperature :

Table 13 – Relation between HCK and TCK

T_{CK} (°C)	0	10	20	30
Air conditioners H_{CK} (hours)	410	3090	6680	7850
Reversible heat pumps H_{CK} (hours)	110	1160	4750	5800
Heating only heat pumps H_{CK} (hours)	110	1160	4900	6610

B.1.8) Average operating conditions of single duct air conditioners

A few words on the simulations

Single duct were sized as for Cooling Only air conditioners. The increase of ventilation rate was not taken into account to do so. Indeed, Single Duct appliances and split systems differ regarding cooling needs since the first ones introduces outdoor air inside the room. However, the effect of outside air introduction has been computed.
Cooling set point was set at 25 for offices and dwellings in all climates except warm countries where it was set at 26 °C. For shop, the cooling set point temperature is 23°C.

2010 sales figures, from task 2

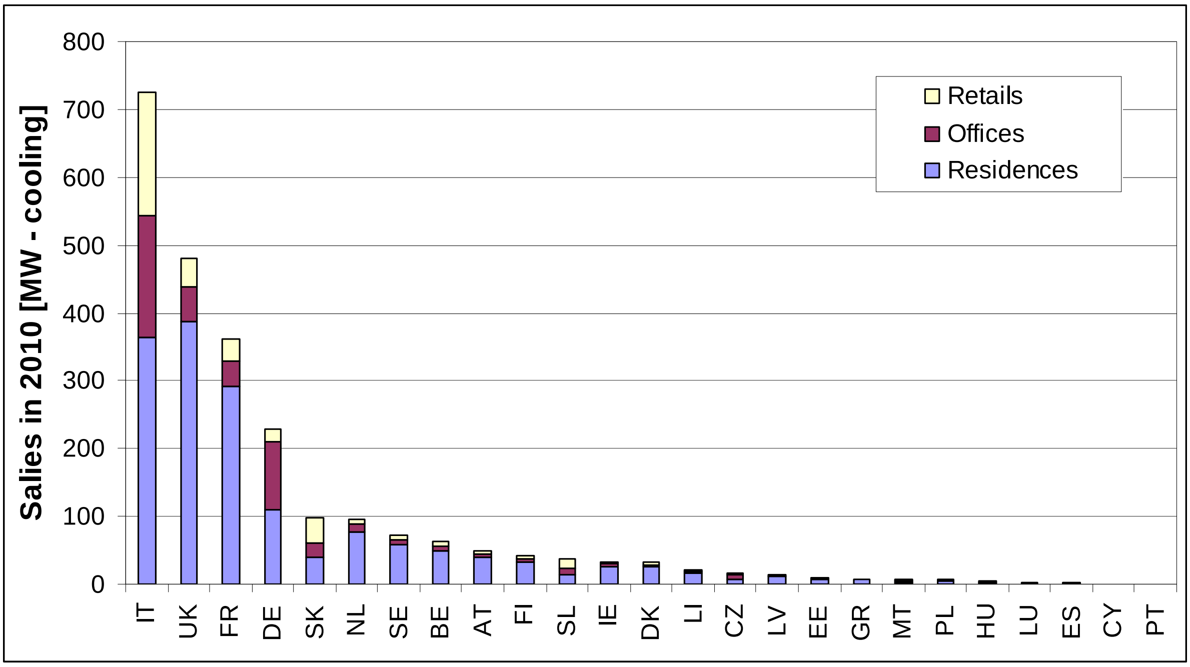


Figure 8 : Sales of single duct air conditioners in Europe, from Task 2

Italy, UK, France and Germany represent about 75 % of the sales in Europe. On the other hand, 65 % of the sales are used in the residential sector. We are going to focus in the ten main countries (regarding sales figures) in what follows.

Doe Single Ducts enable to maintain the set point, according to task 4 simulations?

During the warmest days, single duct cannot always enable to reach the required the setpoint indoor temperature. In the figure below, 2 units are compared, one split and one single duct air conditioner for the same simulation for an Italian shop. Delivered total capacity are in W/m2. Tout is the outdoor temperature, Tin the indoor temperature when the shop is cooled with a split, Qc is cooling power delivered by the split and Qc SD the cooling capacity delivered by the single duct air conditioner. With a set point at 23 °C, it appears clearly that the single duct air conditioner, by introducing hot air (outside temperature is higher than the set point) increases the load. At about 12 am, it comes to maximum capacity and remains stable until the shop closes.

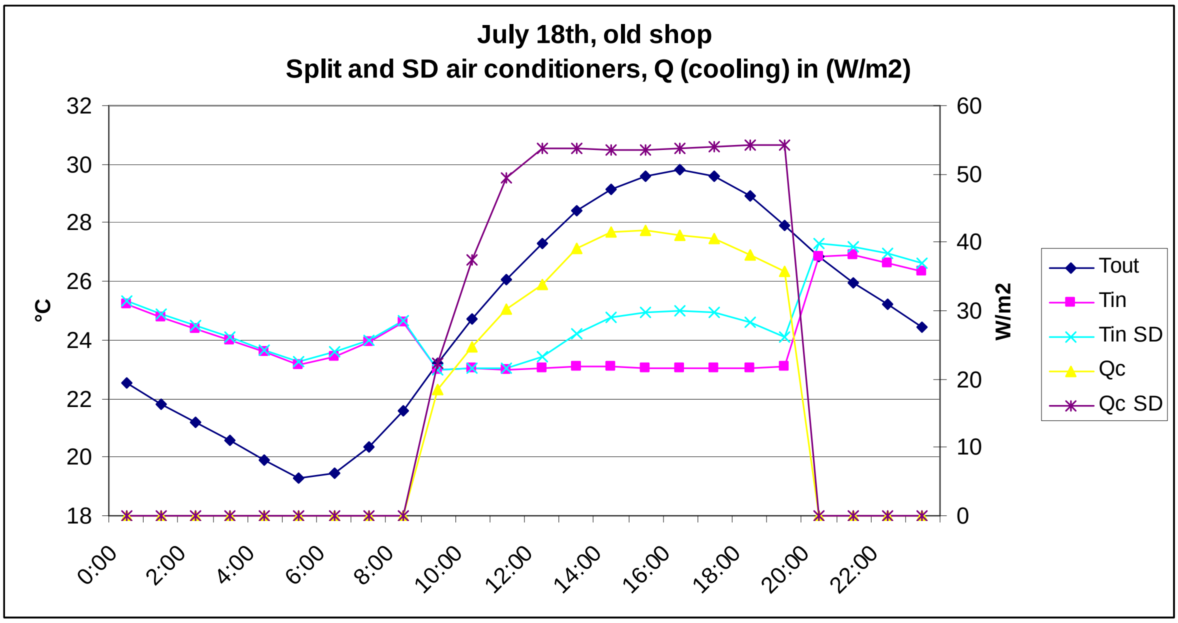


Figure 9 : Comparison of a cooling hot day in a shop in Italy with single duct and split air conditioner

It should be added however than with the timetables of the residential sector, single ducts can help to maintain the indoor temperature to the required set point but not at the hottest moment of the day. In the case of the same hot day in Rome, the figure below shows that the set point can be reached (because outdoor temperature is about the setpoint).

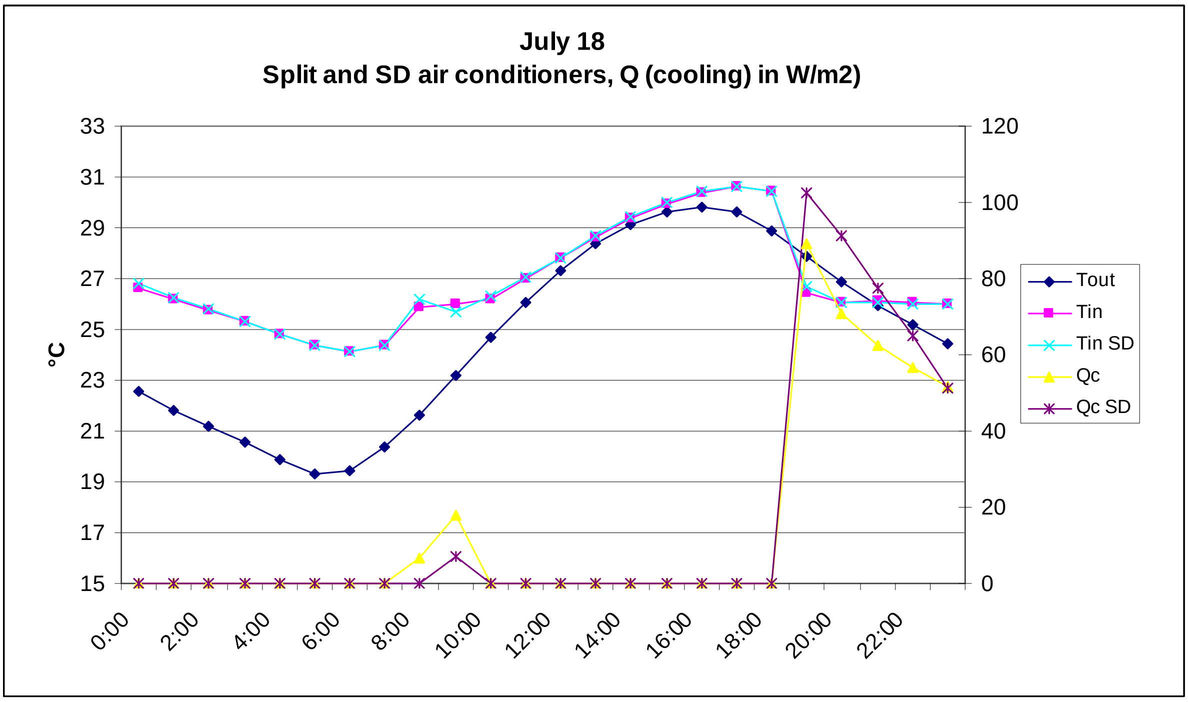


Figure 10 : Comparison of a cooling hot day in a dwelling in Italy with single duct and split air conditioner

Does it mean that single ducts always work at full load, according to task 4 simulations?

Of course, single duct air conditioners do not work only during the hottest days but are controlled to maintain a desired indoor temperature and consequently work when indoor temperature are above the set point within the occupancy hours. Sept 22 in the Italian shop is a mild day but with internal loads, air conditioning is required to maintain the set point. Qc and Qc SD are represented as the ratio of the units cooling capacity over the rated capacity. Average load is about 40 % for this day.

As shown below, the ventilation effect has free cooling effect for Tout below set point and increases the load in the reverse situation.

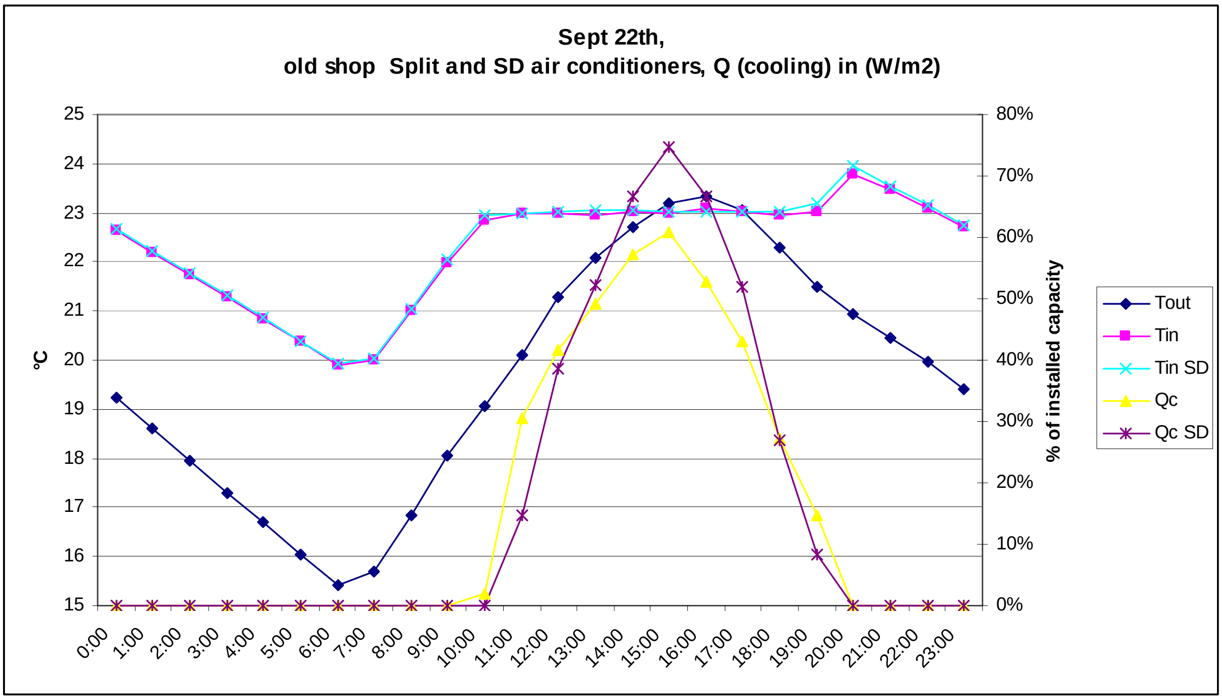


Figure 11 : Comparison of a cooling mild day in a shop in Italy with single duct and split air conditioner

Average operating temperature

An energy weighted average of operation temperatures has been calculated for every climate and every building type, the results are presented hereafter for the ten main (in terms of sales) countries. The hottest days of the year translate only by a little deviation of the average temperature of operation above the set points.

In order to determine a European average operation temperature, a sales weighted average of the national temperatures has been calculated which leads to **24.8 °C**.

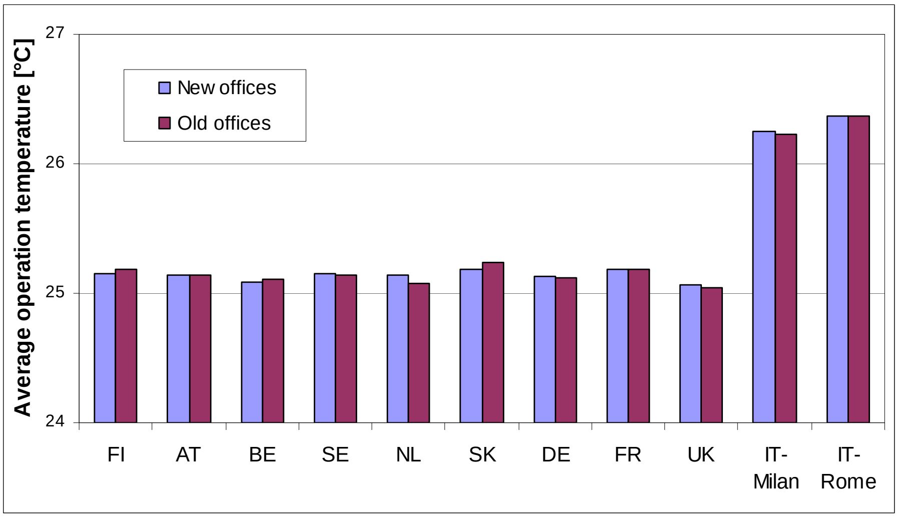


Figure 12 : Average operating temperature of single duct air conditioners, offices

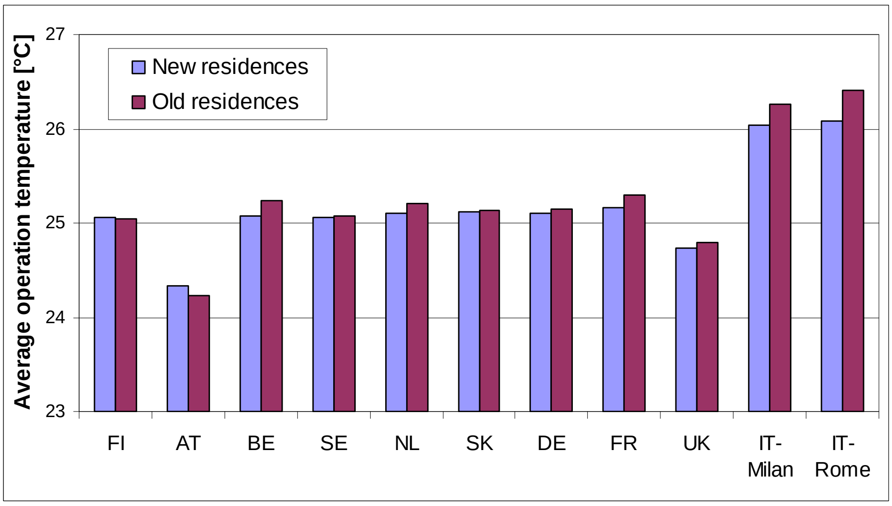


Figure 13 : Average operating temperature of single duct air conditioners, dwellings

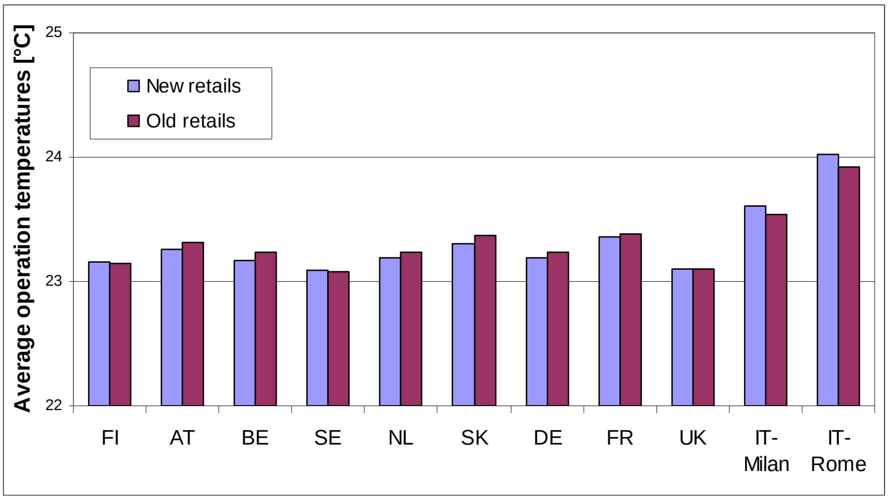


Figure 14 : Average operating temperature of single duct air conditioners, shops

Average operating load

An energy weighted average of operation loads has been calculated for every climate and every building type, the results are presented hereafter for the ten main countries (in terms of sales).
In order to determine a European average operation load, a sales weighted average of the national loads has been calculated which leads to **55.4 %**

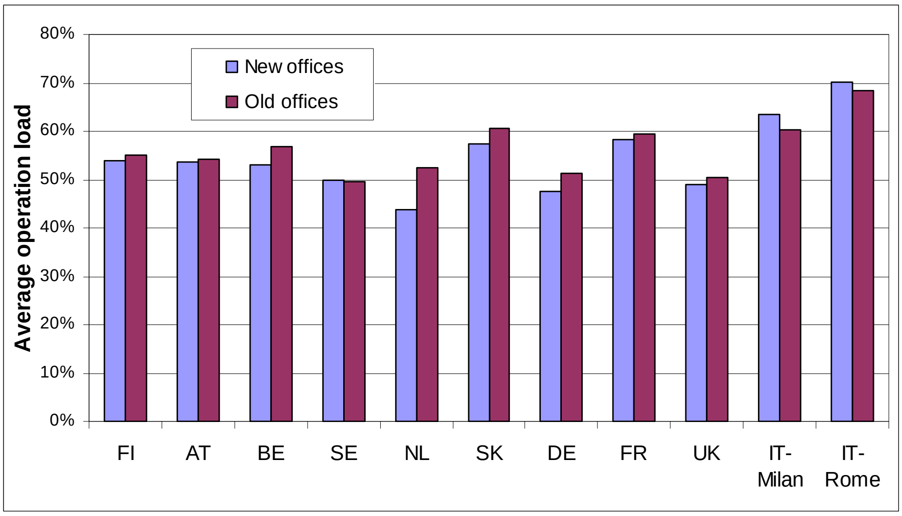


Figure 15 : Average operating load ratio of single duct air conditioners, offices

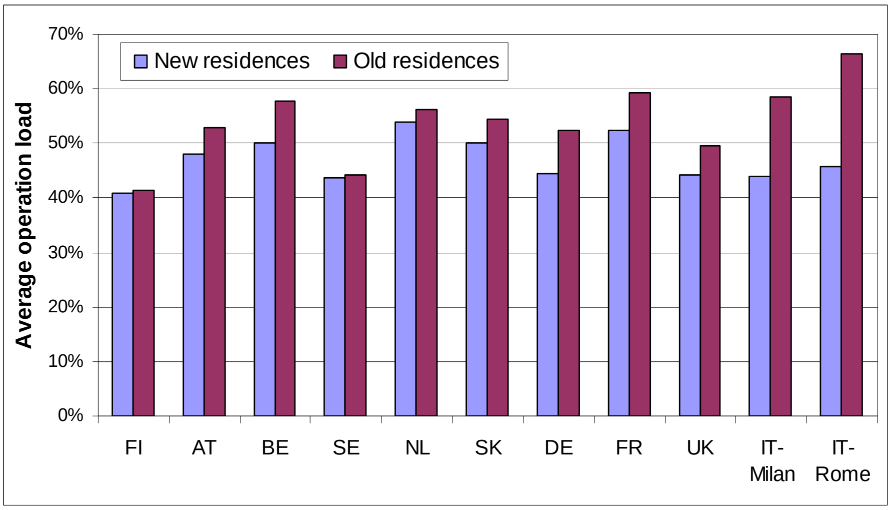


Figure 16 : Average operating load ratio of single duct air conditioners, dwellings

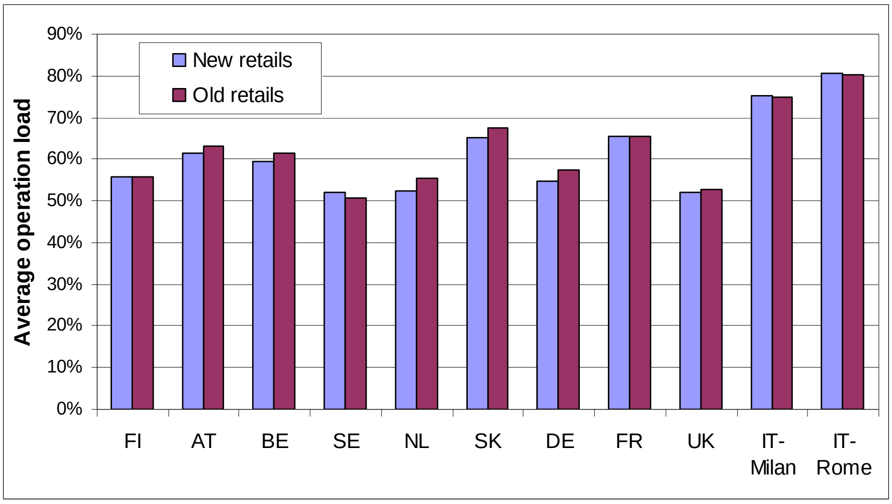


Figure 17 : Average operating load ratio of single duct air conditioners, shops

Average operating humidity

An energy weighted average of operating humidity has been calculated for every climate and every building type, the results are presented hereafter for the ten main countries (in terms of sales).
In order to determine a European average operation humidity, a sales weighted average has been calculated which leads to **55.3 %.**

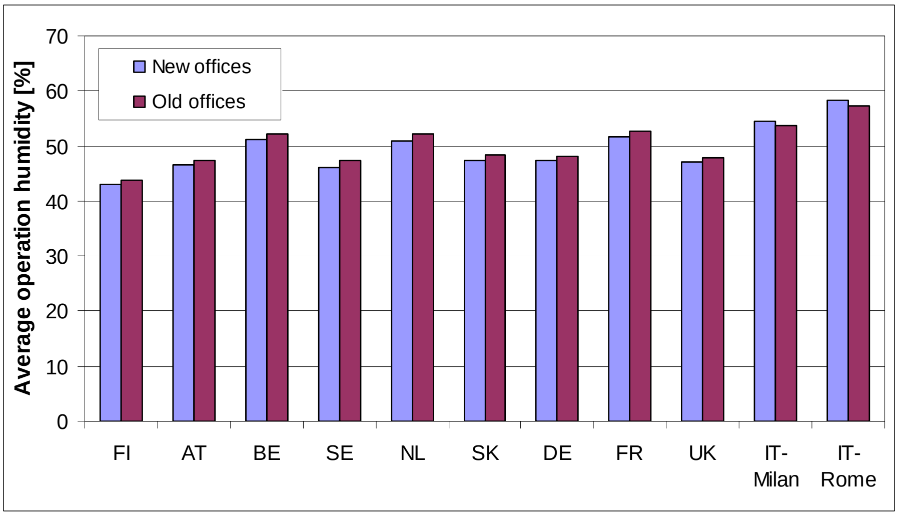


Figure 18 : Average operating relative humidity of single duct air conditioners, offices

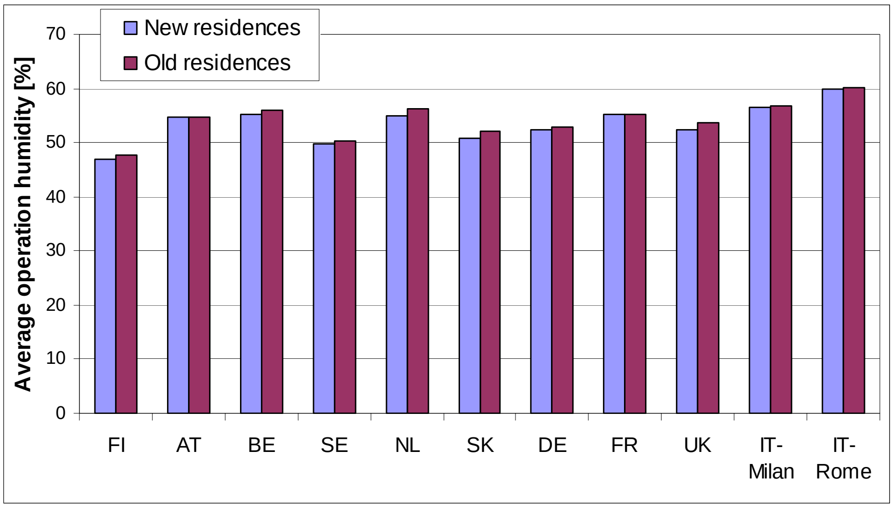


Figure 19 : Average operating relative humidity of single duct air conditioners, shops

Figure 20 : Average operating relative humidity of single duct air conditioners, shops

B.2) Seasonal performance factor in heating mode

A number of caveats appeared along the study with the four points index proposed in heating mode and another solution was required: amongst others, compatibility with lot 1 results, reduction to 4 points, market sales validity …

B.2.1) Climate conditions

Reversible air conditioners are a heating means as another one and as such should have ratings comparable to other heating means in order consumer may choose amongst other solutions. Whether a heat pump market specific seasonal performance indice was to be kept to rate the performances of heat pumps then for instance for solar products we would have to choose something between Austria and Greece. This is not possible whether the goal is to compare all heating means.

Regarding weather data, it was agreed it was required to adopt typical year profiles in order to correctly translate the temperature sensitivity of heat pump performances.The Energy Plus ® hourly data, of easy access and covering largely Europe, that were computed by ASHRAE - IWEC set of data were chosen. These files represent a typical year built with real months from 1982 to 1997 to get representative climates (for energy in buildings computations) of many European cities.

Strasbourg has been identified as the closest climate to the "average" EU climate, average here is to be understood as "population averaged".

Here is the comparison between Strasbourg weather file and PVGIS (JRC 1995-2003) weather data on average day per month. On average Strasbourg IWEC 1982-1997 is 0,7 colder than EU-27 1995-2003 ; 0,3 degrees is climate warming and 0,4 degrees because capitals are more at sea than avg. population.

Avg. EU-27 *Altitude comparable*	**Jan**	**Feb**	**Mar**	**Apr**	**May**	**Jun**	**Jul**	**Aug**	**Sept**	**Oct**	**Nov**	**Dec**	
23-7h	1,9	2,6	4,0	6,8	11,1	14,6	16,2	15,9	12,1	9,7	5,9	2,5	
7-9h	1,9	2,8	5,1	8,8	14,0	17,5	19,2	19,1	14,4	10,4	6,0	2,5	
9-16h	4,2	6,3	9,2	13,1	18,1	21,6	23,6	24,1	19,1	14,3	8,6	4,4	
16-21h	4,3	6,3	8,8	12,4	17,5	21,0	23,1	23,4	18,0	13,5	8,3	4,3	
21-23h	3,1	4,6	6,5	9,3	14,4	18,0	20,0	19,6	14,5	11,4	7,2	3,3	**Y ave**
	3,2	4,6	6,8	10,2	15,0	18,5	20,4	20,4	15,8	12,0	7,3	3,5	
Difference EU-27 - Strasbourg	**Jan**	**Feb**	**Mar**	**Apr**	**May**	**Jun**	**Jul**	**Aug**	**Sept**	**Oct**	**Nov**	**Dec**	
23-7h	0,4	2,0	0,6	0,2	0,0	0,9	0,5	1,1	0,7	0,7	2,7	0,8	
7-9h	0,2	1,9	0,9	0,1	0,6	0,7	0,9	1,5	1,5	1,0	2,4	0,9	
9-16h	0,5	2,5	-0,3	-2,1	-1,6	-2,2	-1,7	-2,6	-2,0	-0,3	1,3	0,2	
16-21h	1,7	3,8	1,3	0,1	1,1	1,5	2,3	1,2	1,3	1,6	3,0	1,1	
21-23h	1,2	3,4	1,2	0,7	1,2	2,2	2,6	2,3	1,2	1,3	3,1	0,9	**Y ave diff**
	0,7	2,7	0,5	-0,5	-0,1	0,2	0,5	0,2	0,1	0,7	2,4	0,7	

Table 14: Average day temperature comparison between Strasbourg IWEC data and "population average" EU climate – PVGIS data - for average monthly day profiles, in °C

The bin representation of Strasbourg is supplied in table 15. The heating season is 8 months: Winter starts October 1st and ends May 31st.

The bin hours are reported in table 16 (the 2 °C bin hours gathers hours during the heating season with 2 °C +/- 0.5 K).

Application climatic conditions

Since heat pump product performances are sensitive to climate, either directly like heat pumps or solar heaters, or indirectly like boilers that will be sensitive to average load ratio or increased share of hours without operation because of higher parasitic energy consumption, there is a need to indicate the end-user with climatic sensitivity. Three main climate zones have been considered there, Northern Europe with Helsinki as the reference climate, Central Europe with Strasbourg and Southern Europe with Athens.
The binned data for Athens and Helsinki are reported in the next table. The heating season is 6 months for Athens: Winter starts November 1st and ends May 1st. The heating season is 9 months for Helsinki: Winter starts September 1st and ends May 31st.

		Athens	**Strasbourg**	**Helsinki**
Bin number	**Outdoor temperature °C**	**6 months**	**7 months**	**9 months**
1	**-30**	0	0	0
2	**-29**	0	0	0
3	**-28**	0	0	0
4	**-27**	0	0	0
5	**-26**	0	0	0
6	**-25**	0	0	0
7	**-24**	0	0	0
8	**-23**	0	0	0
9	**-22**	0	0	1
10	**-21**	0	0	6
11	**-20**	0	0	13
12	**-19**	0	0	17
13	**-18**	0	0	19
14	**-17**	0	0	26
15	**-16**	0	0	39
16	**-15**	0	0	41
17	**-14**	0	0	35
18	**-13**	0	0	52
19	**-12**	0	0	37
20	**-11**	0	0	41
21	**-10**	0	1	43
22	**-9**	0	25	54
23	**-8**	0	23	90
24	**-7**	0	24	125
25	**-6**	0	27	169
26	**-5**	0	68	195
27	**-4**	0	91	278
28	**-3**	0	89	306
29	**-2**	0	165	454
30	**-1**	0	173	385
31	**0**	0	240	490
32	**1**	0	280	533
33	**2**	3	320	380
34	**3**	22	357	228
35	**4**	63	356	261
36	**5**	63	303	279
37	**6**	175	330	229
38	**7**	162	326	269
39	**8**	259	348	233
40	**9**	360	335	230

41	**10**	428	315	243
42	**11**	430	215	191
43	**12**	503	169	146
44	**13**	444	151	150
45	**14**	384	105	97
46	**15**	294	74	61

Table 15: Binned winter data of selected representative climates of Europe for heating conditions (source ASHRAE IWEC data file)

B.2.2) Building heating requirements

Regarding the load curve, a simple straight line is used to draw the average heat demand of the building versus the outdoor air temperature from a zero load at 16 °C design heat requirement.

$BL(T_j) = Pdesign*(T_j-16)/(Tdesign-16)$

Where
- $1 \leq j \leq 46$
-Pdesign is the maximum heating requirement and Tdesign is the design temperature for a specific location.
-Design temperatures are –10 °C for Strasbourg, 2 °C for Athens and –20 °C for Helsinki.

Pdesign might be either:
- a declaration of the manufacturer for certain climate conditions. In that case, proposed profiles follow.
- or standard sizing conditions based on testing points available in the prEN14825 standard for instance –7 °C for Helsinki, 2 °C for Strasbourg and 2 °C for Athens, meaning Pdesign is the rated capacity at 2 °C for Athens, is 186 % of the rated capacity @ 2 °C for Strasbourg and 156 % of the rated capacity @ -7 °C for Strasbourg.

Taking the balance temperature of 2 °C in Strasbourg corresponds to sizing the heat pump @ -10 °C with 75 % resistive heating at design capacity that seems to be more resistive heating than common practice – resistive heating capacity installed is about 115 % of the rated heating capacity of the heat pump at 7 °C. The impact of sizing the heat pump for different outdoor temperature (balance point temperatures) is shown in the following table. As already mentioned, this design temperature has been kept because of potential testing problems in frost and part load conditions.

Balance point	**@2**	**@-2**	**@-7**
Share of resistive heating @ -10 °C	77%	63%	33%
Pdesign @ -10 °C (kW)	6,0	3,8	2,1
Ratio resitive heating / rated heating capacity	115%	60%	18%

Table 0-1: Comparison of different balance point for sizing heat pumps, Strasbourg climate

For the buildings models defined in Lot 1, Pdesign for a specific building is computed following the methodology of the EN 12831- 2003 *Heating Systems in Buildings – Method for calculation of the design heat load*. The standard uses a standard heat load calculation without solar and internal gains, i.e. the calculation of Pdesign follows the formula:

Pdesign = 0,001* 1,15*1,2* (Tin -Tout)*V*{ah*(qv*(1-qrec)+qinf)+AV*U},

where

Tin, Tout are in and outdoor air temperature respectively, in °C [default Tin = 20 °C];

V is volume of the dwelling, in m^3;

ah is specific heat capacity air: 0,33 $Wh/m^3.K$;

AV is ratio between dwelling outer surface (A) and volume (V) in m^{-1};

U is average insulation value of building shell in $W/m^2.K$;

` **qv** and qinf are ventilation- and infiltration rates per unit of dwelling volume in $m^3.h/ m^3$;

qrec is the fraction waste heat recovery from ventilation;

Correction factors, 1.15 is due to internal heat transfer in multi-zone dwelling (for multi-zone systems) and 1,2 is for reheat (setback correction for multi-zone systems). Both coefficients are in line with the EN 12831- 2003 standard.

Load profiles proposed are to be selected amongst the following design load (kW) at –10 °C: [1,04 1,55 2,33 3,5 5,3 8 12 18 27 40 60].

These tabulated values are for products enabling multi-zone temperature control and setback. A benefit is then computed when rating the performance of the unit to take this into account in Lot 1. Capacity required to declare a given profile is decreased for units without setback option (note: the product should be multi-zone to take benefit of the setback credit) - divided by 1.2. For single zone products that do not allow multi-zone control and corresponding heat load reduction, required design capacity is again divided here by 1.15.

The present profiles for heating systems in general, including heat pumps, are summarized in the table below. Typical heat pump capacity (EN14511 rated capacity) when sizing the heat pump to satisfy the design heating requirements have been added – the heat pump capacity loss at –10 °C here has been supposed to be 35 % of the rated heating capacity (EN14511).

	Pdesign (kW)			Sizing at Pdesign Corresponding HP capacity at 7 °C (P(-10)/P(7)=0,65)		
Profile size Lot 1	Single zone no reheat	Multizone no reheat	Multizone reheat	Single zone no reheat	Multizone no reheat	Multizone reheat
4XS	0,75	0,87	1,04	1,16	1,33	1,60
3XS	1,12	1,29	1,55	1,73	1,99	2,38
XXS	1,69	1,94	2,33	2,60	2,99	3,58
XS	2,5	2,9	3,5	3,9	4,5	5,4
S	3,8	4,4	5,3	5,9	6,8	8,2
M	5,8	6,7	8,0	8,9	10,3	12,3
L	8,7	10,0	12,0	13,4	15,4	18,5
XL	13,0	15,0	18,0	20,1	23,1	27,7
XXL	19,6	22,5	27,0	30,1	34,6	41,5
3XL	29,0	33,3	40,0	44,6	51,3	61,5
4XL	43,5	50,0	60,0	66,9	76,9	92,3

Table 16: Proposed default sizes for heating systems (VHK)

All profiles do not correspond to lot 10 products that would be typically be limited to sizes from 3XS to L for a rated heating capacity inferior to 12 kW. Typical rated heating capacities of products in lot

10 are: [2,2 2,5 2,8 3,5 4,2 5 6 7 10 12] but there are presently units of all sizes between 2 and 12 kW rated heating capacity.

This approach is similar to the one developed in the ARI 210/240 2006 standard. In this standard, Pdesign is noted DHR (demand heating requirement). "Design heating requirement (DHR) predicts the space heating load of a residence when subjected to outdoor design conditions." Reversible air conditioners US manufacturers can choose the DHR for which the heating seasonal performance factor of the heat pump is declared (within certain limits noticed DHRmin and DHRmax). Final values must be tabulated to default DHR values at design conditions that helps the end user to pick up the adequate unit (Table 17).

Standardized Design Heating Requirements ARI 210-240 2006	
Btu/h	**kW**
5000	1,5
10000	2,9
15000	4,4
20000	5,9
25000	7,3
30000	8,8
35000	10,3
40000	11,7
45000	13,2
50000	14,7
55000	16,1
60000	17,6
65000	19,0
70000	20,5
75000	22,0
80000	23,4
85000	24,9
90000	26,4
95000	27,8
100000	29,3

Table 17: ARI 210-240 2006 standardized demand heating requirement values

B.2.3) Testing: COP determination in A, B, C and D temperature and part load ratio conditions

The temperature conditions for determining the four part load COP values to be used in part B.2.4 are given in the following table :

	Part load	Outdoor air dry bulb (wet bulb) temperatures (°C)	Indoor air dry bulb temperature (°C)
A	$(T_j-16)/(Tdesign-16)$	-7(-8)	20
B	$(T_j-16)/(Tdesign-16)$	2(1)	20
C	$(T_j-16)/(Tdesign-16)$	7(6)	20
D	$(T_j-16)/(Tdesign-16)$	12(11)	20

Table 18 - Temperature conditions for SCOPon calculation of air-to-air units

Calculation procedure for fixed capacity units

For each part load ratio A, B, C and D, the COP is calculated as follows :

$$COP_{part\ load} = COP_{minload} * (load / (Cc * load + (1 - Cc))) * (1 - Cd*(1-load)) \quad \text{[Eq.2]}$$

Where
$COP_{part\ load}$: COP when the unit operates at the considered part load ratio
$COP_{min\ load}$: COP at the considered minimum part load temperature conditions when the unit is operating steady state continuously

Cd : degradation factor for pressure equalization when unit cycles off (default value 0.2, see below).
Cc : degradation factor for parasitics

load : ratio between Pdesign multiplied by the part load ratio, and the capacity when the unit is operating continuously (and in steady state) at the considered part load temperature conditions

In absence of measurement of the Cd value, Eq 2 should be used with a default Cd value of 0.2 that accounts only for thermodynamic losses when cycling. Cc is always measured as reported in part B.4.5.

In case of measurement, Cd and Cc should be computed separately (respectively degradation of average capacity, and average power increase).

All tests with a continuous and steady state operation of the unit shall be conducted according to EN 14511-3 procedure.

Calculation procedure for staged capacity control units

Determine the part load capacity and EER at each step of capacity control of the unit. If the steps do not allow to reach the required part load ratio within ± 3% (e.g. between 22% and 28% for a required part load ratio of 25%), determine the capacity and EER at the defined part load temperatures for the steps on either side of the control step of the unit. The part load capacity and the EER at the required part load ratio are then determined by interpolation between the results obtained from these two steps.

If the smallest control step of the unit is higher than the required part load ratio (D and/or C and/or B), the EER at the required part load ratio is calculated using Equation [Eq.2] as for fixed capacity units.

All the tests shall be conducted according to EN 14511-3 procedure.

Calculation procedure for continuous variable capacity control units

Perform the tests at the required part load ratios with the corresponding setting of the capacity control of the unit.
If the electronic control of the unit does not allow to obtain the required part load ratio, the calculation procedure given for staged capacity in 4.3.3 shall be applied.

If the smallest setting of the capacity control does not allow reaching one or several part load ratios, the EER at the required part load ratio(s) shall be calculated using [Eq.2] as for fixed capacity units in 4.3.2

All the tests shall be conducted according to EN 14511-3 procedure.

B.2.4) Heating Seasonal Performance Factor (HSPF) computation

The heating seasonal performance factor is computed using the following formula :

$$HSPF_{on} = \frac{\sum_{j=1}^{n} n_j BL(T_j)}{\sum_{j=1}^{n} n_j.((BL(T_j)-R_H(T_j))/COP(T_j)+R_H(T_j))} \quad \text{[Eq 1]}$$

Notations :
n_j : Frequency of hours of operation at the outdoor temperature of bin j
$BL(T_j)$: Building load ratio at the outdoor temperature of bin j
$R_H(T_j)$: Required additional resistive heating at the outdoor temperature of bin j
COP(Tj) : Coefficient of performance of the heat pump at the outdoor temperature of bin j

Building heat load ratio : $BL(T_j)$

$BL(T_j) = (T_j-16)/(Tdesign-16)*Pdesign$

Pdesign is the design heating requirement chosen by the manufacturer amongst the authorized profiles computed for the design outdoor temperature Tdesign.

Heating capacity of the heat pump

$P_H(T_j)$ is the capacity of the heat pump at the outdoor temperature of bin j.

- Whether $12 \leq Tj$
$P_H (T_j)=P_H (D)$

- Whether $7 < Tj < 12$
$P_H(T_j)= P_H (C)+(7- T_j)/(7-12)* (P_H (D)- P_H (C))$

- Whether $2 \leq Tj < 7$
$P_H(T_j)= P_H (B)+(2- T_j)/(2-7)* (P_H (C)- P_H (B))$

- Whether $-7 \leq Tj < 2$

Case 1: $T_j \geq Tmin$
$P_H(T_j)= P_H (A)+(-7- T_j)/(-7-2)* (P_H (B)- P_H (A))$

Case 2 : $T_j < Tmin$
$P_H(T_j)=0$

- Whether $-7 > Tj$
$P_H(T_j)=0$

Resistive heat output : $R_H(T_j)$

For each temperature T_j, the complementary heating is computed as follows :
$R_H(T_j)=max (0, BL(T_j)-P_H(T_j))$.

Electric power of the heat pump

$P_E(T_j)$ is the electric power absorbed by the heat pump at the outdoor temperature of bin j.

- Whether $12 \leq Tj$
$P_E (T_j)=P_E (D)$

- Whether $7 \leq Tj < 12$
$P_E(T_j)= P_E (C)+(7- T_j)/(7-12)* (P_E (D)- P_E (C))$

- Whether $2 \leq Tj < 7$
$P_E(T_j)= P_E (B)+(2- T_j)/(2-7)* (P_E (C)- P_E (B))$

- Whether $–7 \leq Tj < 2$

Case 1: $T_j \geq Tmin$
$P_E(T_j)= P_E (A)+(-7- T_j)/(-7-2)* (P_E (B)- P_E (A))$

Case 2 : $T_j < Tmin$
$P_E(T_j)=0$

- Whether $–7 > Tj$
$P_E(T_j)=0$

Coefficient of performance of the heat pump : $COP(T_j)$

- Whether $T_j < -7$
$COP (T_j)=1$

- Whether $–7 \leq Tj < 2$

Case 1: $T_j \geq Tmin$
$COP(T_j)= P_H(T_j)/P_E(T_j)$

Case 2 : $T_j < Tmin$
$COP(T_j)=1$
- Whether $2 \leq Tj < 12$
$COP(T_j)= P_H(T_j)/P_E(T_j)$

- Whether $12 < Tj$
The coefficient of performance of the heat pump is computed with the cycling formula defined in Eq 2 with the heating capacity and COP of measured point D as a reference for COPmin and load.

B.3) Seasonal Energy Efficiency Ratio in cooling mode

B.3.1) Climate conditions

B.3.1.1) 1 single seasonal performance index in cooling mode

To represent the average EU cooling conditions, it is possible to adopt an average indicator for the whole EU. A weighted average approach as used previously in lot 10 could be adopted. Methodology is proposed by EPEE. The bin method is also adopted. Binned hourly ASHRAE IWEC files for EU cities are kept to represent the EU27 climate. The weighting coefficients are based on an enlarged estimate of air conditioners sales in 2007 in Europe (EU25) by JRAIA.

	JRAIA Estimation Weight (%)
Austria	0.30
Belgium	2.04
Bulgaria	2.51
Cyprus	0.92
Czech	0.53
Denmark	1.05
Estonia	0.27
Finland	0.53
France	9.46
Germany	2.55
Greek	9.89
Hungary	1.40
Ireland	0.42
Italy	29.29
Latvia	0.45
Lithuania	0.66
Luxembourg	0.10
Malta	0.25
Netherlands	1.09
Poland	0.65
Portugal	3.11
Romania	2.25
Slovakia	1.14
Slovenia	0.42
Spain	23.01
Sweden	1.57
UK	4.16
Total	100

Table 20: Sales repartition of air conditioners in 2007, source JRAIA

Reference climate is the market weighted average of EU climates presented in the following table :
- Luxembourg sales are added to Belgium sales
- Baltic countries sales are gathered with the Lituanian climate as no file for Estonia and Latvia is available
- Malta climate is represented by the climate of the Italian Patelleria Island
- Italy's sales are divided half and half between Milan and Rome
- Spain's sales are divided half and half between Madrid and Barcelone
- Cities kept can be found in the table below.

Thus the final weighting coefficients for the different climates are:

Country	Climate	Weight
Austria	Vienna	0,3%
Belgium	Brussels	2,1%
Bulgaria	Sofia	2,5%

Cyprus	Larnaka	0,9%
Czech	Prague	0,5%
Germany	Berlin	2,5%
Denmark	Copenhagen	1,0%
Spain	Madrid	11,5%
Finland	Helsinki	0,5%
France	Lyon	9,5%
UK	London	4,2%
Greece	Athens	9,9%
Hungary	Debrecen	1,4%
Ireland	Dublin	0,4%
Italy	Milan	14,6%
Italy	Rome	14,6%
Lithuania	Kaunas	1,4%
Malta	ITA Patelleria	0,2%
Netherlands	Amsterdam	1,1%
Poland	Krakow	0,6%
Portugal	Evora	3,1%
Slovakia	Bratislava	1,1%
Sweden	Stockholm	1,6%
Romania	Bucharest	2,2%
Slovenia	Ljubljana	0,4%
Spain	Barcelone	11,5%
	TOTAL	100,0%

Table 21: Weighting of weather conditions with market sales

The binned hours by climate are reported below for a 5 months cooling period of 3672 hours. For cold climates, some hours being below 13 °C are to be discounted what explains that the total number of hours varies by climate.

Out Temp °C	Vienna	Brussels	Sofia	Larnaka	Prague	Berlin	Copenhagen	Madrid	Helsinki	Lyon	London	Athens	Debrecen	Dublin	Milan	Rome	Kaunas	ITA Patelleria	Amsterdam	Krakow	Evora	Bratislava	Stockholm	Bucharest	Ljubljana	Sevilla	Weighted ave
3	0	4	3	0	9	7	0	0	40	0	2	0	2	14	0	0	18	0	1	1	0	0	12	0	4	0	1
4	1	11	5	0	16	7	12	4	58	1	17	0	9	24	0	0	25	0	11	3	0	0	26	2	5	0	4
5	8	15	10	0	31	17	15	3	74	3	29	0	19	24	1	0	33	0	25	31	0	0	46	8	12	0	6
6	12	20	9	0	38	36	27	4	97	3	42	0	14	43	2	0	62	0	28	30	0	5	58	13	16	0	8
7	13	35	9	0	54	35	47	10	154	28	51	0	25	51	16	2	88	0	46	57	0	8	81	25	24	0	17
8	27	66	27	0	105	43	84	13	145	20	78	0	62	78	28	6	148	0	71	80	0	20	155	50	75	1	26
9	75	85	46	0	145	85	147	26	184	66	116	5	89	171	40	11	149	0	132	133	0	39	226	61	85	6	43
10	95	125	121	0	184	103	259	39	228	112	145	4	86	253	51	11	225	0	161	204	17	72	275	101	167	7	62
11	166	225	182	4	255	140	293	67	284	115	236	6	110	331	68	18	239	5	211	232	58	120	241	117	230	20	83
12	174	287	168	3	256	181	352	90	313	141	300	9	124	325	93	36	271	17	239	263	95	190	280	124	249	24	105
13	235	328	177	7	292	257	324	155	316	186	388	18	185	402	151	53	283	43	323	304	124	248	293	156	263	46	144
14	261	371	231	8	299	333	366	150	319	216	369	18	231	450	179	66	273	133	370	280	172	282	282	172	276	71	162
15	293	375	231	15	305	331	400	139	281	218	363	33	239	477	200	102	248	140	336	294	214	242	301	196	302	114	176
16	271	321	245	28	284	339	324	165	244	258	294	51	248	357	182	133	233	118	351	270	244	245	293	210	240	135	183
17	256	281	306	55	244	277	307	195	200	264	252	82	219	231	205	161	221	153	380	263	271	241	255	253	225	212	202
18	278	224	313	83	227	261	229	220	158	295	203	128	282	167	245	234	227	153	318	235	262	251	194	208	205	201	221
19	274	215	261	101	156	246	148	227	123	252	156	157	259	117	268	241	197	209	180	181	245	262	157	246	194	259	224
20	219	139	211	142	157	187	106	227	109	265	154	154	251	69	264	270	177	329	124	149	240	245	131	228	148	340	221
21	207	111	200	218	146	166	81	217	71	200	131	216	225	37	244	282	138	423	96	153	219	201	117	170	124	379	214
22	171	96	162	275	114	147	53	198	73	173	99	237	194	11	260	333	124	391	73	124	178	185	80	167	140	382	214
23	144	75	167	332	90	99	36	195	55	155	80	321	136	11	238	375	95	388	48	99	173	144	45	151	142	339	217
24	119	66	130	341	72	69	26	165	37	153	63	347	127	4	220	310	58	346	34	78	169	164	43	157	108	289	197
25	94	53	105	340	64	74	25	154	30	128	41	335	117	0	189	285	50	265	36	60	143	121	31	160	106	241	179
26	83	46	104	299	60	55	10	144	15	112	25	312	106	0	177	230	33	172	22	39	154	99	23	134	83	210	161
27	76	32	92	279	21	53	1	149	9	79	12	272	77	0	119	209	40	136	13	29	165	97	10	115	71	206	136
28	54	21	72	304	14	37	0	112	9	67	7	262	61	0	98	143	6	75	16	27	109	72	0	119	63	132	112
29	33	9	54	291	8	33	0	114	1	60	5	217	57	0	72	94	1	63	10	19	82	54	0	91	44	38	91
30	21	10	26	209	9	20	0	103	0	40	2	149	59	0	36	54	0	40	11	12	91	25	0	86	38	19	68
31	10	6	4	142	8	13	0	69	0	30	2	111	34	0	17	12	0	36	2	20	62	21	0	52	16	1	45
32	2	8	0	91	1	14	0	72	0	21	0	91	22	0	8	1	0	16	3	2	51	16	0	30	10	0	39
33	0	1	0	57	0	4	0	71	0	10	0	58	3	0	1	0	0	15	1	0	41	3	0	32	7	0	29

34	0	2	0	30	0	0	0	56	0	1	0	37	0	0	0	0	0	4	0	0	31	0	0	16	0	0	22
35	0	2	0	13	0	0	0	47	0	0	0	21	0	0	0	0	0	2	0	0	39	0	0	13	0	0	20
36	0	0	0	5	0	0	0	30	0	0	0	19	0	0	0	0	0	0	0	0	15	0	0	1	0	0	14
37	0	0	0	0	0	0	0	19	0	0	0	2	0	0	0	0	0	0	0	0	3	0	0	6	0	0	9
38	0	0	0	0	0	0	0	14	0	0	0	0	0	0	0	0	0	0	0	0	5	0	0	1	0	0	7
39	0	0	0	0	0	0	0	7	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	1	0	0	4
40	0	0	0	0	0	0	0	2	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	2
41	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	1
42	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	1
43	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
Total Hrs	3672	3665	3671	3672	3664	3669	3672	3672	3627	3672	3662	3672	3672	3647	3672	3672	3662	3672	3672	3672	3672	3672	3655	3672	3672	3678	3671

Table 21: Binned data of EU climatic files for summer period (5 months, May-October) and weighted average bin, source of climatic file ASHRAE IWEC data

B.3.1.2) 3 climates

With three climates kept in heating mode, the intend is to enable the design of heat pumps rather for Central and Northern conditions or rather for Central and Southern conditions. Most air conditioning being reversible, it makes sense to adopt the same type of representation with 3 climates.

For a matter of coherence, the same climates than in the heating mode could be kept. Nevertheless, this representation would lead to 2 climates over 3, Strasbourg and Helsinki, with little hours above 28 °C.

Using the bin methodology described in B.3.2 with variable design temperatures and seasons, the cumulated cooling energy versus outdoor temperature of several European climates are represented in the figure hereafter. This representation gives an idea of the outdoor temperature impact on the average operating conditions of air conditioners.

Presently, Strasbourg is changed for Milan (main air conditioning market in Europe is Italy) for the summer that gives a hotter intermediate temperature profile. In addition, Italy cooling requirements computed in Task 4 are very close to the average EU cooling requirements.

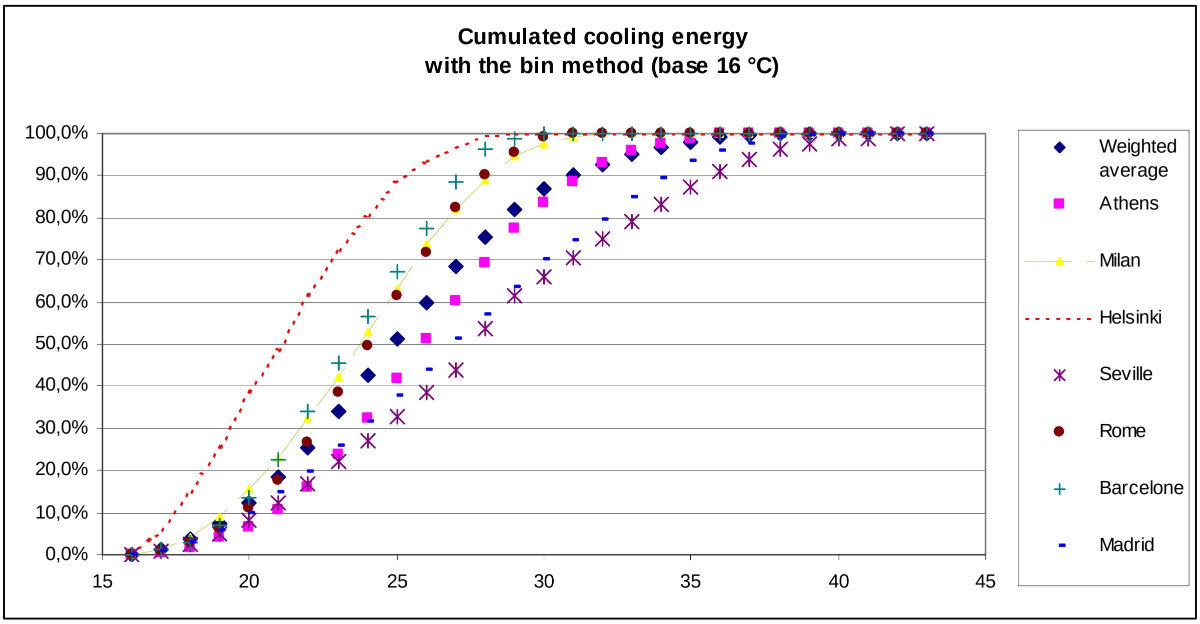


Figure 21: climatic comparison of different climates using the cumulated cooling requirements

In case a single average EU indicator as a EU city, this could be Athens since it is the closest to the weighted average ; however, whether the temperature profile is alike, the energy consumption is in average much higher than for average EU.

On this representation it appears clearly that hottest climates are Seville and Madrid, then comes Athens, while Italy and sea climates that exhibit lower average outdoor operating conditions. The reason for the EU average to be close to Athens is the fact that Spain concentrates important share of the sales and that cities picked up for Spain have very hot climates. Because of their very hot climates, Madrid or Seville should be kept to represent warm weathers. However, even for Spain, they are the hottest locations. Other regions of Spain have milder summers. Keeping Madrid or Seville with a 40 °C design temperature to represent hot climates could be misleading for all other Southern countries including Italy, Greece, South of France and a large part of Spain.

The cooling season is 5 months in Milan, 6 months in Athens and 3 months in Helsinki. The bin hours of summer conditions in Athens, Strasbourg and Helsinki are supplied in table 19. As for winter conditions, the bin hours are hours equal to the bin temperature +/- 0.5 K). The design temperatures are 35 °C for Athens, 31 °C for Milan and 27 °C for Helsinki.

		Athens	Milan	Helsinki
Bin	Outdoor temperature °C	6 months	5 months	3 months
1	17	166	205	184
2	18	216	245	144
3	19	245	268	112
4	20	218	264	102
5	21	278	244	63
6	22	317	260	69
7	23	370	238	49
8	24	376	220	32
9	25	351	189	30
10	26	327	177	15
11	27	276	119	9
12	28	263	98	9
13	29	217	72	1
14	30	149	36	0
15	31	111	17	0
16	32	91	8	0
17	33	58	1	0
18	34	37	0	0
19	35	21	0	0
20	36	19	0	0
21	37	2	0	0

Table 19: Binned summer data of selected representative climates of Europe for heating conditions (source ASHRAE IWEC data file)

B.3.2) Building cooling requirements

Building load curves in cooling mode have been computed from complete building simulations in task 4. It results that the 0 % load point can vary from 12 °C to 20 °C depending on the building type, climate and country (different National Thermal Regulations).
With 16 °C for the zero load in heating mode, similar or higher value should be kept in cooling mode and consequently 16 °C is kept also in cooling mode for the bin method. It can be noticed that in the ARI 210/240 standard, the zero load temperature is 18.3 °C in cooling and in heating mode.

Whether Tj≤Tdesign,
$BL(T_j) = (T_j-16)/(Tdesign-16)*Pdesign$

Whether Tj>Tdesign,
$BL(T_j) =1$

Where
- $1 \leq j \leq 21$
-Pdesign is defined as the maximum capacity of the heat pump at the temperature Tdesign for a specific location.

-Design temperatures are 31 °C for Milan, 35 °C for Athens, 27 °C for Helsinki and 35 °C in the case of the weighted average EU climate.

B.3.3) Testing: EER determination in A, B, C and D temperature and part load ratio conditions

The temperature conditions for determining the four part load COP values to be used in part B.3.4 are given in the following table :

	Part load	Outdoor air dry bulb (wet bulb) temperatures (°C)	Indoor air dry bulb (wet bulb) temperature (°C)
A	$(T_j-16)/(Tdesign-16)$	35	27 (19)
B	$(T_j-16)/(Tdesign-16)$	30	27 (19)
C	$(T_j-16)/(Tdesign-16)$	25	27 (19)
D	$(T_j-16)/(Tdesign-16)$	20	27 (19)

Table 22 - Temperature conditions for SCOPon calculation of air-to-air units

For Helsinki, 100 % is reached below 30 °C. Two tests at full capacity for an outdoor temperature of 25 and 30 °C should be led in order to determine the 27 °C capacity by linear interpolation of capacities and power as presented in B.3.4.

For Milan, the same procedure would result in 5 tests to measure the seasonal performance.

EER calculation procedures for A, B, C and D points are similar to the ones proposed in part B.2.4.

B.3.4) Seasonal Energy Efficiency Ratio in cooling mode

$$SEER_{on} = \frac{\sum_{j=1}^{n} n_j BL(T_j)}{\sum_{j=1}^{n} n_j.(BL(T_j)/EER(T_j))} \qquad \text{[Eq 3]}$$

<u>Energy Efficiency Ratio: $EER(T_j)$</u>

$EER(T_j)= P_C(T_j)/ P_E(T_j)$

<u>Cooling capacity : $P_C(T_j)$</u>

Cooling capacity is interpolated linearly as in the heating mode for bin temperatures between measurements A, B, C and D.

Above 35 °C, the cooling capacity remains constant and its value equals the cooling capacity measured in the A test conditions.

Below 20 °C, the cooling capacity remains constant and its value equals the cooling capacity measured in the D test conditions.

<u>Electric power : $P_E(T_j)$</u>

The electric power absorbed is interpolated linearly as in the heating mode for bin temperatures between measurements A, B, C and D.

Above 35 °C, the electric power absorbed remains constant and its value equals the electric power absorbed measured in the A test conditions.

Below 20 °C, the electric power absorbed remains constant and its value equals the electric power absorbed measured in the D test conditions.

B.3.5) Case of single duct

With an average temperature operating condition close to the set point, application condition 27 (19) is closer to the average operating condition while the rating condition 35 (24) enables to determine the capacity available in the design conditions. Average performance should be computed simply by determining the efficiency with a load of 50 % (rounding of 55 %) in the 27 (19) application condition.

B.4) Hours and parasitic consumptions

B.4.1) Summer period

The seasonal energy efficiency ratio including parasitics is determined as follows :

$$SEER = \frac{H_{CE}.Pdesign}{\frac{H_{CE}.Pdesign}{SEER_{on}} + H_{CTO} \times P_{CTO} + H_{SB} \times P_{SB} + H_{CCK} \times P_{CK}}$$

Hours that can be used for the different locations are in the table below:

HOURS	Athens	Milan	Helsinki	Weighted average EU
Season Hours	4416	3672	2184	3672
Hours Cooling (24h/day)	4108	2661	819	3142
Cooling Thermo-off, H_{CTO} (24h/day)	308	1011	1365	530
Hours Cooling, H_C (10h/day)	1712	1109	341	1309
Cooling Thermo-off, H_{CTO} (10h/day)	128	421	569	221
Cooling Standby, H_{SBC}	2576	2142	1274	2142
Crankcase heater, cooling H_{CKC} (=H_{SBC}+H_{CTO})	2704	2563	1843	2672
Total Hours (Crankcase discounted)	4416	3672	2184	3672
COOLING NEEDS	**Athens**	**Milan**	**Helsinki**	**Weighted average EU**
Equivalent Hours Cooling (24h/day)	1800	1035	265	954
Equivalent Hours Cooling (10h/day)	750	431	110	398

Table 23: Hours to compute SEER for the different climates

In average, occupation represents only 10 hours per day according to the different scenarios in task 4 which is kept as the reference. Operating and thermostat off hours are discounted accordingly.

Because the bin method only considers the part of cooling requirements that are sensitive to outdoor ambient and that for the sake of simplicity the same base temperature has been kept for the 3 climates, it very unlikely that equivalent hours computed directly from climatic data give a good estimate of real consumption of the units in the different climates. Results overestimate Athens cooling requirements while Helsinki values are underestimated (internal gains and strong insulation create cooling load until low outdoor temperature).

Consequently, the results priorly identified in Task 4 for unitary cooling needs and sizing are used to estimate coherent numbers of equivalent hours.
- Average cooling requirements for Athens are around 60 kWh/m2 and unitary size about 130 W/m2, thus equivalent hours are of 500 hours.
- Average cooling requirements for average EU are around 40 kWh/m2 and unitary size about 120 W/m2, thus equivalent hours are of 350 hours.
- Average cooling requirements for Helsinki are around 25 kWh/m2 and unitary size about 100 W/m2, thus equivalent hours are of 250 hours.

With this method, the Milan climate can be used to compute average EU consumption for cooling and the relative weight of parasitics and of the cooling consumption is respected in the different climates.

B.4.2) Winter period

To compute the global seasonal factor for the heating season, parasitic consumption is to be taken into account as follows.

$$HSPF = \frac{\mathrm{H}_{HE}.Pdesign}{\dfrac{\mathrm{H}_{HE}.Pdesign}{\mathrm{HSPF}} + \mathrm{H}_{\mathrm{HTO}} \times \mathrm{P}_{\mathrm{HTO}} + \mathrm{H}_{H\mathrm{CK}} \times \mathrm{P}_{\mathrm{CK}}}$$

Hours that can be used for the different locations are in the table below:

HOURS	Athens	Strasbourg	Helsinki
Season Hours	4344	5088	6576
Hours heating (24h/day)	3589	4909	6445
Heating Thermo-off, H_{HTO}	755	179	131
Crankcase heater, heating H_{HKC} (=H_{HTO})	755	179	131
Total Hours (Crankcase discounted)	4344	5088	6576
Heating NEEDS	**Athens**	**Strasbourg**	**Helsinki**
Season Hours heating	3589	4909	6445
Equivalent Hours heating (*), H_{HE} (rounded) multi zone & setback	1000	1000	1600
Equivalent Hours heating (*), H_{HE} (rounded) multi zone no setback	1200	1200	1900
Equivalent Hours heating (*), H_{HE} (rounded) single zone	1400	1400	2100

Table 24: Hours to compute HSPF

Equivalent hours are the results of a more complete model developed in Lot 1. It is the ratio of the unitary consumption of the average dwelling in kWh/m2 over the design heating requirements (at Tdesign).
As explained before, correction coefficients of 1.15 and 1.2 correspond to multizone and setback requirements.

- Average heating requirements for Athens are of 71 kWh/m2 and design requirement of 71 W/m2, thus equivalent hours at design conditions are of 1000 hours. The default value kept for heat pump in this lot is 1400 hours.
- Average heating requirements for Strasbourg are of 87 kWh/m2 and design requirement about 87 W/m2, thus equivalent hours are of 1000 hours. The default value kept for heat pump in this lot is 1400 hours.
- Average heating requirements for Helsinki are of 148 kWh/m2 and design requirement about 96 W/m2, thus equivalent hours are of 1600 hours. The default value kept for heat pump in this lot is 2100 hours.

B.4.3) Case of cooling only and heating only units

For cooling only and heating only units, the units is not cooling (or heating) during part of the year. In that case, the entire season is counted as off mode with subsequent energy consumption. In addition, these hours correspond also to hours of operation of the crankcase heater that continues to operate if it is not controlled.

For cooling only units (resp. and heating) only units, the SEER (resp. HSPF) formula is completed with off mode hours.

$$SEER = \frac{H_{CE}.Pdesign}{\frac{H_{CE}.Pdesign}{SEER_{on}} + H_{CTO} \times P_{CTO} + H_{SB} \times P_{SB} + H_{CCK} \times P_{CK} + H_{OFF} \times P_{OFF}}$$

$$HSPF = \frac{H_{HE}.P_{H}(A)}{\frac{H_{HE}.P_{H}(A)}{HSPF} + H_{HTO} \times P_{HTO} + H_{HCK} \times P_{CK} + H_{OFF} \times P_{OFF}}$$

B.4.4) Test of average power of parasitic modes

Thermostat off is tested after D test in cooling mode and after C test in heating mode. (30 mn)

Standby is tested right after thermostat off test in A test conditions in cooling (or the C test condition in heating mode for heating only units) after unit is switched off by remote. (30 mn)

Off mode is tested at 20 °C in cooling mode (12 °C in heating mode for heating only) in cooling mode after unit is switched off manually. (30 mn)

Crankcase heater hours of operation have been defined in the previous tables. The rationale behind these hours is explained below.

Case 1 : Whether crankcase is not controlled, its consumption will be accounted for in thermostat off, standby and off mode and also during cooling and heating hours. However, in cooling and in heating active modes, its power will already be accounted for so that active mode hours should not be accounted.

Case 2 : Whether crankcase is controlled by simple compressor contact, its consumption will be accounted for in thermostat off, standby and off mode. So the number of hours of operation outside hours of heating and cooling is the same as for the previous control and hours are already accounted for.

Case 3 : Whether crankcase is controlled by simple compressor contact and by temperature difference between compressor oil and condenser side, there would be no crankcase heater consumption accounted for in thermostat off, standby and off mode (if making sure standby, thermostat off and off mode are measured right after compressor stop for less than a few hours) and a dedicated test should be done. In that case, there may be a second control to cut crankcase over a threshold temperature: this is indicated by Japanese manufacturers for rotary compressors.

Thus, computing the hours for crankcase heater is equivalent to count hours with no heating and / or cooling periods. A specific crankcase heater test is meaningful and will allow to distinguish units with a smart control. These units will not be penalized by the large number of potential hours of the oil heater because of very low consumption in this test.

The test can be done as suggested by EPEE below.

"The energy consumption of the unit shall be measured after the compressor reached stable temperature for the "A" temperature conditions test in heating mode and stopped with the control device. Ambient temperature of compressor unit shall be maintained at 2 ± 2 K for at least 8 hours. During this 8 hours period from stop of the compressor, the power input for crankcase heater shall be measured and averaged. The standby power consumption is deducted from this measured total energy consumption of the unit to determine the crankcase heater power."

In addition, whether there is an outdoor temperature control with a declared Tck value lower than 16 °C, it can be checked as follows.

After this 2 °C test for crankcase, room temperature is raised to T_{CK}. 1 hour after the room temperature has been stabilized, crankcase heater power is measured over 30 minutes and this gives the power consumption to be used P_{CK} in the computation of crankcase energy consumption in cooling mode.

B.4.5) Default coefficient to translate cycling losses

In the ARI 210/240 standard, a 0.25 value for cycling degradation is used as a default value.
This coefficient gathers the two types of performance degradation foreseen in the PrEN14825 standard : thermodynamic cycling, losses with coefficient Cd, and parasitic losses with coefficient Cc. Test measurement is done at 20 % part load ratio. Degradation formulas are shown in Eq 2 reported below :
$COP_{part\ load} = COP_{min\ load} *(load / (Cc * load + (1 – Cc))) * (1 –Cd*(1-load))$

Cd equals 0.25 means that degradation of EER or COP is 20 % at 20 % load.

For the Lot 10 3.5 kW base case ON-OFF unit, there is about 36 W consumed by the unit when cycling and the compressor is stopped. For 3.5 kW cooling capacity and 4 kW heating capacity with EER 3.1 and COP 3.4, electric power would be 1.13 in cooling mode and 1.18 in heating mode.
Cc coefficients in cooling and in heating mode would be :
1-36/1130= 97 % in cooling mode
1-36/1180= 97 % in heating mode

In order to get an equivalent US Cd coefficient of 0.25, it means 0.2 should be used in the PrEN14825 formula.

This value should be kept as reference for Cd value for air to air units. This would be particularly relevant for units with high and uncontrolled crankcase heater power for which a simple Cd coefficient would underestimate efficiency degradation at lower loads.

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