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Lot 6: Air-conditioning and ventilation systems

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Air conditioning systems
Draft report of Task 5

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Contents

5. TASK 5 – TECHNICAL ANALYSIS BAT AND BNAT

INTRODUCTION

The task 5 description is given in the methodology and reported below.

SCOPE*: Best Available Technology (BAT) entails a technical analysis not of the current products on the market but on currently available technology, expected to be introduced at product level in the shorter term. Best Not yet Available Technologies (BNAT) summarise the state-of-the-art in research and development for a product, indicating market possibilities in the longer term. The environmental performance of BAT and BNAT both provide part of the input for the identification of the improvement potential (task 6).*

Subtask 5.1 – Definition of BAT
BAT should be defined for the products defined in subtask 1.1:
- *"Best" shall mean most effective in achieving a high level of environmental performance of the product;*
- *"Available" technology shall mean that developed on a scale which allows implementation for the relevant product, under economically and technically viable conditions, taking into consideration the costs and benefits, whether or not the technology is used or produced inside the Member States in question or the EU-27, as long as they are reasonably accessible to the product manufacturer;*
- *Barriers for take-up of BAT should be assessed, such as cost factors or availability outside Europe.*

Subtask 5.2 – Definition of BNAT
BNAT should be defined for the products defined in subtask 1.1:
- *"Not yet" available technology shall mean that not developed yet on a scale which allows implementation for the relevant product but that is subject to research and development;*
- *Barriers for BNAT should be assessed, such as cost factors or research and development outside Europe.*

This task 5 report is organized by main products in the scope of the study, cooling generators, terminal units and heat rejection units. For each product group, the BAT and BNAT are identified.

For all three product groups, the improvement potential achievable through the Ecodesign Directive is discussed as well as the potential efficiency metrics to reach it. The focus is on the main environemental impacts identified in Task 4, meaning energy consumption and refrigerant direct emissions for air conditioners and chillers, and on energy consumption for terminal units and heat rejection units.

NB: Part of the technical description already shown in the task 5 report of ENTR lot 6 on ventilation is also useful for air conditioning systems. To ease the reading of the report, useful information is past here rather than quoted and completed with information regarding conditioning products. This regards information on motor and fan efficiencies. The report also bases upon the Lot 10 study reports regarding air conditioner improvements.

5.1. IMPROVEMENT POTENTIAL UNDER THE ECODESIGN DIRECTIVE

5.1.1 SYSTEMS, PRODUCTS AND EXTENDED PRODUCTS

The level playing field of the Ecodesign Directive is limited to measures on products that manufacturers or their representative should satisfy to apply the CE marking. The gap between products and systems is very important for energy efficiency and is mainly addressed in the Directive on the Energy Performance of Buildings 2010/31/EU.

However, following the MEEuP, products should be evaluated here on the basis of the function they provide and not upon their technology. This is a complex issue for cooling/heating generators since some products can supply the function independently and others require additional components to do so.

The approach adopted in the latest working document on boilers, the “extended product approach”, can help.

For cooling products, it would enable for instance to compare air cooled and water cooled air conditioners on a single scale, using:

- Default values for the energy consumption of the pump and of the heat rejection unit of the water cooled air conditioner;
- Standard sets of operating inlet temperatures for the inlet temperatures of the air condenser and of the water cooled condenser.

The extended product approach thus appears as a means to ensure fair requirements amongst different product types, here illustrated for air cooled and water products. Note this does not imply forcedly to change the metrics in force. It can also simply be used to compare and adjust the requirements of the different products.

5.1.2. MULTI-FUNCTION PRODUCTS

Products supplying cooling or heating

Each function is evaluated independently. BAT and BNAT are looked at by function. Potential trade-off designs between heating and cooling modes are investigated.

Products supplying simultaneous cooling and heating (or hot water)

With water based systems and with variable refrigerant flow systems, it is possible to simultaneously heat and cool different parts of the same building. In both cases, free heat, normally released to the ambient, can be recovered for simultaneous heating application and in some case with temperature levels compatible with sanitary hot water production.

Different solutions exist for water based systems:
a) the addition of a heat reclaim condenser in parallel to the chiller or air conditioner air/water-to-refrigerant condenser to recover heat for space heating, sanitary hot water or other heating use,
b) a water to water cooling generator, the cooling side supplying a chilled water network and the heating side supplying a hot water network; if the heating energy supplied by the heat pump is in excess, it has to be released; more efficient installations make use of a geothermal installation to store this heat in the ground; the reverse situation may also occur in winter time and in that case, the excess cooling can be compensated by extracting heat from the ground.
c) a central water loop, controlled in temperature thanks to a cooling tower and a boiler generally, and water-to-air cooling/heating generators ensuring the room temperature treatment; this is called a water loop heat pump system.

In case b) and c), the system design enables the simultaneous heating and cooling. This is thus not a product improvement but a system one.

In the case a) of a chiller with a heat reclaim condenser, this is clearly an improved product design. It can provide free heat to a boiler system or to a water heater. This type of interaction can be taken into account in the Energy Performance of Buildings Directive. Under the Ecodesign Directive, it could also be included, via in the calculation method of ENER Lot 2 in a single "extended product" with a gas backup or another generator to make a complete hot water solution. The product would then have two labels, one for cooling and one for the hot water production. However, it is not possible via the evaluation of the cooling function alone, to take into account this potential improvement.

With refrigerant flow systems, some of the indoor units may operate in cooling mode while other operate in heating mode. The system enables to recover the heat extracted in the cooled area to supply part or totality of the heating needs, depending on the coincident heating and cooling loads.

The findings in the air conditioning task 3 of this study suggest however that simultaneous heating and cooling requirements is not a general requirement for all buildings but more likely limited to specific designs with contrasted zones and conditions of operation.

Thus it is not intended to include a potential credit on heating or cooling indexes for the VRF systems having this capability in a standardized metrics used for cooling or for heating, but rather to let that potential improvement in the field of the EPBD. Under the Ecodesign Directive, it is possible to require more information from manufacturers in order for designers to exploit the potential energy savings associated with this solution for specific applications. It can also be noticed that in the US ASHRAE 90.1 2010 standard[1], there is an allowance on the MEPS requirements for VRF systems with a heat recovery function (which may imply supplementary head losses when operating in cooling or in heating mode).

Products supplying cooling/heating and ventilation

Some AHUs may be fitted with an integrated cooling/heating generator, which can be predesigned for a given AHU's airflow (as for rooftop air conditioners) or sized according to the customer requirements. This is the case of rooftop air conditioners. These cooling/heating generators should be judged regarding their cooling/heating function here, in addition of their ventilation function already considered in the ventilation part of ENTR Lot 6. The cooling/ventilation positive interactions are discussed hereafter.

5.1.3. ELECTRICITY GRID SUMMER/WINTER PEAK CONTRIBUTION

The MEErP bases upon the least life cycle cost approach in order to fix the product minimum performance requirements. Energy costs are balanced with investments costs. For cooling/heating generators, the annual cooling/heating requirements being fixed, the energy efficiency is the main parameter. Regarding cooling/heating generators, the standards in Europe are moving from full load to part load index.

This move should lead to a better characterization of the energy consumption of the products and foster the development of important improvements regarding the energy consumption, as variable speed drive compressors.

Nevertheless, in economies that are summer peaking as California, minimum SEER for air conditioners are completed with minimum full load EERs, in order to limit the impact of the air conditioners on the grid. Several economies have requirements at full load and at part load for air conditioners (in cooling and in heating mode) and for chillers.

The MEErP methodology does not give a method to limit the efficiency requirements at full load in design conditions, although electric cooling/heating products clearly add an important burden to the grid: cooling contributes to the summer peak in southern countries (as Spain or Italy) and electric heating to the winter peak in central and northern Europe. At the end, the electricity costs are likely to increase for the customers because of the increased peak. Several methods can be used to evaluate

[1] Please refer to the Task 1 report

the impact of this externality: using the structure of the electricity tariffs, which generally have both a kWh and a kW cost as in the EECCAC project or by estimating the price increase of electricity due to higher loads, as is made on an hourly basis in California (Price et al., 2011).

Giving a value to this externality would also contribute to the development of alternative technologies including renewable cooling and storage, to show the importance of the refrigerant choice, not only for its TEWI[2] emissions but also for its peak efficiency.

It is possible to derive orders of magnitude for added cooling and heating capacity for countries in which the added capacity increases the electricity demand at peak time. For each added kW cooling in summer peaking countries, say with a peak EER cooling of 3, it is necessary to add 1/3 kW electric power generation. The cost incurred varies with the production means used, from 300 €/kW for a fuel turbine (DGEMP, 2003) and 700 €/kW (Kaplan, 2008) for a combined cycle gas turbine. So for each kW cooling added to the peak, there is an overcost from 100 €/kWcooling to 230 €/kWcooling. This investment cost may be doubled by transmission and distribution costs (Price et al., 2011). Note that this externality order of magnitude is comparable to the manufacturer selling prices per kW cooling. It should also be added that the situation is worst in heating mode as the peak performance at low ambient, say -7 °C for instance, is rather 2 than 3.

This shows the need to look for BAT and BNAT not only to reduce the energy consumption but also to increase the full load efficiency.

[2] TEWI : total equivalent warming impact, defined in the air conditioning report Task 1.

5.2. ELECTRICALLY DRIVEN MECHANICAL VAPOR COMPRESSION CYCLES IMPROVEMENT

It has been shown in the task 4 report that the main environmental impact of electrically driven mechanical vapor compression cycles improvement was mainly due to their electric consumption and to the direct emissions of refrigerant fluids along their lifetime. Energy efficiency and refrigerant direct emissions are thus the main parameters to reduce the environemental impact of these products.

5.2.1. REFRIGERANT FLUID

Refrigerant fluid choice

Main fluids in use today are HFCs R134a, R410A and R407C. Ammonia chillers have been commonly used for refrigeration and some air conditioning chillers are available in Europe[3]. Before the restriction on ozone depleting substances, the HCFC R22 was the main refrigerant for all applications. It was replaced with R134a, R407C and R410A. As a near drop-in of R22, R407C was first preferred but soon the superior volumetric capacity of R410A and better transport properties led R410A to dominate the air conditioner segment, the scroll chiller segment, and now chiller manufacturers are also developing screw chillers with R410A instead of R134a, still the dominant refrigerant for screw and centrifugal compressor chillers.

Along to dominant HFC, prototypes and even commercial solutions with natural refrigerants have been developed. CO_2 is available for refrigeration and for high temperature heat pump applications (as heat pump water heaters in ENER Lot 2). Daikin announced a VRF model optimized for CO2 in a design optimized for heating which is now available for sales. The performance is lower than with R410A in both cooling and heating mode at full load. Performances at part load are not known.

Several prototypes of air conditioners and chillers (some heat pumps too) have been developed with propane with performances equivalent or even better than R410A. Gree is selling propane based air conditioners, whose installation is presently restricted by the safety standard EN378[4] in Europe because of its flammability.

In the latest years, the perspective of the entering into force of the ban on HFC refrigerants with a GWP higher than 150 (European regulation 2006/842 on fluorinated gases) led the chemical industry to look for alternatives to the R134a, the main fluid presently in use for car air conditioning. One main fluid is foreseen at the moment, with GWP equal to 4, the R-1234yf – near drop-in of R134a, which is planned to be adopted by the automotive car industry in the USA and in Europe despite its being slightly flammable.
R32 is a flammable refrigerant with a lower GWP than R410A (650 versus 1875) and with higher performances, both for the cycle efficiency and for its transport properties (Brown, 2009) and thus makes an excellent candidates for R410A replacement.

Ammonia is a refrigerant with high cycle performance and refrigerant transport properties but it requires a complete redesign of the circuits, to face high temperature at the compressor outlet (which problem can be solved by mixing it with ethane which gives an azeotrope), to replace copper and copper alloys by aluminum and which additionally is toxic and thus of limited application following the safety requirements of EN378.

Water may be envisaged although the costs are likely to be prohibitive and the 0 °C freezing temperature a problem both in cooling and in heating mode.

The properties of the present refrigerants and potential candidates are summarized in the table below.

[3] See www.sabroe.com for instance.
[4] See Task 1 for more details.

Refrigerant number	Chemical name	Chemical formula	Safety group EN378	PED fluid group	Practical limit (kg/m3)	ATEL/ODL(kg/m3)	Flammability LFL (kg/m3)	Vapour density 25 °C. 101.3 kPa (kg/m3)	Molecular mass	Normal boiling point (°C)	ODP	GWP (100 yr)	Auto-ignition tempe-rature (°C)
22	Chlorodifluoromethane	CHCIF2	A1	2	0.3	0.3	—	3.587	86.5	– 40.8	0.055	1700	635
134a	1.1.1.2-tetrafluoroethane	CH2FCF3	A1	2	0.25	0.25	—	4.258	102.0	– 26.2	0	1 300	743
32	Difluoromethane (methylene fluoride)	CH2F2	A2	1	0.061	0.085	0.306	2.153	52.0	– 51.7	0	650	648
290	Propane	CH3CH2CH3	A3	1	0.008	0.09	0.038	1.832	44.0	– 42	0	3	470
717	Ammonia	NH3	B2	1	0.00035	0.00035	0.104	0.704	17.0	– 33	0	0	630
744	Carbon dioxide	CO2	A1	2	0.1	0.036	—	1.808	44.0	– 78 c	0	1	—
1234yf	2,3,3,3-tetrafluoro-1-propene	CF3CF=CH2	A2	1	0,06	0,467	0,299	4,766	114	– 29,4	0	4	405
Refrigerant number	**Composition**	**Comp. Tolerance**	**Safety group EN378**	**PED fluid group**	**Practical limit e (kg/m3)**	**ATEL/ODL(kg/m3)**	**Flammability LFL k (kg/m3)**	**Vapour density 25 °C. 101.3 kPa a (kg/m3)**	**Molecular mass a**	**Normal boiling point a (°C)**	**ODP a f**	**GWP (100 yr)**	**Auto-ignition tempe-rature (°C)**
410A	R-32/125 (50/50)	+ 0,5 – 1,5/+ 1,5 – 0,5	A1	2	0,44	0,387	n/a	3,007	72,6	– 51,6 to – 51,5	0	1980	N.D.
407C	R-32/125/134a (23/25/52)	± 2/± 2/± 2	A1	2	0,31	0,268	n/a	3,582	86,2	– 43,8 to – 36,7	0	1650	704

Table 5 - 1 . Properties of refrigerant fluids used for air conditioning and potential candidates for replacement, source EN378-1:2010.

Nota bene

ATEL Acute Toxicity Exposure Limit

ODL Oxygen Deprivation Limit

PED Pressure Directive Equipment (97/23/EC) fluid group, to classify the fluids and qualify the refrigerant equipment components

To summarize the Table 5 - 1, 0 ODP lower than traditional HFCs refrigerants are either flammable (hydrocarbons, HFOs, R32), toxic (ammonia) or of low performance for air conditioning application (CO2). It appears that there is not one perfect refrigerant with a GWP lower than the one of traditional HFCs for stationary air conditioning applications.

Alternative refrigerants still have to be evaluated on a TEWI basis and in the life cycle cost analysis in Task 6 of this report. To do so, in addition of potential overcosts induced by safety measures related to toxicity and flammability, it is necessary to take into account their thermodynamic and transport performances. (Brown, 2009)[5] compared several refrigerant fluid amongst the ones of interest here, including fluids of the propene isomer family, to which the R-1234yf belongs.

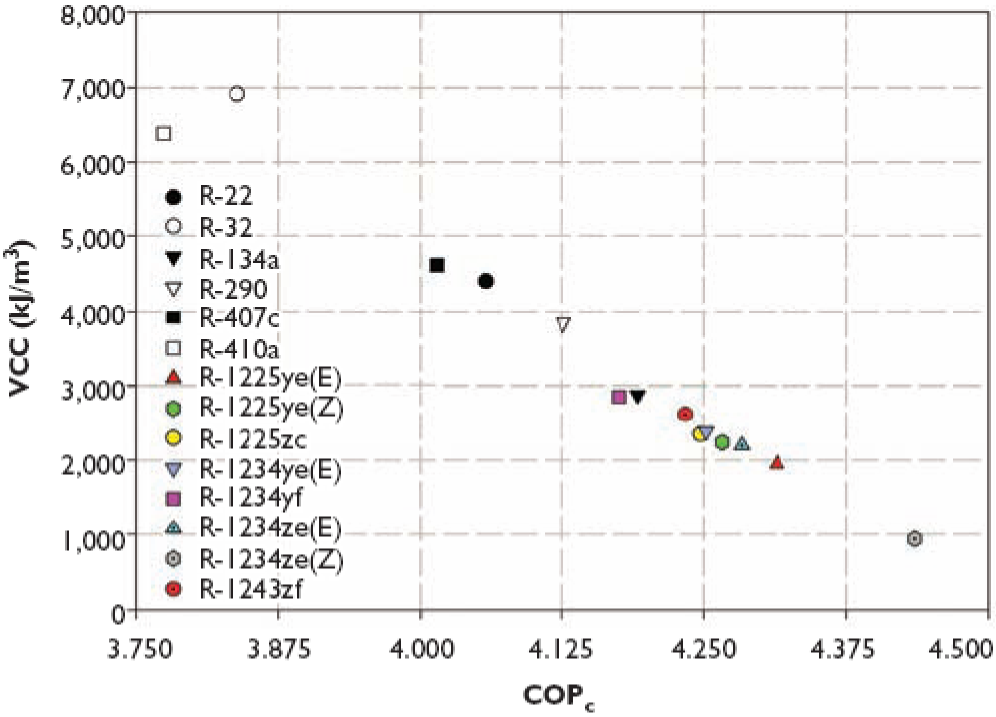


Figure 5 - 1 .Volumetric cooling capacity (VCC) versus COPC for unitary A/C application, source (Brown, 2009)

In the figure 5 – 1, the COP is shown for typical standard A/C application (close to the ISO 5151 T1 condition[6]). It is drawn versus the volumetric cooling capacity, which gives an indication on the compacity of the system. This first graph shows that these isomers have cycle performances comparable or better than present HFCs. Regarding R134a, the R1234yf exhibits similar performances. R410A and R32 exhibit lower COP values but have much higher compacities, twice the one of 1234yf and R134a, and 1.5 times the one of propane.

On Figure 5 – 2 below, two other essential parameters are represented:

- the penalty factor PF: “it is a concept for comparing the relative heat transfer performance potentials—at least for condensation—of refrigerants. In particular, the PF is a product of two local energy penalization terms, both of which negatively affect the amount of required compressor work: the first penalization term is associated with the frictional pressure drop of the condensing refrigerant, and the second penalization term is associated with the driving temperature difference for the heat transfer process.”
- the heat transfer coefficient, which will condition the size of the heat exchangers at equal temperature difference.

[5] Brown’s article was published before detailed test data were available for most propene isomers and values with exact fluid properties may slightly differ form the thermodynamic performance models used here.

[6] Outdoor air 35 °C / Inlet indoor air 27 °C dry bulb and 19 °C wet bulb

This figure shows that in addition of offering an excellent compacity, R410A and R32 refrigerants also offer superior transport and heat transfer properties as compared to most concurrents, except ammonia. The combination of their relatively lower cycle performance but higher compacity and transport properties is likely to result in a more compact design to reach the same performance levels as with other alternatives. Note that R1234yf appears to have comparable performances as R134a, even if slightly lower.

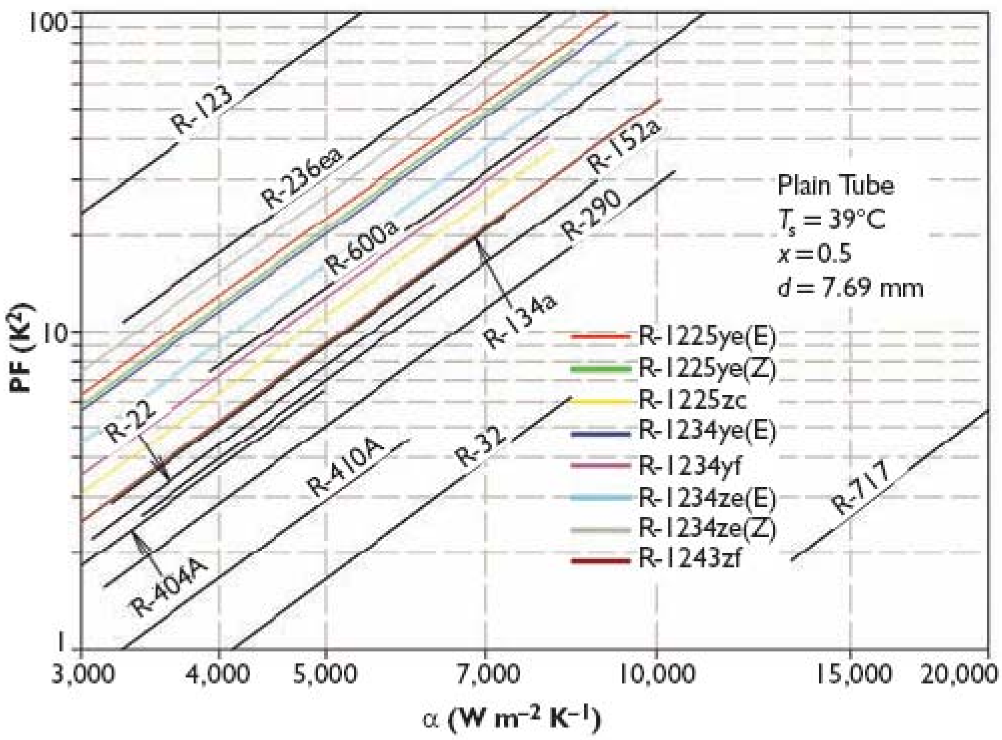


Figure 5 - 2 .Penalty factor (PF) for condensation versus heat transfer coefficient (α) for eight fluorinated propene isomers and several additional refrigerants, source (Brown, 2009)

The combination of thermodynamic and transport properties enables to compute the efficiency of air conditioners and chillers and to design their heat transfer surface areas. Compacity and transport properties are essential in order to reach affordable efficiency levels and in order to avoid adopting solutions with a poor level of resource efficiency. This will be taken into account in Task 6 when studying the economic viability of alternative solutions. Possible alternative refrigerants by type of product are discussed in parts 5.2.10 and 5.2.11.

Refrigerant management

As explained in Task 4, the refrigerant yearly leakage rates indicated in the literature let think that these figures ar rather based on total inventories at country levels and that refrigerant leakage rates include both the machine leaks and the refrigerant losses corresponding to management operations (charging, recharging, complete refrigerant charge loss after failure or faulty operation). It appears that, from scarce leak measurements, the machine leak rates measured ranged between less than 0.5 % and 4 % maximum yearly (Huchet et al., 2006). In the meanwhile, leak rates based upon the national inventory in France have figures ranging from 5 to 10 % for commercial air conditioning depending, close to the figures that can be found in (IPCC/TEAP, 2005).
The same is true at the end of life: technical capability to recover all refrigerant in the circuit is above 90 %, but the refrigerant management at country level is not optimal. This leads to direct emissions due to refrigerants not recovered at the end of life: for instance in France for commercial air conditioning (Clodic et al., 2010), from 40 to 70 % of the nominal charge is estimated to be dumped depending on the product range. Note that at the moment, this mainly regards R22 refrigerant as HFCs installations are not older than 2000 and are thus still in operation.

Regarding refrigerant management however, it should be acknowledged that the level playing field for Ecodesign is limited as the refrigerant management and its recovery and treatment is already managed by the regulation 2006/842/EC. This limits the scope for Ecodesign improvement at product level. A BAT scenario can still be built for Europe to reach as good refrigerant management as in Japan where most refrigerant charge is recovered and treated with only about 10 % leak at the end of life plus the leaks of the product.

Refrigerant charge and leakage: options at product level

Nevertheless, manufacturers work to decrease the direct refrigerant emissions in several directions:

- the development of microchannel heat exchangers in order to reduce the refrigerant charge of their products,
- some manufacturers propose leak detectors included in the products (this is required by the regulation 2006/842/EC for products with a charge superior to 300 kg of HFCs),
- in addition, the results of the measurement of leaks at component level made by (Huchet et al., 2006), revealed the leaky components in the products; supplementary tests performed on new components showed that even new ones could be leaky and a new European standard has been developed in order to increase the leak tightness of refrigeration components; while the field study had found a distribution around 5g/year per component when measuring existing installations from 2004 to 2006, the standard defines classes for component leakage of hermetically sealed and other components from < 1g/year up to <4 g/year[7].

So it appears that BAT leak rates enable to reach 0.5 to 1%/year leakage rate and lower for package units and 1.5 %/year for split units and multi-split units (Onishi et al., 2004).

5.2.2. MOTOR

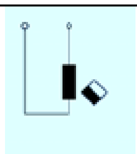
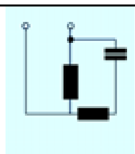
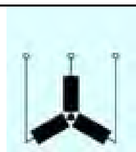
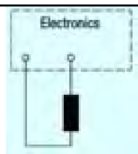
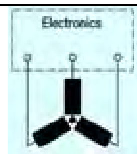
Motor and drive efficiency is an essential part of the air conditioning product efficiency.

Motor types

This paragraph gives some further discussion on types and characteristics. Table 5 - 2 gives an overview of common types of fan motor.

Table 5 - 2 . Comparison of features of main motor types used for fans [source figures EBM]

[7] The precise conditions for the leak measurement are described in the standard - EN 16084:2011 published in July this year and which should be added in the task 1 of the air conditioning system part of ENTR Lot 6 study.

	AC induction motors			EC motors 1	
	Shaded pole	Permanent split capacitor	3-phase induction	Single core	3-core
Circuit design					
Power supply	1-phase AC	1-phase AC	3-phase AC	1-ph. AC or DC	any
Capacity	< 50 W	< 0.5 kW	> 1 kW	< 5 kW	< 5 kW
Integrated VSD	N	N	N	Y	Y
Rotor type	Squirrel cage	Squirrel cage	Squirrel cage	Magnetic rotor	Magnetic rotor
Efficiency	Poor	Medium	Good	Better	Best
Noise	Poor	Medium	Good	Medium+	Good+

Part of the air conditioning product fans are already equipped expensive DC motors with electronic commutation (EC[8]). EC motors have integral speed control (VSD) with higher efficiency than VFDs for AC motors. Some small units even have a fan-specific sensorless control algorithm that can maintain constant flow rate or constant pressure rise. These motors have much lower losses than AC induction motors due in part to use of permanent magnets instead of electrical currents in the rotor. There is also no slip in rotor speed, unlike AC induction motors.
Larger motors are traditionally AC induction (asynchronous) motors, of which 3-phase motors with 4-poles are the most efficient. However large EC motors are gaining popularity for both axial and centrifugal fans. They cost more than the combined cost of AC motor with VFD, yet have lower LCC due to their higher efficiency, especially under part-load operation. They are also quieter and smaller, enabling to gain compacity for AC products. All the largest motors (either AC or EC) run on 3-phase mains.

Motors for compressors are mainly three phase induction motors. EC (BLDC) motors are now commonly used on scroll compressors and recently, some screw and centrifugal compressors were equipped. This allows, in addition to a more than premium efficiency to benefit from the VFD, an important benefit for air conditioning application with variable loads.

Motor efficiency at rated power

The efficiency of electric motors depends on many factors. Figure 5 - 3 shows how peak motor efficiency depends on size. Larger motors are more efficient than smaller ones because losses do not scale up in proportion with power. For example, magnetic leakage at the ends of stators is scaled by a

[8] Brushless DC Motors with permanent magnet rotor (full acronym BLDC), commonly known as 'EC-motors'

length-to-volume ratio per unit power, and heat loss is scaled by a surface-to-volume ratio per unit power. For the same reason, small motors have lower part-load efficiency (Table 5 - 2).

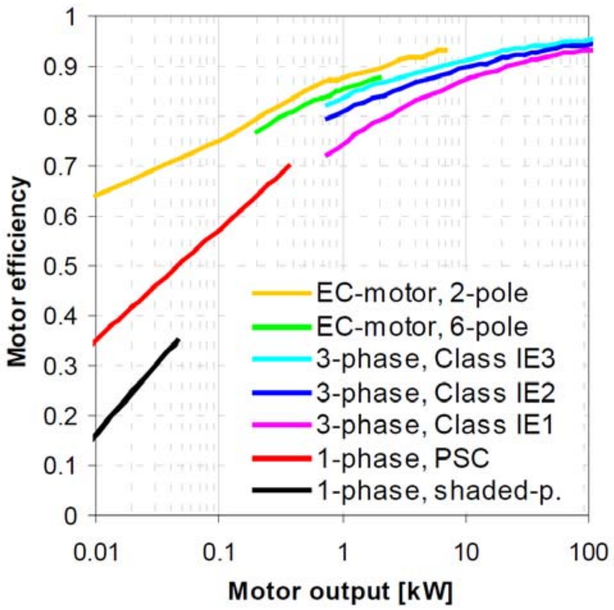


Figure 5 - 3 . Peak motor efficiency depending on motor size, for different motor types. The three IE classes for 3-phase motors are for 4-pole induction motors at 50 Hz (60 Hz motors below 25 kW have approx 87% of the losses)

It is to be noticed that improvements in the air conditioning segment in Japan led to improved EC motor performances both for small fan motors (mostly rated below 100 W electric) and small to medium capacity compressors (between 500 W and about 5 kW electric). (ECCJ, 2008) indicated in 2005 already fan motor efficiencies higher than 80 % and motor efficiency for compressors close 95 %. These supplementary gains are reached through the improvement of the line area ratio of winding and the reduction of the low-iron-loss magnetic steel sheet.

As a conclusion, it seems that BAT motor rated efficiencies may be as high as 85 % efficient EC motors for the power range around 100 W and having the same maximum efficiency of 95 % as larger AC motors already from a few hundred W, using the latest generation of EC motors. (Barrett, 2011) suggest even higher motor performances of 96.5 % for EC motors of high capacities, as reported in Figure 5 - 8.

Motor part load performance

In most air conditioning products, fan and compressor motors operate at variable load. An efficient load control is thus primordial for the product efficiency. This can be made using a variable frequency drive. VFD have their own inefficiencies (power converter), which vary with the load, as does the one of motors. Table 5 - 2 shows the typical part load performances of different system components of different sizes. Figure 5 - 4 illustrates the overall efficiency of a 3-phase AC induction motor together with a VFD. It shows a clear drop below approx. 50% of maximum load.

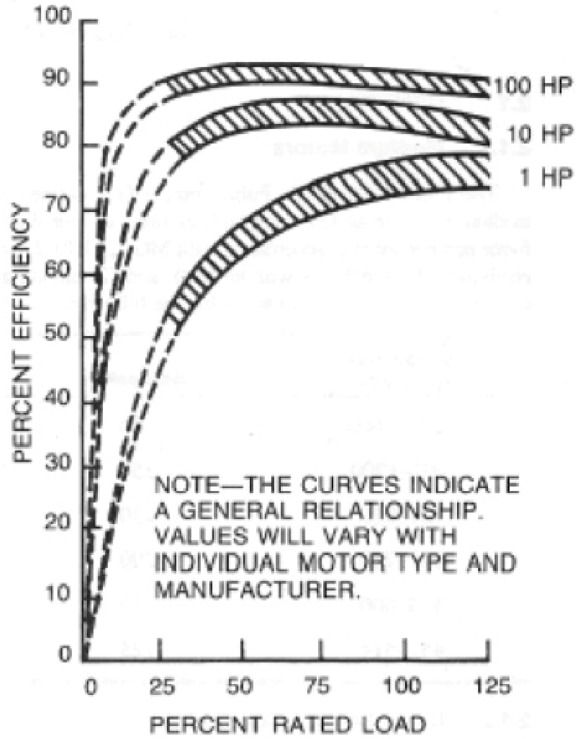


Figure 5 - 4 . Part-load efficiency curves of 3-phase AC motors [source: NEMA, Standard MG-10]

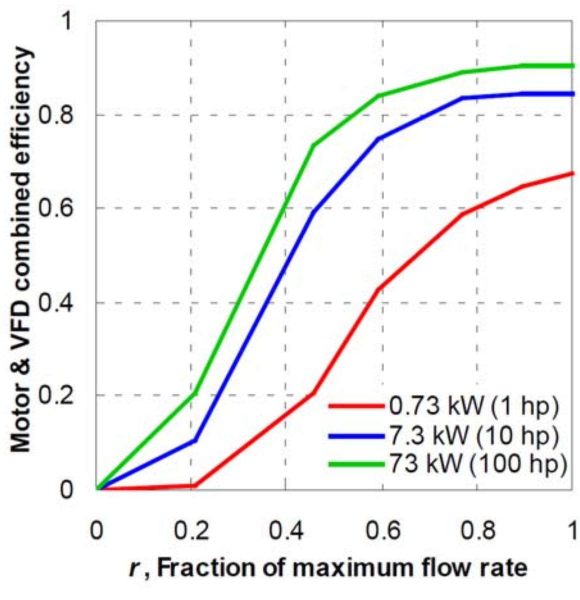


Figure 5 - 5 . Approximate part-load efficiency curves of the combination of direct-drive AC fan motor and VFD, depending on size (kW)

(ECCJ, 2008) indicates much better part load curves regarding the motor alone for rotary and scroll compressor EC motors of the last generation, as reported Figure 5 - 6 below.

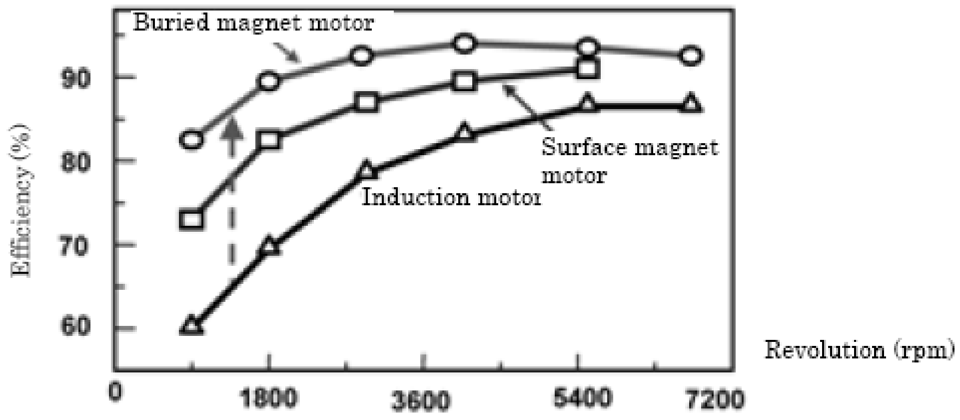


Figure 5 - 6 . Part-load efficiency curves of compressor EC motors for air conditioners bor business use (cooling capacity between 3 and 30 kW), source (ECCJ, 2008)

For higher power ratings, typically used with the magnetic bearings centrifugal compressor, McQuay (Barrett, 2011) indicates higher efficiencies, reported in the figure below, that are for the power range of this technology, i.e. from about 40 kW motor output and above (according to chiller models using this technology available on the EU market).

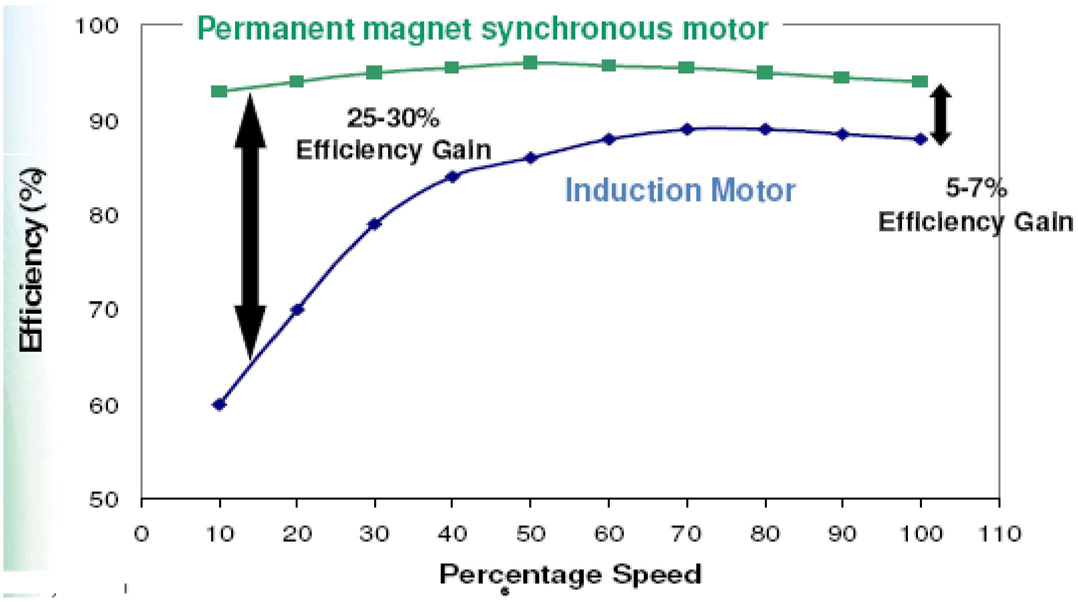
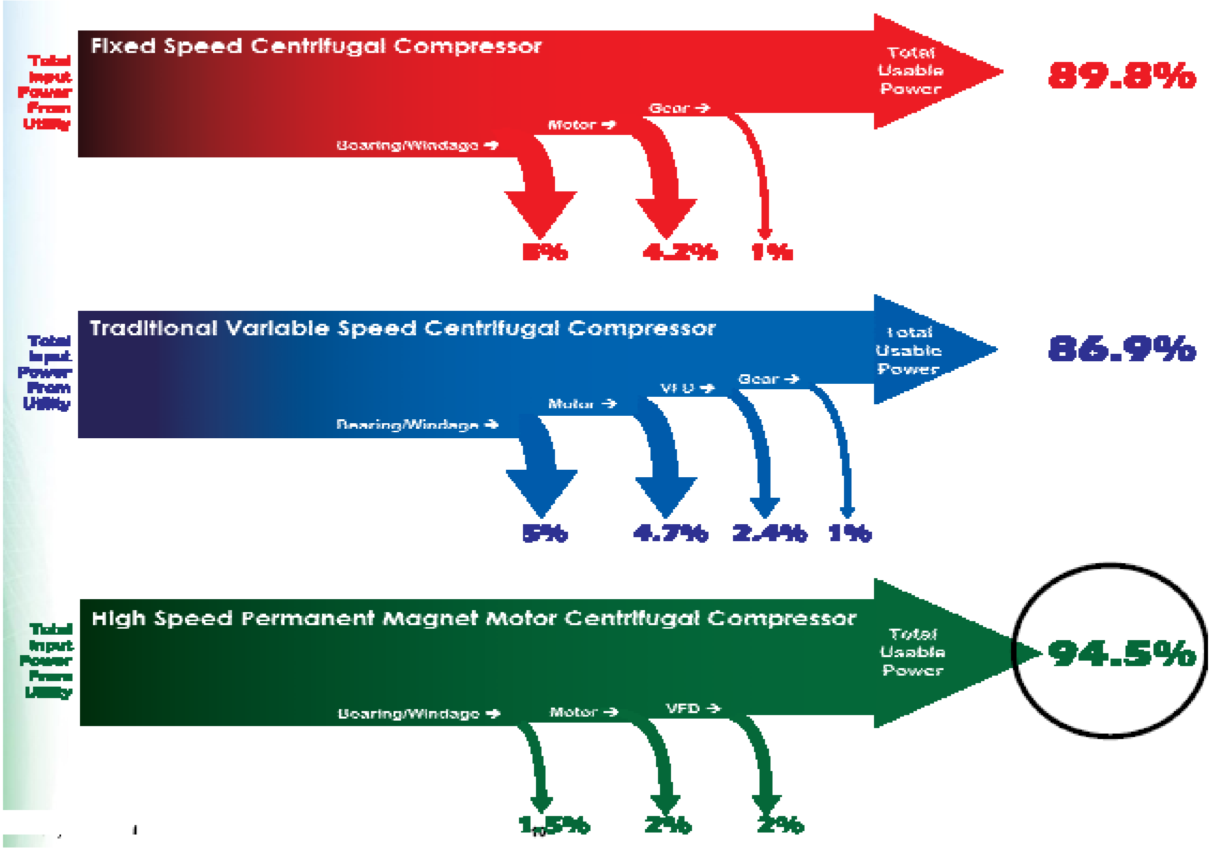


Figure 5 - 7 . Part-load efficiency curves of compressor EC motors plus drive for magnetic bearings centrifugal compressor (typical motor rating above 50 kW), source (Barrett, 2011)

For a recent inverter using a pulse modulated control on a scroll compressor, (Cuevas, 2009) identified VFD losses to be about 98 % at full load and less than 95 % at very low load. On this topic, specific efforts have been made on the form of the signal generated to drive and control the motor rotation speed with pulse modulated control, which is sinusoidal and no longer of square form (ECCJ, 2008). (Barrett, 2011) indicates the total motor + VFD efficiency of traditional motor and variable frequency drives for fix speed, variable speed and recent EC motor and drive. The total efficiency of the motor plus drive can reach 94.5 %, with still 2 % at the maximum speed for the VFD efficiency.

Figure 5 - 8 . Full-load motor and drive efficiency for 3 different types of centrifugal compressors, source (Barrett, 2011)



Motor regulation

The European Commission (EC) decided to phase in MEPS from 2011 as part of the Energy-using Products Directive (EuP). By 2011 these countries will generally require minimum class *IE2*. By 2017, both USA and EU will require *IE3* (Premium efficiency).

Different laboratory test methods are used for motor performance rating, most notably IEC 60034-2-1 and IEEE 112-B. They measure losses in a slightly different way, which can give up to 1-2 % difference.
Motors can be given an efficiency rating based on the above measurements. Standard IEC 60034-30 defines International Efficiency (IE) classes for 3-phase induction motors over 0,75 kW. The 3 classes are: *IE1* (Standard), *IE2* (High efficiency) and *IE3* (Premium). A fourth class, *IE4* (Super Premium) will be added in future to rate higher efficiency motors such as EC motors. The standard harmonizes earlier rating schemes such as European CEMEP[9] (Their *EFF1* rating is equivalent to *IE2*) American NEMA[10] (Their '*NEMA Premium*' rating is equivalent to *IE3*) and the mandatory American Energy Policy Act (The *'EPAct'* rating is now equivalent to *IE2*).
Most industrial states have implemented, or soon will implement mandatory minimum energy performance standards (MEPS) for large electric motors. The first country to do so was the USA in 1997.

[9] Committee of European Manufacturers of Electrical Machines and Power Electronics
[10] National Electrical Manufacturers Association

Regarding compressors, motors of hermetic and semi-hermetic compressors are not included in the Commission Regulation (EC) No 640/2009. For open compressors, the motor efficiency is regulated for AC induction motors (2, 4 and 6 poles) up to 375 kW output. However, an ErP study is to be launched which includes compressor (and their motors).

The 750 W lower end limit means that an important part of indoor fan motors of split, VRF and fan coils are not covered by the motor regulation (which may additionally benefit from the exemption for motors integrated into products). This thus leaves some place to improve the motors in the air conditioning products using the latest generation of EC motors as BAT.

5.2.3. COMPRESSOR EFFICIENCY

Compressor types

The five main technologies of compressor for stationary air conditioning are used for products in the scope of ENTR Lot 6, rotary vanes, reciprocating, scroll, screw and centrifugal although these last three dominate the chiller market in terms of total installed capacity.
Air conditioners above 12 kW generally make use of scroll compressors. Chillers below 12 kW may use rotary compressors or scroll compressors. Up to 100 kW, scroll and reciprocating hermetic compressors can be used. From 100 to 200 kW, scroll compressors compete with semi-hermetic rotary screw compressors for a few models with still some semi-hermetic reciprocating compressors. From 200 kW up to 900 kW, centrifugal models entered the market (from only a couple of years) with the introduction of new DC brushless motor and magnetic levitation bearings. Above that the competition is mainly limited to centrifugal and screw compressors.

Compressor efficiency in standard conditions

The improvement potential of compressor efficiency at full load is thought to be low at present design conditions and were already predicted so in 1998 (Figure 5 - 9).

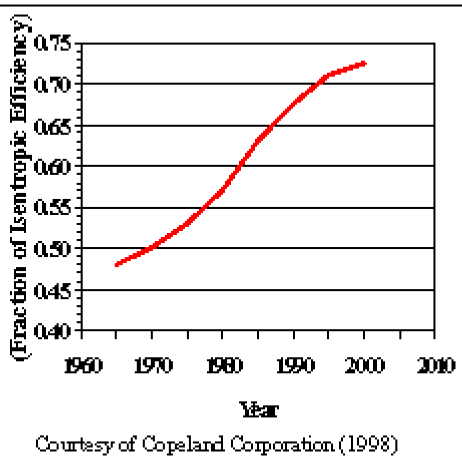


Figure 5 - 9 . Compressor efficiency limit, (DOE, 2001)

In the meanwhile, (ECCJ, 2006) completes this information by giving best isentropic efficiency levels reached by air conditioner compressors nowadays. The total “heat insulated” efficiency, that is the ratio between the work supplied to the fluid and the electric power delivered by the motor of the compressor, reached 80 % in 2004 with a motor efficiency of 95 %, or a total efficiency of about 75 %. This was estimated as a 3.4 (11.6 US units) EER according to the ARI 540 standard on positive displacement compressor in the Lot 10 study (with higher commercially available efficiencies in the USA being 3.34 (11.4), a figure which did not change since 2004 (Ashrae, 2004)). So as to reach these performance levels, it was necessary to add a control valve which controls back pressure of the swirling scroll so that the back pressure can be adjusted based on operating state (optimal trade-off between pressure on the scroll to minimize leakage and avoid friction losses) and to increase the precision of the machine process to make the sliding parts (ECCJ, 2008). BAT scroll compressor efficiency at design point could reach slightly higher performances 1 or 2 % but at cost that scroll manufacturers do not want to reach. In order to reach higher design point efficiencies, it is necessary to move to screw or centrifugal technologies, which are not available below about 100 kW cooling capacity at the moment.

Regarding screw compressors, the ASHRAE (ASHRAE, 2004) indicates a standard maximum peak isentropic efficiency of 80 % to 85 % (at design point), the 85 % being reached for a single screw compressor with a single gate rotor or a modern twin screw compressor, the more common design on the market in Europe. So BAT total efficiencies for screw compressor may be higher than 80 %, using more efficient EC motors and best screw compressors.

Centrifugal chillers enable to reach similar performances as screw compressors for design in standard air conditioning conditions. This latter remark is based upon the observation of chiller full load efficiencies on the EU market and performance requirements abroad, so is rather not precise as many other parameters should be taken into account to draw such a comparison.

What is the optimal design point?

As for cooling products, compressors almost never work in the conditions they were optimized to, e.g. full load for rating conditions of EN14511 standard. The figure below shows the typical performance curve as a function of the compression ratio (ratio of the high pressure to the low pressure, main parameter influencing the compressor efficiency) for a scroll compressor operating at fix speed. Air conditioning compression ratio design generally lies between 2.5 and 3.5 (Reindl and Jekel, 2003) and is around 2.9 on the Figure 5 - 10 below.

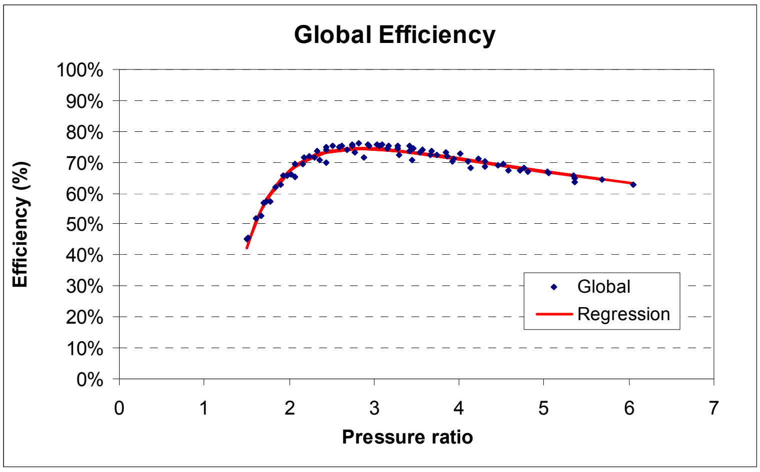


Figure 5 - 10 . Scroll compressor total efficiency (motor, drive an compressor) as a function of the compression ratio, source (Kinab et al., 2010)

As performances decrease when the compressor ratio deviates form its optimum, and that the optimum only corresponds to a limited occurrence in the year in the seasonal performance metrics, this leaves space for optimization. For scroll compressors, this optimum is presently fixed at the time of manufacturing although it should be possible to make it vary with simple mechanisms along the year. For screw compressors, a slide valv may be integrated which enables to have a variable optimal compression ratio (generally noted variable Vi, the V being for a ratio in volumes rather than in pressures).

For reversible compressors, the same thinking is to be done with heating and cooling conditions. In general, heating conditions require higher compression ratios in the seasonal performance indexes as the difference between the outdoor air and indoor air conditions are higher. This means that it may be interesting to have a higher than standard compression ratio for reversible heat pumps, with the consequence that cooling performances would be lower.

However, it should be noticed that means exist to increase the isentropic efficiency of compression when compression ratio increases. A common system is the economizer with liquid injection at an intermediate stage of the compression. The Figure 5 - 11 below illustrates the efficiency gains that can arise from such a system. It appears that the economizer may compensate the efficiency loss at higher pressure ratios. A compressor optimized at lower compression ratio than standard and built with an economizer for higher pressure ratios for high outdoor temperature in cooling mode or low outdoor air temperature in heating mode for reversible products thus may enable to maintain the peak efficiency almost constant over the whole range of operations, which clearly is a BAT for this study.

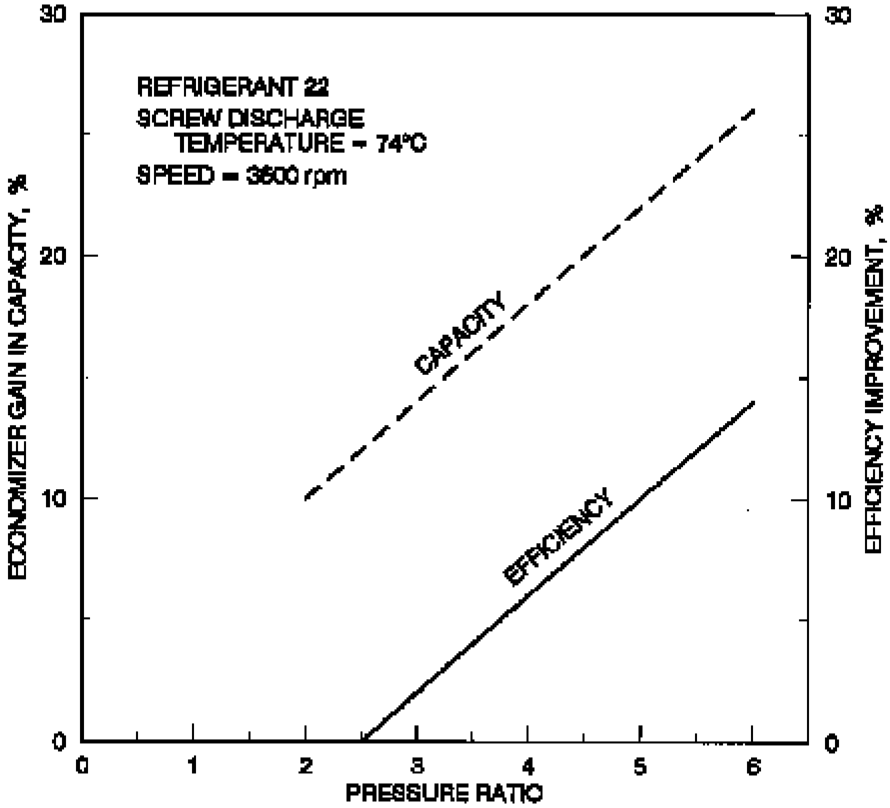


Figure 5 - 11 . Impact of an economizer system upon the isentropic efficiency at high pressure ratios, source (ASHRAE, 2004)

Part load: variable frequency drive, unloading and compressors in parallel

Peak conditions in cooling (resp. heating) mode require higher pressure ratio than under part load and lower (resp. higher) outdoor air/water temperatures. This leads, whatever the refrigerant fluid used is, to higher thermodynamic cycle performances, and basically implies that the part load efficiencies that can be reached in part load and reduced temperature conditions are higher than at full load. Depending on the way used to control the capacity of the air conditioner or chiller, this may however results in higher and lower performances.

When cycling on and off the compressor, the performance is generally lower than at full load. Performance curves for cycling air conditioners are given in the task 4 of Lot 10 study (Rivière et al., 2009). Default performance curves are now available in the prEN14825:2010 standard, including for water based chiller/heat pumps.

One common technique to larger than 20 kW air conditioners and chillers is to use two or more compressors in parallel on the same refrigeration circuit. Doing so, when the load is reduced, one of the two compressors is short circuited and the refrigerant flow rate is decreased in the circuit. The heat exchangers thus become oversized, the temperature difference across the heat exchanger is reduced as is the compression ratio thus improving the cycle efficiency. This tandem configuration is only common for scroll compressors. This is the most common capacity control type used on one circuit.

For the lower end capacity of air conditioners and chillers, scroll compressors may be equipped with a variable frequency drive. As there are losses associated, the variable frequency solution is less efficient at full load (about 2 %, see above) and also at part load (up to 5 %, see above) than the combination of severall scroll compressors on the same circuit. However, the VFD may enable to reach lower capacity ratios before to begin to cycle on and off. In addition, using a variable speed drive may help to reduce the first costs and maintenance costs as compared to using several scrolls in parallel on a single circuit.

A common and efficient solution on VRF is to use on the same circuit a VFD compressor and a fix speed compressor. Hence, the VFD compressor is used to match the capacity, the single speed compressor being swithed on when the load is superior to 50 % of the capacity, and off if inferior.

In order to ensure redundancy, most large equipement generally include two refrigeration circuits, which enables one to continue working in case of failure of the other. In that case, the part load performance obtained by working with only one of the two circuits is the same at unfer full load conditions.

Regarding scroll compressors, other techniques are available to control the capacity. Emerson[11] offers two options to that purpose, the UltraTech and the digital scroll® control. The UltraTech technologies is similar to the technology developed by Bristol, it consists in designing a hole with a valve between the scrolls so that the compressor may operate with more or less volume. The digital scroll® control consists in moving one of the scroll volutes along the axis so that for a short period of time the refrigerant passes within the compressor without being compressed. An external controller produces this phenomena at high frequencies, giving a high frequency. The result is a type of on-off control which ensures the precision of a variable speed one. The control is good but the efficiency at part load is not comparable to the one of variable speed because when the scoll volutes do not compress, there is still an electric consumption. The efficiency at part load of both technologies are given below.

	UltraTech	Digital
Target	Ducted Systems	Non-Ducted Systems
Number of Evaps	One	Multiple
Ratings	SEER Based	EER/IPLV Based
Capacity Range	67% Or 100%	10% To 100%
Full Load Efficiency	95%	100%
67% Load Efficiency	94%	87%
SEER Rating	Best	Good
IPLV Rating	Good	Best

Table 5 - 3 . Part load performances of UltraTech® and Digital Scoll® Emerson technologies, source Emerson

Regarding screw compressors, the standard means used to control the capacity is a slide valve. It is a valve with sliding along the rotor. It controls capacity and the location of the part load port at part load (main mechanism used for twin-screw compressors). This leads to important inefficiencies when operating under part load conditions. These inefficiencies may be compensated at higher loads by the increase of cycle performance due to lower compression ratio but below about 66 % load leads to part load performances poorer than at full load (EECCAC, 2003).
A standard solution to improve the situation is first to avoid to use the slide valve and to rely as much as possible on several circuits in order to limit the needs to work at part load on a single circuit. More recently, some manufacturers developed, for single speed screw, a variable speed solution with a performant BLDC motor. This leads to very good seasonal performances on the ESEER scale.

11 http://www.emersoncanada.ca/pages/energy/presentations/Scroll_Compressor_Technology_Optimizing_Efficiency_Feb08.pdf

Centrifugal compressors are turbomachines and needs higher rotating speeds to work efficiently. Their capacity control is generally done with speed rotation control above about 50 % capacity and using mechanical prerotation vanes. Both methods of control are rather inefficient, the part load performance (at equal boundary pressur conditions) being rather inefficient. Because of very high full load performances, these machines may still be competitive on a seasonal performance basis.

The best performer on a seasonal performance basis is the centrifugal oil-free magnetic bearing compressor. ESEER of water cooled units largely dominate the EU market with values above 8 and up to 9.5. They have by far the higher ratios ESEER/EER of the market, about 33 % higher than competitors. The same occurs for air-to-water chillers: between 5.1 and about 6, only this solution appears, single screw with VFD reaching about 5.1 at best. In the centrifugal technology, compressor rotation is normally ensured by mechanical bearings, which requires a complete oil system. Going to magnetic bearings enables to remove the oil system which leads to excellent part load performances as compared to standard centrifugal solutions and also to scroll and screw solutions. This is clearly a BAT for the chiller segment. However, it is not used for reversible chillers and below 200 kW cooling capacity.

Oil flooded compressor (Bell et al., 2010)

On the longer term, one possible means of increasing cycle efficiency is to flood the compressor with a large quantity of oil to achieve a quasi-isothermal compression process, in addition to using a regenerator to increase refrigerant subcooling. In theory, compressor flooding and regeneration can provide a significant increase in system efficiency over the standard vapor compression system. The effectiveness of compressor flooding and regeneration increases as the temperature lift of the system increases. This technology is particularly well suited for refrigeration applications could still be very beneficial for typical air-conditioning applications, for which improvements in cycle efficiency greater than 5% are predicted. The beneficial effects of compressor flooding can only be realized if a regenerator is used to exchange heat between the refrigerant vapor exiting the evaporator and the liquid exiting the condenser. This gives a BNAT with a maximum improvement potential of 5 %.

5.2.4. HEAT EXCHANGERS OF AIR CONDITIONERS

The thermodynamic cycle efficiency of air conditioners and chillers increases when the evaporating and condensing refrigerant temperature difference decreases. This difference is primarily constrained by outdoor fluid temperatures, as the refrigerant condensing temperature should be higher than the outdoor fluid temperature, and the evaporating refrigerant temperature should be lower than the indoor fluid temperature.

Nevertheless, there is still a temperature difference between refrigerant and outdoor fluid that can be reduced by improving the heat exchanger performance.

Part of the reasoning and figures below is adapted from the task 6 and 7 reports of the Lot 10 study (Rivière et al., 2008). Nevertheless, this is perfectly applicable to larger air conditioners. Possible restrictions due to the larger size are shown as required.

The thermodynamic cycle efficiency of air conditioners increases when the evaporating and condensing refrigerant temperature difference decreases. This difference is primarily constrained by outdoor fluid temperatures. Nevertheless, there is still a temperature difference between refrigerant and outdoor fluid. This can be reduced by a flow reduction using an efficient capacity control means or by improving the heat exchanger design.

Increasing the heat transfer area

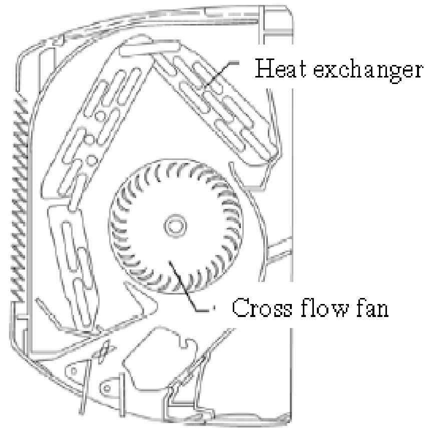
Increase of the frontal area of the coils and/or the number of tube rows is the simplest way to improve the energy efficiency of air conditioners. Amongst the consequences, it also means more copper for the tubes and more aluminium for the fins and potentially larger refrigerant charge.

(Perrotin, 2004) figures out the performance increase with larger heat exchangers for a standard 2.8 kW unit. The improvement is studied at constant cooling capacity, by downsizing (or reducing the frequency) of the compressor. For an 80 % increase in the heat exchange area (by increasing the number of tubes at constant horizontal tube spacing and other physical parameters of the coil), the efficiency of the unit (fans are not taken into account) is increased by 35 % whether the air flow rate is set constant and thus the air speed at the coil first tube row is reduced nearly by a half. With constant air speed at the coil, the improvement in (compressor only) efficiency is also of 80 %. The air flow rate is doubled in that case.

There are limitations to the possibility offered by this technique, in particular space constraints. This is mainly true for indoor units, although in the commercial sector, space may matter also for the outdoor unit depending on its location. Too large heat exchanger surface area may necessitate to change the outdoor and/or indoor cabinet size with significant manufacturing overcost. Also, the increase in efficiency is optimal whether the air flow rate is increased. This means that the noise of the unit is likely to increase too.

A typical wall high efficiency indoor unit is presented on **Erreur ! Source du renvoi introuvable.**. As compared to 1990 units, the indoor unit only had one or two coil parts while nowadays, indoor units with 4 and 5 coil parts appear in order to maximize the indoor heat exchanger area for a given cabinet size.

Figure 5 - 12 . Split high efficiency indoor unit (ECCJ, 2006)



As for 4-direction cassette type indoor units which are used in the most of air conditioners for business use, the change of unit shape, such as from usual square-shape to pentagon-shape, has been attempted, in order to enlarge the heat exchange area without interrupting internal electrical components (ECCJ, 2008).

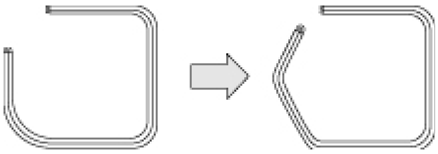


Figure 5 - 13 . Evolution of inner refrigerant copper tube design (Daikin, 2007)

(ECCJ, 2008) shows the evolution of the sizes of the units following the fast efficiency improvement in Japan with the top runner program. For an efficiency improvement of Japanese COP 2.8 to COP 6.2 (EER + COP / 2), the mass of both heat exchangers increased by about 50 %. On the less than 12 kW segment, more recent data show that manufacturers found ways to decrease the mass of the heat exchangers for the same efficiency levels but that dimensions are still at the same level or even higher.

Given the energy efficiency increase associated to heat exchanger surface area, the heat exchanger area increase could by itself explain more than the half of the increase in efficiency.

The heat exchanger area can also be enhanced by increasing the fin density. For reversible units, there is a trade-off between increased performance in cooling mode and decrease in performance in frost conditions in the heating mode. Also in cooling mode, too high fin densities may lead to premature coil fouling (LNBL, 2001). Fan power is increased by higher pressure losses on the air side. Recent progress also includes decrease of the fin thickness, which will also contribute to increase the heat transfer performance.

Increasing the heat transfer performance

The resistance to heat exchange between refrigerant fluid and air can be decreased by improving refrigerant tube design or fin design. For refrigerant tubes, high quality copper is used, and conduction is of very high quality already.

Evolution in the fin pattern was already presented in task 4. Heat transfer rates increase from smooth to louvered, and interrupted surfaces. Evaluation of potential gains on the air conditioner performance suggests that the efficiency could be improved by about 10 % when using slit fins instead of plain fins. According to (Daikin, 2007), several fin patterns have been introduced to maintain good heat transfer quality while achieving lower noise levels and lower cost productions. Different patterns are presented in the figure below. More recent evolutions may introduce air side heat transfer gains larger than 2 as compared to plate fins.

The evolution of the internal groove shape of refrigerant tubes is shown hereunder. Latest inner tube design patterns would enable to cut the heat transfer resistance by a factor four as compared to smooth tubes. This evolution is also coupled to a decrease in the tube diameter and thickness. In order for the heat exchange increase not to be too detrimental to the refrigerant pressure loss in the heat exchangers, the diameter of the copper tube is adapted to the refrigerant conditions, with larger diameters for the gas state and lower diameters for liquid or diphasic conditions (ECCJ, 2006) and (ECCJ, 2008). These measures also enable to decrease the refrigerant charge. More recent evolutions may introduce refrigerant side heat transfer gains larger than 3 as compared to plate smooth tubes.

Figure 5 - 14 . Evolution of the fin pattern (Daikin, 2007)

Use	Indoor				Outdoor				
Type	Type A	Type B	Type C	Type E	Type A	Type B	Type C	Type D	Type E
Fin shape (Line pace Step pace Fin pace)									

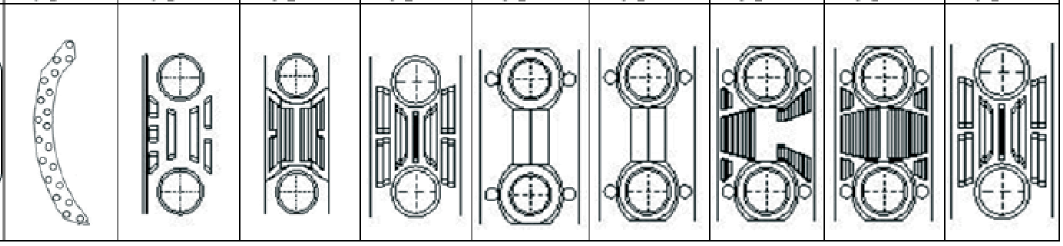


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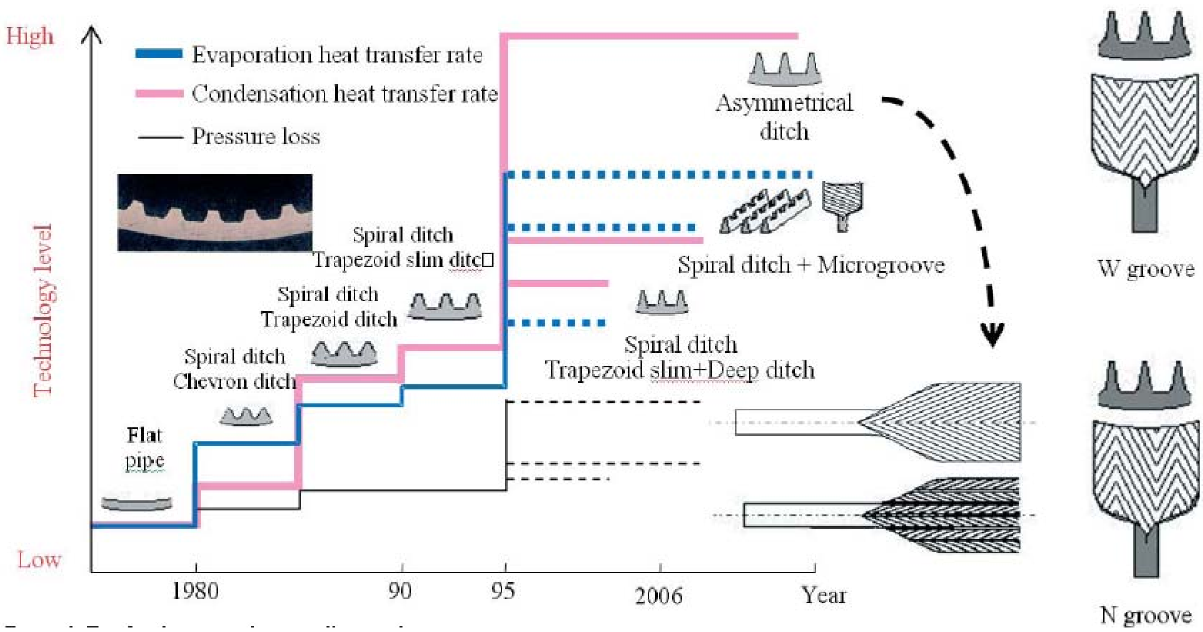
Figure 5 - 15 . Evolution of inner refrigerant copper tube design (Daikin, 2007)



5.2.5. HEAT EXCHANGERS OF CHILLERS

Types of heat exchangers

Air-to-refrigerant heat exchangers

Air cooled chillers use the same refrigerant to air coils as large air conditioners, except the designs differ to adapt to the larger capacities. An alternative is to use micro-channel heat exchangers, as shown herafter.

Regarding refrigerant-to-water evaporators

This type of heat exchanger may be a liquid cooled or a condenser. There are three main solutions in use today for liquid coolers in air conditioning chillers:

- Shell and tube (refrigerant in tube)
- Shell and tube (water in tube) : flooded type
- Brazed plate heat exchanger

Shell and tube heat exchanger with refrigerant evaporating inside tube.

The figure below shows a typical shell-and-tube heat exchanger of this type. A series of baffles channels the fluid throughout the shell side. The baffles increase the velocity of the fluid, thereby increasing its heat transfer coefficient. (ASHRAE, 2008)

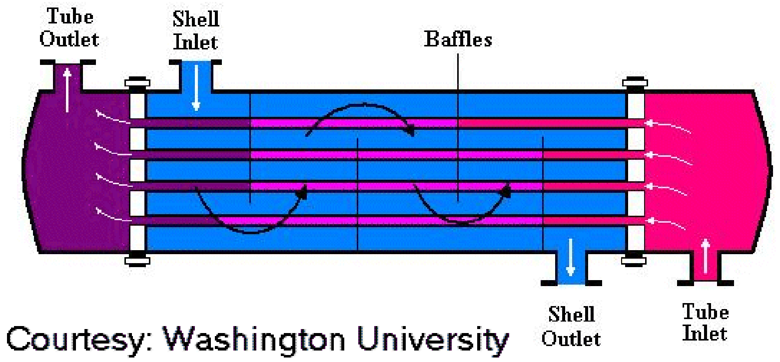


Figure 5 - 16 . Shell and tube heat exchanger

Shell and tube heat exchangers are the standard solution for refrigerant-to-water heat exchangers.

For condensation, refrigerant is flowing inside the tube and water outside the tubes in the shell.

For evaporation, this configuration is possible although the reverse solution may be adopted, rather on water cooled equipement than for air cooled ones. In flooded type heat exchangers, refrigerant is introduced at low sides and heats up on the water tubes up to the top where it can leave to compressor aspiration. It can thus provide refrigerant vapor close to saturation conditions (with very low or null superheat). A mist separator must be used to prevent liquid to reach the compressor. On these systems, oil is likely to accumulate in the heat exchanger if a dedicated oil circuit is not added to ensure the oil return to the compressor. Oil in excess in the flooded heat exchanger is lkely to decrease importantly the performances of the heat exchanger.

Plate heat exchangers consist in stacked metal plates with intermediary spaces in order for water and refrigerant to circulate. For air conditioning chinllers, in most cases, plates are brazed. The distance between the plates is low (3 to 6 mm); hence, the hydraulic diameter is very low and ensure high transfer capability. The plate thickness is about 0.15 mm, which gives an excellent conduction. Turbulency is created thanks to corrugations on the plates. The lower the β angle, the higher the heat transfer coefficient, as well as pressure losses. The ratio between the distance between two corrugations and the distance between plates strongly affects the performance of the heat exchanger (P_{co} on the figure below). Plate heat exchangers may be supplied with different corrugation angles for the two fluids, which enable to optimize independently both side heat transfer coefficients and pressure drops.

These heat exchangers are more compact than shell and tube ones for the same capacity and enable to lower the refrigerant charge significantly. They are common in air cooled chillers equipped with scroll compressors. For the larger capacities, shell and tubes are more common at the evaporator side, and at the condenser side. Since they are brazed, it is not possible to repair in case of failure.

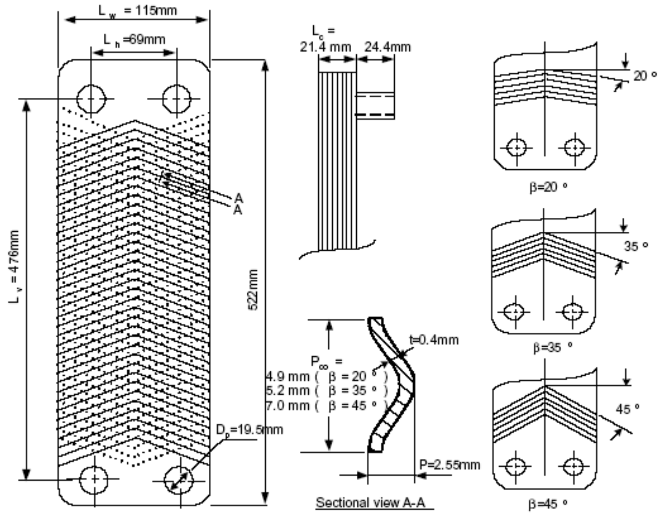


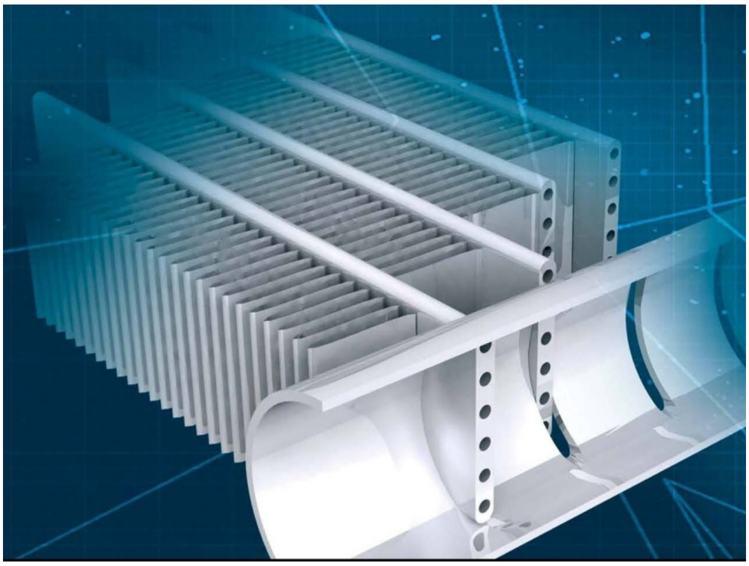
Figure 5 - 17 . Vertical brazed plate heat exchanger, source (Han, 2003)

Microchannel heat exchangers

Microchannel heat exchangers are made of flat tubes with rectangular cross section with dimensions of 1 to 3 mm. Fins pass between the tubes and are brazed to the tubes. The resulting microchannel coil transfers more heat per unit of face area than present heat exchanger of comparable capacity. It does so with a lower airside pressure drop, yielding reduced fan power consumption (USDOE, 2001). There is then a good opportunity to decrease the material required and the costs and thus to offer units with largest equivalent heat exchange areas at equal cabinet size. The refrigerant charge could be decreased by 20 to 40 % for package units (HPC, 2007) for a coil of equal capacity. In the meanwhile, the heat exchanger performance can be increased by 10 % (Carrier, 2007) at equal front coil area as compared to traditional round tube and fins air coils.

This innovation has been envisaged for a long time already as the future trend for air conditioner heat exchangers. Already adopted by the car industry for more than 5 years, it has been recently (2007) adapted to cooling only chillers in the EU by Carrier.

Figure 5 - 18 . Microchannel heat exchanger (Carrier, 2007)



The aluminium industry advertises the quality of aluminium heat exchangers with the following points:

"Transition from copper coils to aluminum flat tubes heat exchanger:

- 3 to 5 times longer product life,
- half of the refrigerant charge,
- 95% recycling of aluminum,
- Resistant to ammonia,
- Mobile AC units become lighter,
- Halve the production costs,
- Double energy efficiency by flat tubes. "

For reversible air cooled chillers, Carrier (HPC, 2007) gives the following advantages as compared to traditional condenser coils:
- Heat transfer and thermal performances improved by 10%
- 20% lower air-side pressure drop
- Refrigerant charge reduced by between 20 and 40%
- 50% reduction of coil weight
- Increased reliability as a result of better corrosion resistance
- No increase in chiller cost

Despite, this does not seem to be envisaged as a technical option to improve the efficiency of split (ECCJ, 2006) or single duct air conditioners at the moment. Microchannel heat exchangers are used by several central air conditioner manufacturers in the US for the indoor unit to produce SEER 13 units. (Cremaschi, 2007) suggests that microchannel heat exchangers are primarily used for the indoor heat exchanger and not for the condenser since it would increase EER in cooling mode but decrease COP in frost conditions. One of the problems for the adaptation of microchannel heat exchangers to reversible products is that it prevents condensate coil drainage and consequently may be a concern for the heat pump evaporator.

(Delphi, 2011) announces the microchannel heat exchanger coils are still being developed. Prototypes are already being tested in Europe on air-to-water reversible chiller series products (CTB, 2010).

Thus, aluminium microchannel heat exchangers are a BAT for cooling only air conditioners and for chillers and potentially for reversible units.

5.2.6. EVAPORATIVELY-COOLED CONDENSER

Most air conditioners and chillers are air cooled although some water cooled models are available.

Smaller water cooled air conditioners may be installed as terminal units in water loop heat pump systems. Larger may use a cooling tower, or be installed with geothermal horizontal or vertical heat exchangers or with underground water as the hot source. Water cooled equipment could theoretically leads to higher performance installations (cooling tower + water cooled unit) but there is no or limited market for these appliances. Consequently, the efficiency of water cooled air conditioners is limited, mainly because the models have not inherited the more recent efficiency options of air cooled equipment.
The higher efficiency is due to the lower refrigerant condensing temperature that can be reached with a cooling tower than with an air cooled condenser. The air cooled condenser must release the heat above the dry bulb air temperature (plus the necessary temperature difference between the air and the refrigerant) while the cooling tower must release it above the wet bulb temperature (plus the temperature difference between the wet bulb temperature and the inlet water temperature).

It is still possible for air cooled condensers (with refrigerants, like in air conditioners or in chillers, or with water, like for dry coolers) to extract heat at the wet bulb temperature (or close to the wet bulb temperature) using the evaporative cooling concept to humidify and cool the air before the condenser coil.

A scheme of a dry cooler (also called adiabatic) with wetted pads is shown below. Tap water is circulated over wet pads before entering the condenser coil.

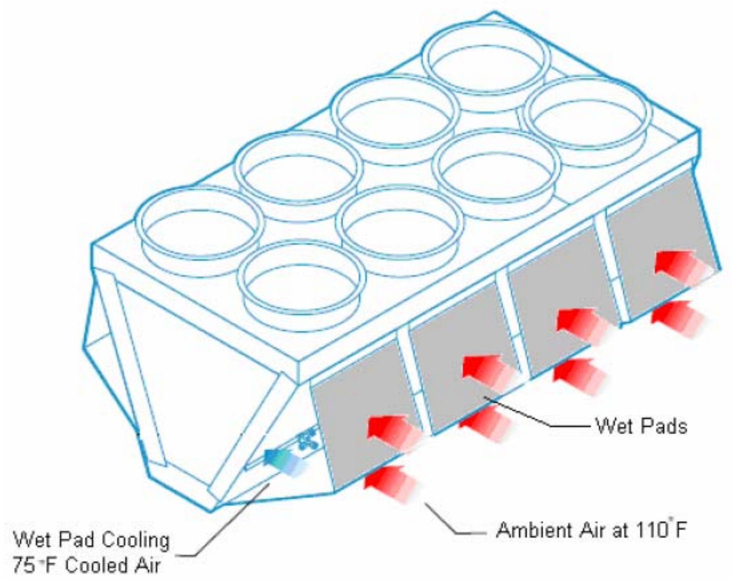


Figure 5 - 19 . Scheme of an adiabatic cooler (cedengineering.com)

There are two main techniques used, a wet pad cooling the air or the direct aspersion of tap water onto the coil to be cooled. In both cases, part of the water evaporates in the air and another part drops down in a sump where it is recirculated. The main problem with direct aspersion is that external fouling of the condenser tube is likely, especially on the hottest part of the condenser and the wet pad option is generally kept.

The energy efficiency gain has been computed on the reference climates used in prEN14825:2010 over the cooling season, supposing the dry bulb air temperature could reach the wet bulb temperature. The estimated gains range from 15 % to 25 % in average yearly from Helsinki to Athens. As the European climates are not too humid the gains are important. In Milan, reputed as a hot and humid climate, performance gains are still about 15 %.

The counterpart is of course the water consumption. A complete product, meaning a standard US central air conditioner condensing coil + the system to humidify the air as a single unit, was accredited in California (Freus system[12]). The California Energy Commission decision (CEC, 2005) is negligible. Simple calculations show that the water consumption to wet the pad is lower than the water economized from avoided electricity production form power plants, which are among the largest water consumers.

With the following default values from the Freus equipment and testing:

- 3 % EER decrease per K condenser inlet air temperature increase
- the default consumption is about 2 liters per kWh of cooling energy delivered,
- 28 l/kWh_electricity as standard water consumption for electricity generation (MEErP figure),

It appears the water consumption on total could be reduced although in Europe.

Also note that the larger gains are at peak load. Indeed, in standard climates, the temperature difference is larger at peak conditions, being 9 K or more on the three climates below. 15 K in Athens may represent an economy up to 45 % on the power consumed at peak time.

[12] http://www.freus.com

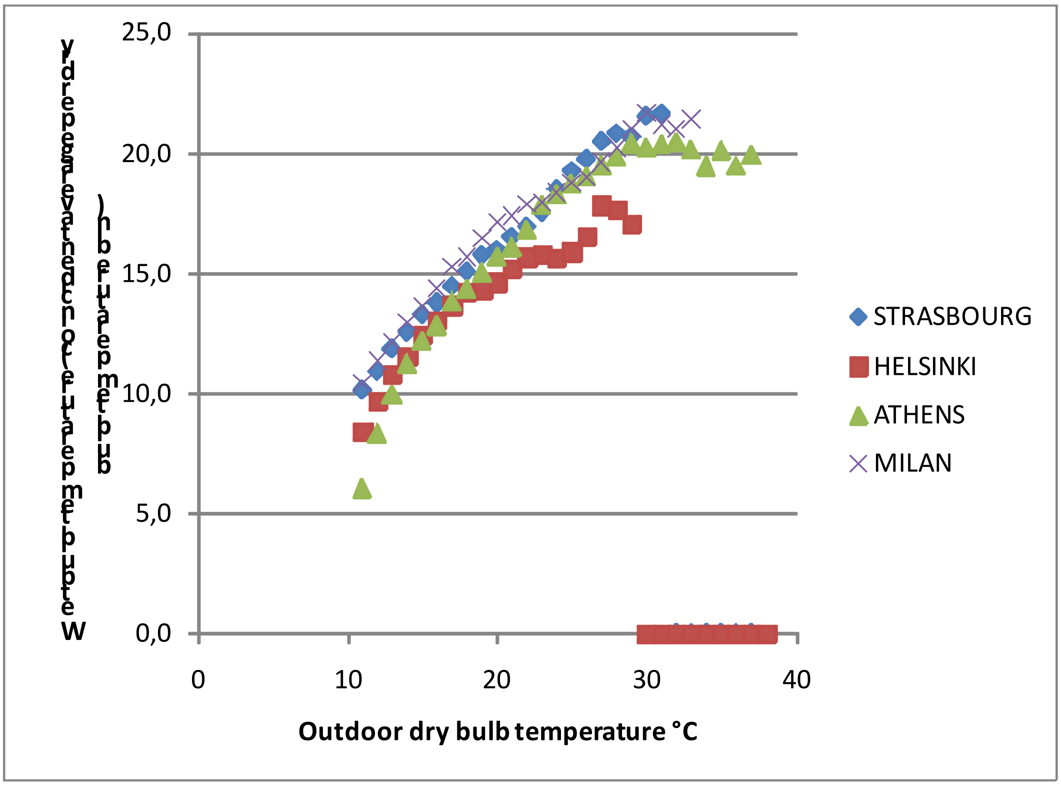


Figure 5 - 20 . Dry bulb versus wet bulb temperatures for different EU climates, source ASHRAE IWEC data files

5.2.7. Fans

The different fan types found in the air conditioning units are summarized in the table below.

	Outdoor fan	Indoor fan
Air cooled chillers	Propeller fan Centrifugal fan (ducted outdoor unit)	- -
Split, multi-split and VRF air conditioners	Propeller fan	Tangential Centrifugal
Air conditioner integrated in AHUs	Propeller fan	Centrifugal
Fan coils	- -	Tangential Centrifugal
Dry coolers	Propeller fan	-
Cooling tower	Propeller fan Centrifugal	- -
Fan coils	- -	Tangential Centrifugal

Table 5 - 4 . Different types of fans in air conditioning products

Centrifugal fans

The same types of centrifugal fans used in air handling units and presented in the task 5 ventilation report of ENTR Lot 6, are also used at the indoor side of air conditioners integrated in air handling units (as stated before, a type of AHU) and cooling towers. In smaller sizes, they are also used in part of the indoor units of air conditioners and fan coils. The different types of centrifugal fans are shown below.

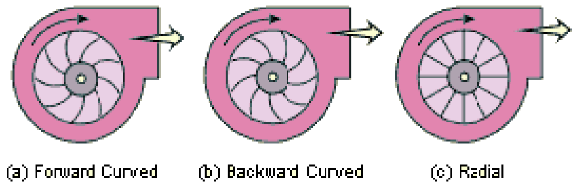


Figure 5 - 21 . F-wheel, B-wheel and T-wheel centrifugal fans in scroll housing, source: US EPA

The figure below shows a vertical fan coil with built-in centrifugal fan.

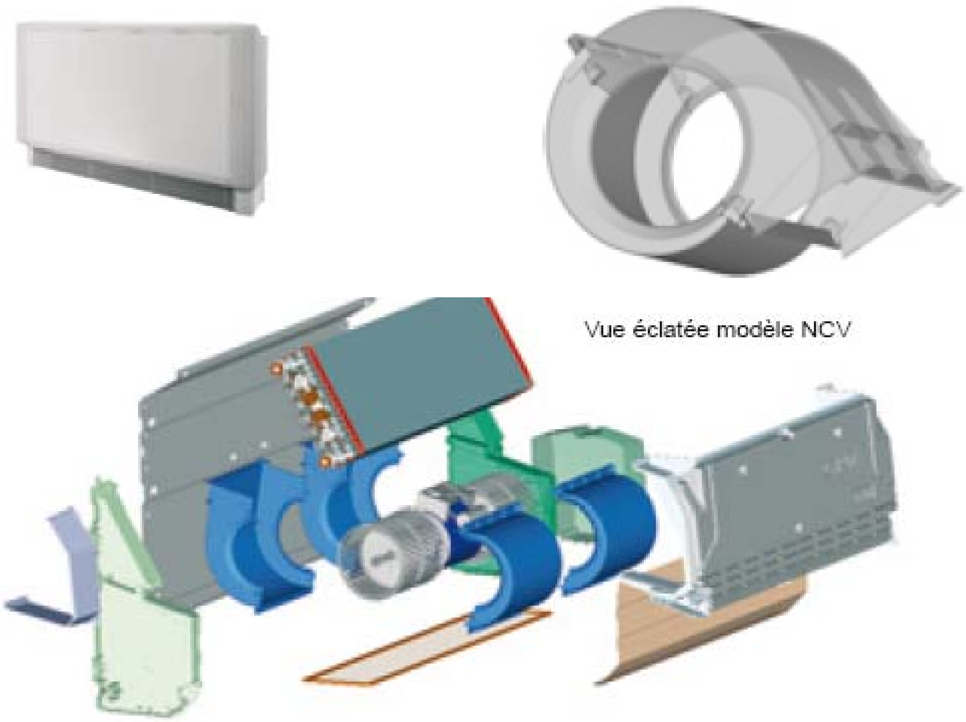


Figure 5 - 22 . Fan coil with centrifugal fan, source CIAT

The figure below shows a backward curved centrifugal fan, free inlet, free outlet for cassette indoor unit of split system.

Figure 5 - 23 . Backward curved centrifugal fan, free inlet, free outlet for cassette indoor unit of split system, source (ECCJ, 2008)

Propeller (axial)

Axial fan is the most common type used at the condenser side of air conditioner and chillers, but also in dry coolers. It is the most efficient type of fans for large volume and low pressure drop. These fans are louder than centrifugal ones and developments have occurred on the shape of the wings in order to reduce the sound power levels.

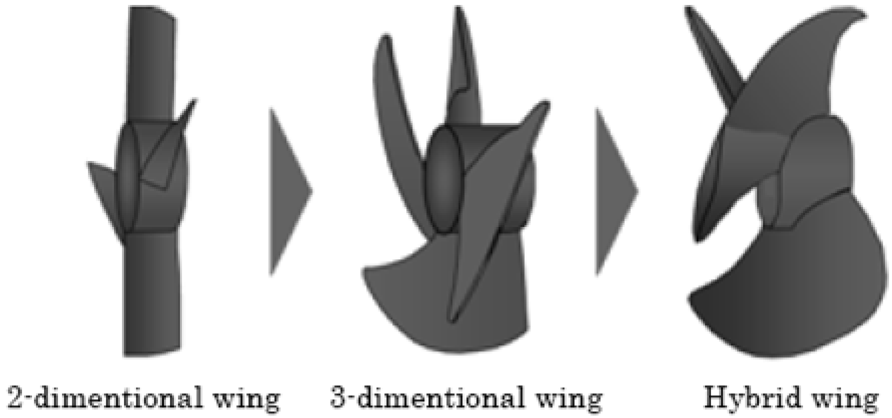


Figure 5 - 24 . Evolution of the shape of propeller fan to reduce their sound power levels, source (ECCJ, 2008)

Tangential

Tangential fans are used for application with low pressure drop and large flows, for instance in wall indoor units of air conditioners and fan coils. These fans, also called cross-flow fan, as the air goes through the impeller, are the lowest efficiency type.

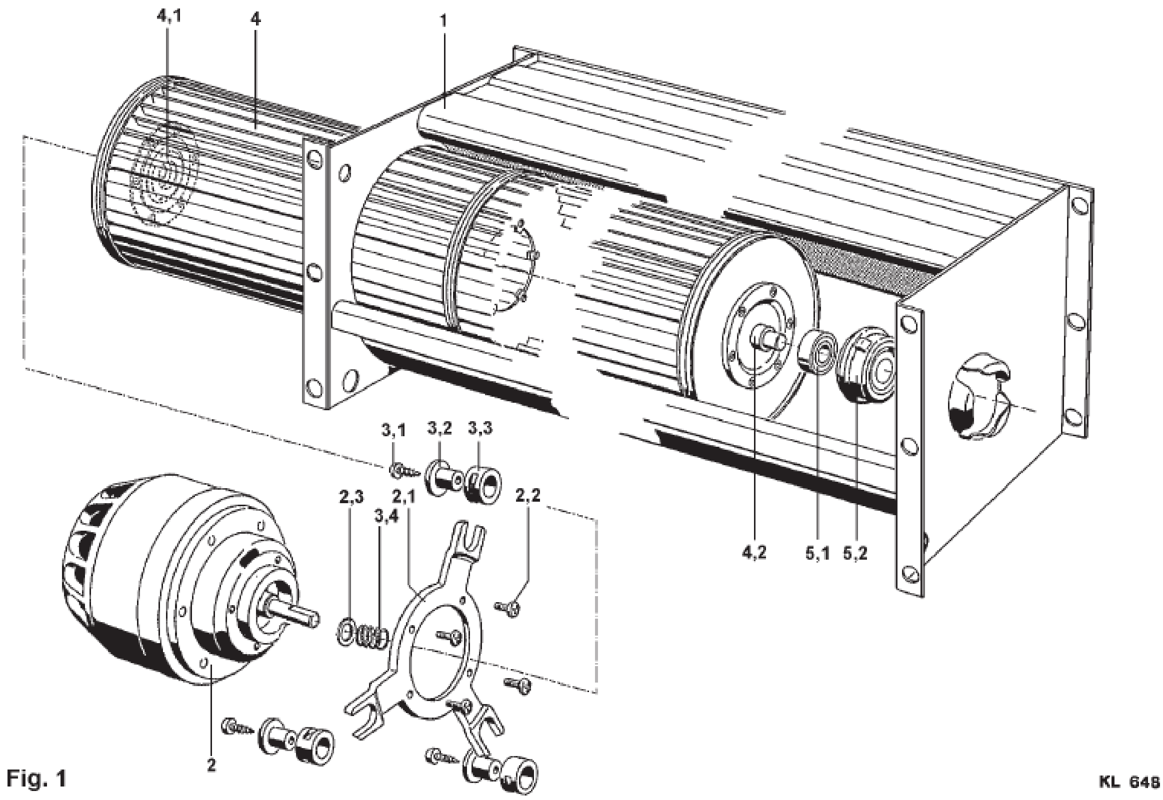


Figure 5 - 25 . Tangentiel fan, mounting and components, Source ZIEHL-ABEGG

Fan selection

Fan selection is primarily a question of air flow and pressure. This explains the design variations amongst the product types. The different characteristic curves for the fan presented above are shown on Figure 5 - 26.

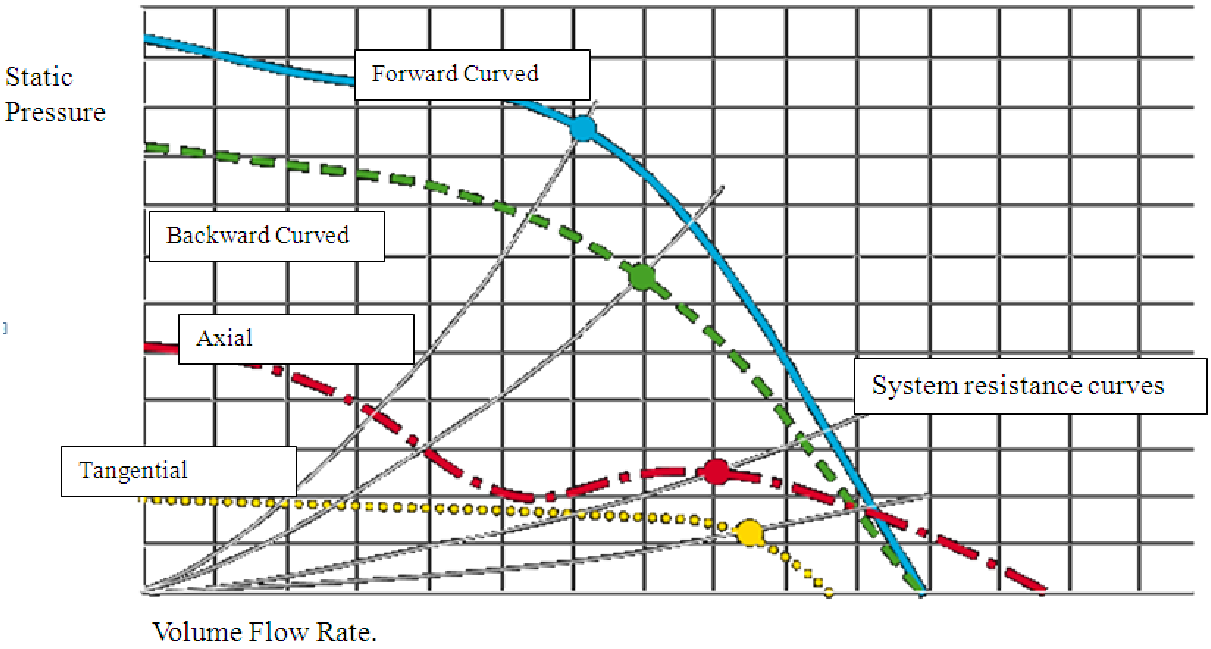


Figure 5 - 26 . Characteristics of different types of fans, Source [www.aircontrolindustries.com]

Fan efficiency: definitions

Fan system efficiency η_{tot} is defined in EN 13799 based on the efficiencies of the single components (fan, motor, belt drive, speed control, etc.)

$\eta_{tot} = \eta_{fan} \cdot \eta_{Motor} \cdot \eta_{Drive} \cdot \eta_{Control}$

η_{fan}	Fan efficiency
η_{Motor}	Motor efficiency
η_{Drive}	Drive efficiency e. g. belt drive
$\eta_{Control}$	Speed control efficiency e. g. frequency inverter

with fan efficiency

$$\eta_{fan} = \frac{q_{fan} \cdot \Delta p_{fan}}{P_{fan}}$$

where

q_{fan}	is air flow through the fan in $m^3 \times s^{-1}$
Δp_{fan}	is total pressure rise from the fan inlet to the outlet in Pa
P_{fan}	is fan shaft electric power demand in W

Fan laws

The so-called ‘Fan Laws’, describe the relationship between volume flow, rotation speed, wheel or impeller diameter, pressure and absorbed power. An overview is given below:

1. **Volume flow:**

$$q_{v2} = q_{v1} \times \left(\frac{n_2}{n_1}\right)^1 \times \left(\frac{d_2}{d_1}\right)^3$$

2. **Pressure:**

$$p_2 = p_1 \times \left(\frac{n_2}{n_1}\right)^2 \times \left(\frac{d_2}{d_1}\right)^2 \times \left(\frac{\rho_2}{\rho_1}\right)^1$$

3. **Absorbed power:**

$$P_{R2} = P_{R1} \times \left(\frac{n_2}{n_1}\right)^3 \times \left(\frac{d_2}{d_1}\right)^5 \times \left(\frac{\rho_2}{\rho_1}\right)^1$$

4. **Density:**

$$\rho_2 = \rho_1 \times \left(\frac{B_2}{B_1}\right) \times \left(\frac{T_1}{T_2}\right)$$

5. **Sound power:**

$$L_{W2} = L_{W1} + 55_{\log 10}\left(\frac{n_2}{n_1}\right)^1 + 55_{\log 10}\left(\frac{d_2}{d_1}\right)^1 + 55_{\log 10}\left(\frac{\rho_2}{\rho_1}\right)^1$$

6. **Efficiency %:**

$$\frac{q_v \times p_t F}{10\ P_R}$$

7. **Total pressure:**

$$p_t F = p_s F + p_d F$$

8. **Velocity pressure:**

$$p_d = 0.6\ v^2 \text{ (Standard air)}$$

Nomenclature for symbols used:

q_v = volume flow of air, m^3/sec
n = rotational speed of fan (e.g. in rpm= rounds per minute)
d = diameter of fan
p = pressure developed by the fan
p = density of air, kg/m^3
P_R = power absorbed by the fan, kW
L_W = sound power level, dB
B = barometric pressure
T = absolute temperature, K (K = ^oC + 273)
*p*tF = fan total pressure, Pa
*p*sF = fan static pressure, Pa
*p*dF = fan dynamic/velocity pressure, Pa
*p*d = system dynamic/velocity pressure, Pa
v = velocity of air, m/sec

The Fan Laws show that fan diameter and the rotational speed of the fan play a very important role.:

- the fan diameter (d) influences the absorbed power (P_R) to the fifth power. For example, a 10% increase of impeller diameter results in a 41% ($1\text{-}0{,}9^5$) energy saving.
- the fan rotational speed (n) influences the absorbed power (P_R) to the third power. For example, a 20% decrease of the speed leads linearly to a 20% reduction in air flow, 36% in pressure difference and almost 50% decrease ($1\text{-}0{,}8^3$) in absorbed power.

Fan efficiency and size

The consequence of the fan law is that higher efficiency can be reached with larger diameters. This is shown in the Figure 5 -27.

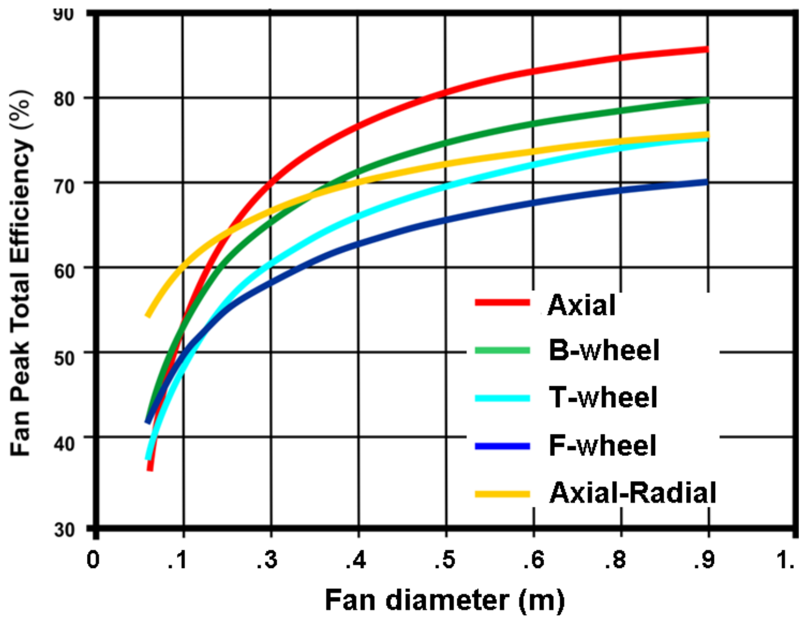


Figure 5 -27 . Fan peak efficiency curves, source Engineering & Manufacturing Corp

There is only limited information regarding the tangential fan efficiency, which is however reputed to be lower than for the other fan categories.

Air conditioner manufacturers have adopted three-dimensional process of blades to optimize their efficiency and limit their soun power emission (ECCJ, 2006) (ECCJ, 2008).

Interest in varying the fan rotation speed for air conditioning products

For all air air conditioning systems, variable air volume (versus constant air volume) air handling is a clear improvement regarding the total air conditioning system efficiency.

Regarding dry coolers, cooling towers, the air flow is used as the main capacity control means, and the rotation speed of the fan is one of the means used to reduce the capacity of these products (two-speed or variable speed drive).

For fan coils, there may be two capacity control types used to reduce the cooling/heating power, less water flow rate, or less air flow. In practice, reducing the air flow leads to much higher efficiencies at reduced capacities and this is consequently the main control used (typically with several speeds offered).

For air conditioners and air cooled chillers, the speed variation of the fan(s) is not a priori an improvement. The larger the air flow and the better the heat exchange, which is likely to lead to the lower compressor energy consumption. This is true except at low cooling loads. In those conditions, the relative share of the fan(s) over the compressor energy consumption increases and the optimal performance may be for a lower air flow. in general variable rotation speed of the fan gives more design flexibility, for instance it enables to adjust the right air flow for cooling/heating mode (which may differ) or to improve the performance of dynamic operation (cycling, defrost).

Fan drive efficiency

The fan can be coupled to the motor using a belt. The motor torque is transferred to the fan by a rotating belt. A disadvantage of belt operation is that it incurs energy loss of over 10 % if poorly designed or maintained, and the losses are substantially higher at low load. Because of these particles, there should be a fine air filter downstream in the supply air path, if the air moved is to be introduced indoors. Belt operation makes it possible to change the fan speed by adjusting the exchange ratio between the motor and fan.

Belt drive full load efficiency varies between 70 and 97 % for more than 1.1 kW fans, and increases with capacity (see Task 5 ventilation report, ENTR lot 6). Flat belt are the more efficient ones.

Some of the air conditioning products are still using belt drives: rooftop air conditioners (indoor fan), cooling towers.

However, most modern fans are generally direct-driven, that is, the fan sits on the motor shaft. Direct-drive fans avoid transmission energy losses.

Large direct-drive fans usually have VSD. For AC motors this is costly and incurs a similar loss to belt operation.

However, VSD provides a number of advantages:
- The ability to regulate the amount of air to a minimum level, instead of shutting off air flow completely.
- Optimal efficiency of the fan motor

Fan material

For indoor fan and propeller fan, processed meta sheets have been replaced in a number of situation by plastics.

5.2.8. EXPANSION VALVE

Thermostatic expansion valves (TXV), electronic expansion valve (EXV), capillary tubes, short tube orifices are throttling devices used in refrigeration systems as flow regulating devices.
Capillary tubes and short tube orifices are constant area expansion devices. They are simple and low in cost, but proper flow regulation is limited to a very narrow range of flow conditions. EXV comprising a stepper motor can accommodate a wide range of flow rate regulation and allows a quicker response to variations in operating conditions. Although the TXV is not appropriate for a large range of flow conditions, it has a simple control mechanism and a lower price than the EXV (Kinab et al., 2010). Thus, the thermostatic expansion valve TXV is widely used in the heat pump and air conditioning industry.
TXV and EXV are used as flow rate control device, which feeds back the superheat, and adjusts the mass flow of the evaporator, in order to maintain a constant superheat to prevent liquid refrigerant from entering the compressor.
TXV is growing fast in the air conditioning segment, together with the development of VFD compressors and replaces progressively capillary tubes and thermostatic expansion valves. For some products, still equipped of thermostatic expansion valves, the electronic throttling valves are a BAT.

5.2.9. AUXILIARY POWER MODES (INCLUDING CONTROLLERS)

Auxiliary power modes are less important for larger air conditioners and chillers as the electronic component size is not proportional to the size of the units. Crankcase heaters may remain a hidden and potentially important consumption post for some products. With the entering into force of the prEN14825:2010 standard, the part load formula used to compute the SEERon (and SCOPon for reversible machines) of air conditioners and chillers will include the oil heaters that are energized as soon as the compressor is stopped. This should give a sufficient incitation to keep this power low and to improve the basic control.

5.2.10. AIR CONDITIONERS

Alternative refrigerants

Following the first screening of refrigerant fluids in part 0, specific solutions for air conditioners are discussed.

CO2

At the moment, the only proved and allowed candidate for replacement with the present standard architecture of more common products is CO2. This fluid was in the past used as a refrigerant before being replaced by R12. Its properties make it a quite an efficient fluid but the drawback is that its boiling point is low and therefore it is necessary to work at very high pressures. (Ortiz, 2003) gives a comprehensive overview of the cycle modifications that are needed for CO2 systems to become as efficient as “standard” R410A systems in heating and in cooling mode; the base R410A system is about 3 ton (11.5 kW) and has an EER similar to our base case EER of 3.1 in heating and in cooling mode. Base cycle performances is supposed to be 60 % lower than the R410A system EER and COP and could be increased to 12 EER (3,5 SI) and 13,5 HPF (COP SI = 3,95) with important design modifications. Performances are noticed to be limited by compressor efficiency (that ranges between 0.53 for a reciprocating compressor and 0.70 for a double stage rotary vane compressor following data available in the literature). Whether it is possible to improve the cycle efficiency to values similar to the standard R410A, it is necessary to do so to implement important cycle modifications by adding either a liquid vapor heat exchanger or a turbine (or a piston expander) instead of the expansion valve. Best available technology enables to reach 3.5 EER and COP approximately which is far lower than present proven best available R410A technologies.

However, CO2 is now used for heat pump water heaters (see lot 2 final report for more details) and air conditioning in cars. Improvement in the design of the components including for the compressor and the necessary micro channel heat exchangers could benefit to air conditioners.

(Jakobsen, 2006) reports the development of a CO2 reversible split air conditioner and the comparison of the prototype’s performances with the one of the best available R410A reversible split air conditioner in the Eurovent directory in 2005. It appears the seasonal performances of both units are comparable in heating mode for two different climates, Athens and Oslo. In cooling mode, performances of the CO2 unit at higher outside temperatures give 17 % losses on the SEER in Athens while the same efficiency values can be reached for the Oslo climate. Hence, these conclusions are much more positive than the ones of (Ortiz, 2003) and based additionally on a prototype. However, by lack of detailed data it is difficult to assess the exact efficiency of the system with the indices adopted in the frame of this study.

Daikin developed a VRF working with CO2, and as reported before, the efficiency at full load is lower than for R410A in cooling and in heating mode, while part load performances are not known.

A TEWI analysis of the implementation of CO2 as a refrigerant is proposed in Lot 10 Task 6, part 6.6, for a small capacity split air conditioner and uses the conservative assumptions of (Ortiz, 2003). The results showed that the gain in direct emissions due to the lower GWP may not be compensated bu the efficiency losses. This was done at part load. A complementary analysis is required at full load in order to measure the increase of peak power that CO2 may cause.

Other alternatives

Other alternative refrigerant have one or several drawbacks:

- Propane: performances comparable to the ones of R410A, but the cost of efficiency improvement is much higher due to lower transport and heat exchange performances; it is additionally highly flammable (class A3 EN378),
- HFC 32: performance and potential for improvement of efficiency is similar to R410A; the GWP is higher than 150 (650) but still would enable a consequent reduction in the direct emissions as compared to R410A; it is midly flammable (class A2 EN378); the cost of adaptation could limited due to its low flammability (UNEP, 2010).

- R1234yf: performance and potential for improvement of efficiency is lower than for R410A and the cost of efficiency improvement is much higher due to lower transport and heat exchange performances and volumetric cooling capacities; the GWP is 3 so much below 150; it is midly flammable (class A2 EN378)); the cost of safety adaptation could limited due to its low flammability (UNEP, 2010) but the cost of the product adaptation starting from R410A is high.
- R1234yf blends (UNEP, 2010): this could be a viable option to formulate a blend with R1234yf in order to reach performances closer to R410A and then the situation would be similar to the one of R32;
- ammonia: performance and potential for improvement of efficiency is high due to excellent transport and heat exchange performances; the GWP is null; it is only midly flammable but highly toxic (class B2 EN378) and not advised to be used in air conditioners.

At the moment, EN378 recommends not to use these alternative fluids for air conditioners in buildings, except for hermetically sealed system with a very low charge, which is not adapted to split, mutisplit and VRF systems, which represent the largest market share of air conditioners.

However, midly flammable refrigerants, HFO1234yf and R32 could be used in air conditioners, according to (UNEP, 2010), with an affordable overcost to include safety mechanisms in the air conditioners.

If the GWP 150 target of the regulation EC 842/2006 was adopted for fix installations also, it would most likely lead to use either the R1234yf solution (or one of its isomers more suitable to replace the R410A) or CO2. To use propane and ammonia, an indirect system, like chiller based system, would have to replace the present split architecture (UNEP, 2006).

More information on the technical modifications and associated costs related to alternative refrigerants for air conditioners would help to improve the LCC analysis of these options in Task 6.

Best available products

It is difficult to assess the energy performance of European products as only their full load efficiency is available presently. The distribution of efficiency is indicated in task 4 for the different air conditioning products. For instance, the best available split product above 12 kW has an EER of 4 and a COP of 4.4.

In addition, when part load efficiency is given by manufacturers as in the case of VRF, the rating method used to assess it may not be fully comparable. Here is an example ofr the VRF situation: part load performance may be indicated while shutting off part of the indoor units (case 1) or by reducing the refrigerant flow at all indoor units simultaneously (case 2). The final result is very different in terms of performances.

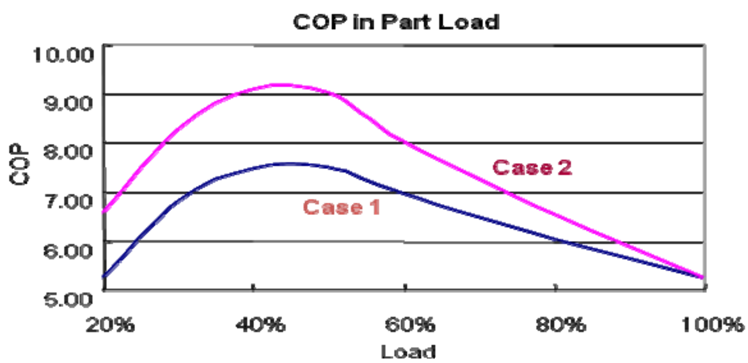


Figure 5 - 28 . Different part load COP reached with VRF products, depending on the capacity control; case 1 – only part of the indoor units work; case 2 – all indoor units work at reduced refrigerant flow

In order to get an idea of best available products, abroad markets reknown for their efficiency as Japan and in some cases the USA are to be used in order to estimate best available products. This gives useful references for policy making.

Best available products globally use part of the best available technologies already, EC motors for fan and compressors, highly efficient and largely oversized heat exchangers, microchannel heat exchangers at the indoor side and even outdoor for cooling only compressors, with the R410A refrigerant, highly efficient compressors and fans, very low low power modes consumptions…

Screening products and best available technologies may help to evaluate the individual and combined effects of improvement options.

Looking at the US market, the screening of the AHRI directory for split products gives already interesting informations (Figure 5 - 29). The EER is low, between 2 and 3, but the part load efficiency SEER is relatively high. On the contrary on the Japanese market, both full load performance and part load performance are high. The reason behind is that the US do not require minimum full load performance while Japan added minimum APF requirements for commercial air conditioners to previous existing ones.

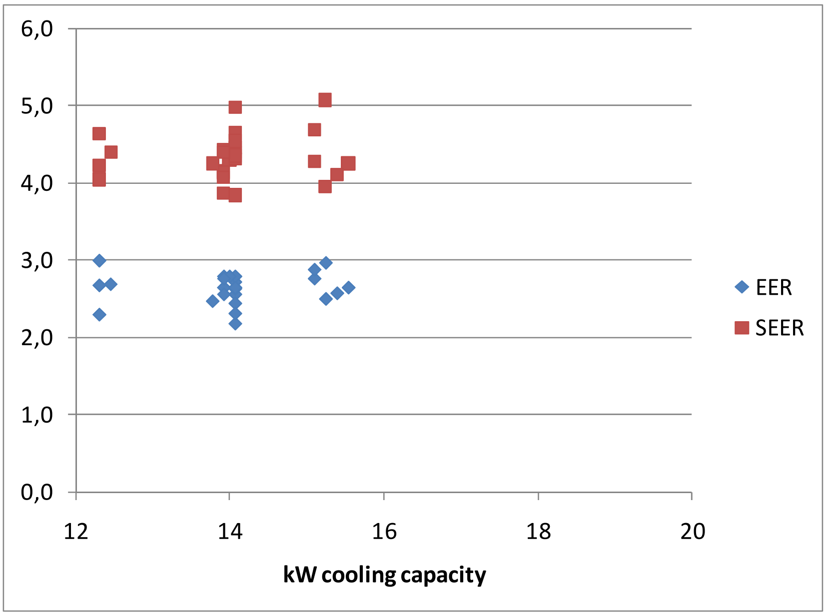


Figure 5 - 29 . US EER and SEER of mini split products with cooling capacity above 12 kW in the AHRI directory (SI units)

The screening of both markets will be considered in order to get insight on best available products. To do this it is necessary to finalize a seasonal performance indicator for Europe (see the part on efficiency metrics below).

About the growing size of air conditioners

With the constraints on full load performances, the size of the heat exchangers is to grow. (ECCJ, 2008) notes several problems accompanying the growing size of air conditioner heat exchangers, a key factor to increase their efficiency:

- Space constraints indoors, for instance for ceiling 4 way cassettes, which size should fit the one of the panels, and outdoors, to replace the older and smaller outdoor units,
- Potential problems with comfort as the rising heat exchanger area may limit the capacity of dehumidification of the air conditioner in cooling mode, or lead to blow warm air not hot enough to the people with the decreasing condensing temperature in heating mode.
- Resource problems, the main one being the availability of rare earth magnets for EC motors.

It should be noticed that these remarks concern efficiency increase of the commercial air conditioners above the present top runner targets in Japan, which is already relatively demanding (see Task 1 for more details).

To be noted also, for air conditioners in air handling units, the space constraint is likely to be more severe than for split or VRF air conditioners. Nevertheless, these types of air conditioners present other advantages.

Air conditioners in AHU (including air conditioning condensing units)

The integration of air conditioning units in air handling units enables to add a number of sedign efficiency options to the air conditioning function. These options are screened hereafter by main product architecture.

Combined cooling generators and AHU

Above the best available technoliges for motors, fan, compressors and heat exchangers, the possibility of the direct fresh air introduction into the building offers a number of efficiency features for air conditioners included into air handling units.

The first largely available option is the free cooling. Several manufacturers propose optimized controls for dampers, and the control can be adapted as a function of the difference in enthalpy conditions outdoor/indoor instead of the standard dry bulb temperature control. This is clearly a feature available to all AHUs except that its effect can be amplified if the units are sized for the cooling function and not only for the ventilation function. In counterpart, the oversizing of the fan induces a supplementary consumption and the balance of both effects is highly dependent on the specific building and climate. So the evaluation of this type of solution is rather in the field of the EPB Directive than in the one of the Ecodesign Directive. What can be certainly included in a AHU standard is to judge the efficiency of the free cooling damper and controls, but this is not thought to be in the scope of this present air conditioning study.

Heat recovery and CO2 sensors are commonly advertized for combined cooling and AHU units ; this is treated in the ventilation study in ENTR Lot 6. Variable air speed drive of the indoor fan of such units is clearly an improvement. However, this mainly contributes to reduce the fan energy consumption with a possible overconsumption of the fan itself. If the whole fan energy consumption is included in the cooling ratings, this is an overall improvement (see for instance the examples in the AHRI 340/360 standard). However, if only the fan power required for the cooling coil pressure losses are included in the energy consumption of the product, in that case, it might not be a smaller improvement or even not an improvement at all. This is a complex issue which might be solved at the level of the EN14511 standard by introducing a regime with reduced air flow at part load.

Hybrid solutions in cooling mode, coupling heat driven desiccant air conditioning and/or evaporative cooling are dealt hereafter in the corresponding paragraphs.
Hybrid solutions in heating mode, combining gas and electricity may also be an efficient options in terms of primary energy consumption. This is to be treated in ENER Lot 21.

Optimized heating or cooling

There exists a number of design choices which are likely to favor the efficiency in cooling or in heating mode. A heating oriented product, may have compressors optimized at higher compression ratios, larger fin spacing, a different condifuration of the evaporator, optimized for frost conditions rather than for optimal cooling performances, or still using an alternative refrigerant as CO2 whose performance in Northern Europe in heating mode may be relatively high, but having low performances in cooling mode.

This is not yet common that manufacturers make these types of choices as it is much less costly to build one single product for a geographical area or even for the whole planet. Nevertheless, such situations may occur. In that direction, the study team approach is to establish the improvement potential in cooling mode and to check that the heating performance which may be reached with such products may respect heating requirements issued from other studies or measures (as ENER Lot 21). Otherwise, this is to be taken into account to establish the cooling potential. The approach is thus similar to the one developed for air conditioners < 12 kW in the ENER Lot 10 study.

Efficiency metrics

In order to establish the improvement potential of air conditioners, it is necessary to crystallize the rules by which the products are evaluated.

The study plans to base on the present prEN14825, which defines the SEER and SCOPindices also for larger than 12 kW air conditioners. Some points may require adaptation for these larger units, which should be discussed in ad-hoc technical groups, this regards for instance:

- Load curve for larger air conditioners
- Climate
- Low power mode inclusion or not, and if necessary with which equivalent hours …

The study wiewpoint on this subject at the moment is to base upon the available index as it is and to use only the SEERon index for larger than 12 kW air conditioners.

Supplementary points have to be raised as well:

- Definitions of default piping length (and height) to declare the performance of VRF systems,
- Definitions of the part load means used to declare the performance of VRF systems,
- Testing conditions for evaporative cooling technologies under part load conditions (link between EN15218 and EN14825)

5.2.11. CHILLERS

Alternative refrigerants

The main fluids in use today are one one side the R134a for large capacity chillers and on the other side the R410A for smaller units. The R407C is still used. It enables the retrofit of existing R22 installations. In addition, it is developed in smaller capacity reversible units as its critical temperature, higher than the one of R410A, enables to design of water based heat pumps (reversible chillers in our case) suitable for higher flow temperatures than with R410A.

The discussion on the alternative refrigerant fluids to be used for chillers is similar to the one for air conditioners except these products are included into indirect systems, which let more freedom to use flammable or toxic refrigerant fluids.

Chillers with propane, ammonia and CO2 (for low temperature application) are already available. So what needs to be discussed to enable the mass development of these techniques is the adaptation of safety requirements for the products in order they can operate with these fluids.

It should be noted however that the potential reduction in global CO2 emissions from replacing high GWP fluids by lower GWP fluids is lower than for air conditioners, as charges are lower, leaks are estimated to be lower because a large part of the units is of the packaged type, and that the efficiency of the recovery increases with the capacity of the units (Clodic et al., 2010) (the total kW and thus refrigerant charge of larger units is important over the total sales).

Thus the potential candidates for the technico-economic evaluation of improvement options in Task 6 are HC32, R1234yf, propane, ammonia and CO2.

Best available products

Air conditioning chillers, standard use

The screening of best available products in Europe, thanks to the Eurovent Certification catalogue gives insight on the potential efficiency increase due to best available technologies at both full load and part load. A more detailed analysis is to be done in order to establish the LCC curves in Task 6.

The distribution of product full load efficiencies and ESEER (Eurovent index) have been displayed in Task 1 and in Task 4. The two figures for air cooled chillers and water cooled chillers are reported hereunder.

The figures below show that there is a larger energy efficiency potential for water cooled chillers, ESEER from 3 to 9 than for air cooled chillers, with ESEER ranging from about 2 to 6. The penetration of the oil free centrifugal compressor extended this potential of 30 % in the range of water cooled chillers. However, its application is limited in capacity range and to cooling only units.

Figure 5 - 30 . Eurovent Certified chillers, air cooled package AC chillers, ESEER Vs EER[13]

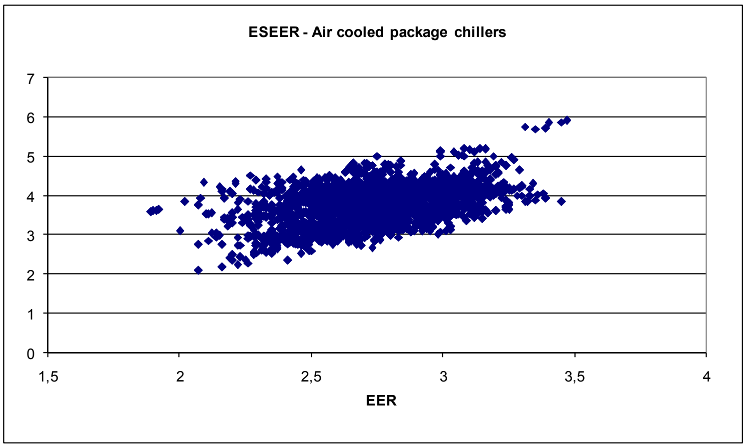
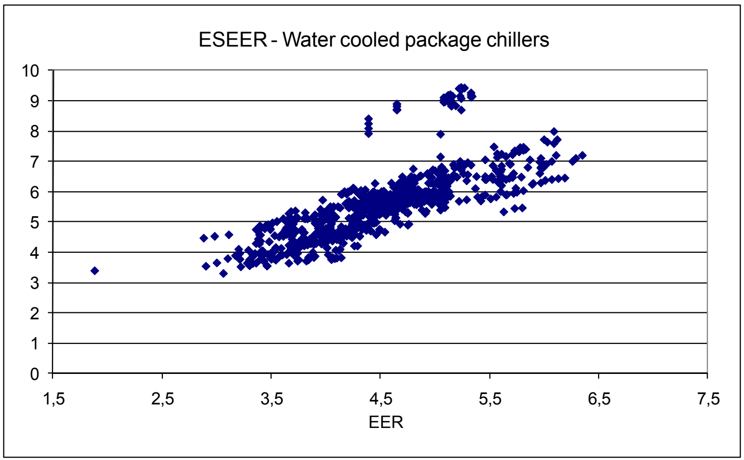


Figure 5 - 31 . Eurovent Certified chillers, water cooled package AC chillers, ESEER Vs EER[14]



The ESEER should be replaced soon by the SEER index from the prEN14825 standard. The first estimates from one manufacturer show that the performances obtained for the average cooling conditions of the prEN14825 to compute the SEERon are close to the ones of the ESEER. This is a point to be deepened within the study. This point is discussed in the efficiency metrics part below.

Air conditioning chillers used for industrial applications

Air conditioning chillers may be used for other applications than comfort air conditioning. More information on these other end-uses can be found in the ENTR Lot 1 report. Non comfort applications generally require longer operation time and potentially more constant regimes of operation. Thus, it is likely that minimum performance requirements for these chillers should be higher, and more relying on the full load performance than on the part load one.

[13] Reproduced from Task 1

[14] Reproduced from Task 1

It could be possible to perform a sensitivity analysis of the findings for air conditioning chillers whether a specific operating profile was included in this study. This could lead for instance to include what a supplementary metrics for non standard operating conditions, as in the ARI 550/590 standard, the NPLV, which stands for Non standard Part Load Value (versus the IPLV which is the seasonal performance index obtained for the

Air conditioning chillers used in non standard application

Chillers may be installed in parallel in chilling plants in what is generally called multiple chiller application. Some manufacturers propose softwares to design building engineers in order to optimize the chilling plant (least life cycle cost combination of individual chillers to match a give nload curve and climte conditions). In the same spirit as for industrial end-uses, chillers may operate very differently depending upon their installation conditions, particularly whether they are included in a chilling plant using several chillers in parallel. A NPLV like approach could also be adapted.

The use of higher than 5 K temperature difference across the evaporator heat exchanger may help economizing energy. The gain in efficiency comes first from the reduced pumping power and also from the increase of the average evaporating temperature of the chiller. 10 to 15 % are figures oftenly quoted (% over the consumption of the chilling plant). Other manufacturers offer variable speed compatible chillers (or chillers with the pump integrated in the small capacities). In both cases, this leads the chiller to operate in non standard conditions.

Free cooling

Chillers may integrate two different types of free cooling options, via the refrigeration cycle or by adding a supplementary (see task 1). As for rooftop, the gain that can be hoped for such dispositive is potentially important, especially for chiller operation at low ambient temperature, but is also very much dependent on the climate. Additionally, this type of options is in competition with other free cooling techniques, as air handling free cooling and free cooling with the heat rejection unit. What is necessary for designers to make rational choices is to characterize the performances of this option. Standardization in this matter could help.

Efficiency metrics

In order to establish the improvement potential of chillers, it is necessary to crystallize the rules by which the products are evaluated.

The study plans to base on the present prEN14825, which defines the SEER and SCOP indices also for chillers. Some points may require adaptation for these units, which should be discussed in ad-hoc technical groups, this regards for instance:

- The potential need to add dry cooler operating conditions for water cooled chillers at full and at part load as the present standard mainly reflect the installation with a cooling tower,
- The potential need to add evaporatively cooled operating wet bulb temperatures at part load,
- Load curve for chillers
- Climate
- Low power mode inclusion or not, and if necessary with which equivalent hours …

The study wiewpoint on this subject at the moment is to base upon the available index as it is and to use only the SEERon index for larger than 12 kW air conditioners.

Although a number of chiller specific technologies are available above the generic best available technologies previously described, most of them are very case dependent and can hardly be evaluated in a single merit figure.

For some of them however, a non standard performance index could be used as in the USA in order to take into account the specific operating conditions associated. This regards:

- the use of higher than 5 K temperature difference across the evaporator,
- multiple chiller applications,
- air conditioning chillers used for other than air conditioning applications.

5.2.12. ALTERNATIVES TO ELECTRICALLY DRIVEN VAPOUR COMPRESSION UNITS USING GRID ELECTRICITY

Increasing the share of renewable energy

Electric vapour compression cycles extract part of the cooling they supply from the ambient air, the ground or natural water in aquifers. For one unit cooling, they supply the following renewable energy quantity (noted RE here), when taking into account the fact non renewable energy was used to produce the electricity:

RE = kWcooling – prim * (kWcooling/SEERon)[15]

With

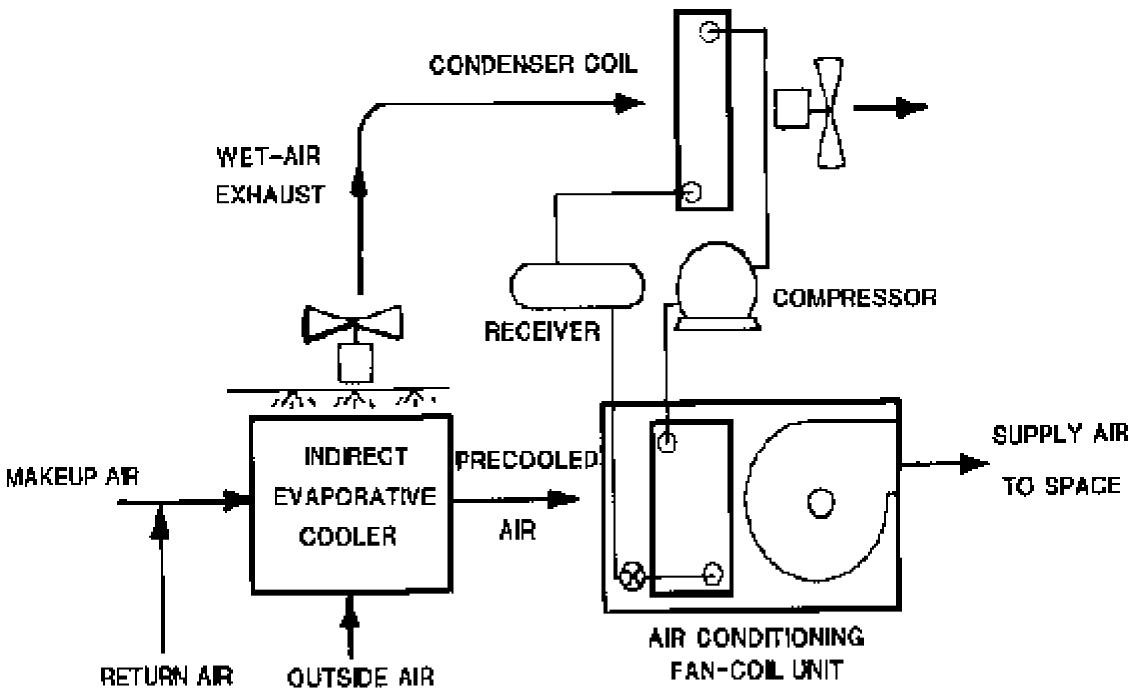
- prim: the primary energy factor set to 2.5 in the MEErP,
- SEERon refers to prEN14825 standard here. Please refer to the Task 1 report for a detailed description of the metrics.

For instance, for an air cooled electric chiller with a SEERon of 4, about 37 % of its cooling capacity is obtained from renewable energy (extracted from the air).

In addition to pure increase of the energy efficiency of present archetypes, a number of alternative solutions are being developed in order to reach higher shares of renewable energy in the cool energy supply.

Evaporative cooling hydrid solutions

These techniques are reserved to cooling machines that also ensure the ventilation, i.e. air conditioner integrated in air handling units. In addition of supplying cooler air at the air condenser, evaporative cooling may be used to decrease the dry bulb temperature of the air delivered to the conditioned space. If as explaind in task 1, evaporative cooling may not enable to reach sufficiently low temperature for air conditioning, for instance in humid climates, it may wel be combined with a standard air conditioner in order to reduce the compression work of the compressor. An example is shown below.



[15] SEERon refers to prEN14825 standard here. Please refer to the Task 1 report for a detailed description of the prEN14825:2010 project standard.

Figure 5 -32 . Example of a hybrid / indirect evaporative air conditioner, source (ASHRAE, 2008)

In the USA, where rooftop air conditioners represent a large market share, a number of prototypes have been tested (Reichmuth, 2007). As for the evaporative condensers presented above, this solution enables to decrease the peak power significantly and in addition supply lower temperature cool air to the indoor coil. This system is particularly well suited to hot and dry climates but can also supply energy efficiency gains for European climates apart Seville.

It should be noticed that air handling unit manufacturers offer the same type of functionality in Europe.

Solar photovoltaïcs plus electric driven machine

It consists in producing electricity with a solar photovoltaic panel in order to generate renewable electricity to supply the compressor. Some products are commercially available. However, the coupling remains theoretical as long as the feed-in-tariffs do not encourage the customer to consume locally the electricity produced, and the two systems may be considered separately, as is presently the case.

Motor driven compressors

Gas engine heat pump (GEHP) and air conditioners are available for more than 20 years. A combustion engine, generally using natural gas or propane, drives the mechanical compressor of the vapour compression cycle. With gas engine efficiency ranging between 30 and 45 % (Hepbasli, 2009), these systems may be competitive with traditional cooling/heating generators. Indeed, the 30 to 45 % gas engine efficiency is to be compared with the primary energy factor of 2.5 used in the MEErP methodology (that can be understood as a 40 % average efficiency conversion from fossil fuels to electricity).
They have the supplementary advantage that they may not only cool and heat but also supply sanitary hot water by recovering heat on the gas engine in winter time and in summer time.
They suffer however of high first cost which limits their market penetration. In addition, they are likely to be more competitive at higher capacities for bigger engines that may reach higher performances. Below 50 kWe, the efficiency is rather close to 30 % than 45 %[16]. But as the cooling/heating cycle is the same as for standard air conditioners, their potential for improvement, apart from the engine itself is similar. Integrating the externality of peak power from cooling/heating/hot water generators would help their development by compensating part of the higher investment cost.

Stirling engines are external combustion devices that offer high thermal efficiency (> 30%) at small sizes <1 kW. Free-piston Stirling engines with integrated alternating current linear alternators are commercially available for micro-cogeneration fired by natural gas. These motors could be coupled with solar thermal collectors in order to produce cooling from renewable heat.
The Stirling engine may be coupled with a linear compressor to avoid the losses to produce alternative current to run an alectric compressor. Theoretically, this may lead to very efficient cooling/heating generators. Nevertheless, the linear compressor technology was never commercialized. A prototype developed for residential air conditioning (Janssen, 2002) showed efficiency levels comparable to the ones of average reversible split air conditioners in Europe today, but the prototype had a smaller capacity than required. The theoretical potential for energy efficiency improvement is higher than for the vapor compression cycle but technical barriers remain. Despite several decades of research, the Stirling cycle has never been commercialized for air conditioning (IPCC/TEAP, 2005) and remains a BNAT for air conditioning.

Ab(ad)sorption machines

Absorption cooling machines have been developed in the past as air conditioners and chillers but only the chillers are still commercially available. Single effect absorption chillers enable to reach a GUE (gas utilization efficiency, ratio between the cooling capacity delivered and the gas input power[17]) of about 0.7, while double effect absorption machines may reach a GUE of 1.1, as shown on Figure 5 - 33 below. Low to medium capacity units are commercially available in Europe for single absorption

[16] Technical documentation Aisin GEHP.
[17] Defined in task 1.

machines working with the NH_3-H_2O fluids, typically lower than 100 kW. Regarding double effect chillers, working with LiBr-H_2O, they exhibit larger capacities, typically from 200 kW to several MW (Nunez, 2008).
The review of international standards[18] showed that in the USA, performance standards were set at about these levels and differenciated for single and double effect technologies. With the primary energy ratio of 2.5, double effect air cooled absorption machines may compete with air cooled chillers while electric water cooled chillers are much more efficient on total than double effect absorption machines. Triple effect absorption machines, could reach GUE above 1.5 but this technology is still under development and so is a BNAT for absorption chillers.

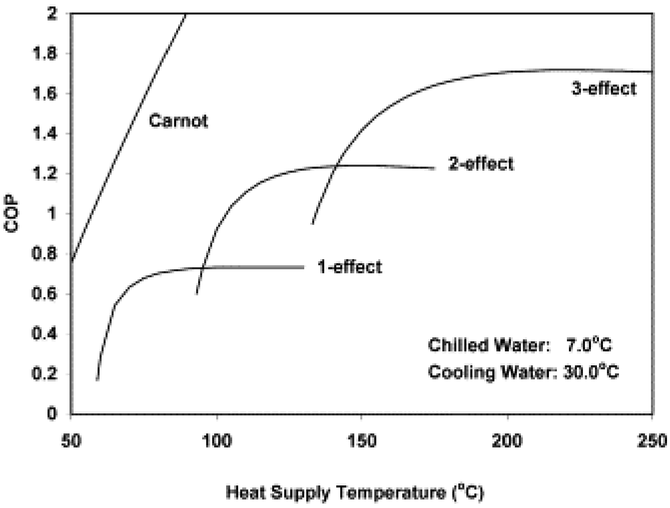


Figure 5 - 33 . Coefficient of Performance (COP) as a function of solar heat supply temperature for single-, double- and triple-effect LiBr–water absorption chillers, source (Grossman, 2002)

Giving the higher investment cost, > 250 €/kWcooling (without cooling tower) according to (Nunez, 2008), and lower total efficiency, absorption machines are normally reserved to niche applications like trigeneration, or to markets with very high electricity prices as compared to gas, as can be the case for highly constrained electricity grids in the summer. However, the market is rapidly expanding in Asia. “A new report from Global Industry Analysts (GIA) suggests the worldwide market for absorption chillers is forecast to reach 924.2 million USD by 2017. In contrast to the European and US markets where centrifugal and positive displacement chillers occupy a dominant position, absorption chillers drive demand in the Asian chiller markets, particularly in Japan, China and Korea, which account for a significant 75% of the global market. The high demand for absorption chillers in these regions is mainly attributed to the scarcity of fuel resources and poor electricity infrastructures that compelled various governments in Asia to promote usage of absorption chillers. Absorption chillers are primarily driven by waste heat and therefore their integration with heat recovery and power production units is now the mainstream option in many developed countries.”

Adsorption machines working with silica gel (solid sorbent) and water as refrigerant are commercially available in Japan. At the moment the GUE are about 0.65 and the costs higher than for absorption chillers, > 500 €/kW (without cooling tower) according to (Nunez, 2008). However, their margin for improvement may be higher as intensive research is still going on regarding adsorption (research on liquid desiccant and alternative solids to silica gel).

Heat driven technologies

Heat produced from thermal solar collectors or waste heat from industries or other processes may be used to generate cooling. This heat can be used to produce cooling from absorption and adsorption machines or cooling, with a stirling motor supplying electricity to a standard electric vapour compression cycle or with a directy coupled linear compressor (see above the part on Stirling cycle).

Alternatively, it can be used in a desiccant open cycle.

[18] See task 1, part 3 on legislation.

As explained in Task 1, desiccant evaporative cooling systems appear as an alternative to classical air-conditioners. The principle consists in drying the air in order to get a high potential of evaporative cooling of air. This technique is refrigerant free and uses few of electricity. In the other hand, as it is necessary to regenerate the desiccant wheel, thermal energy is required to heat up the wheel at temperatures in the range of 50–100 °C. An illustrating scheme is given below for a solar assisted desiccant system, which normally rather uses a water solar collector than an air collector.

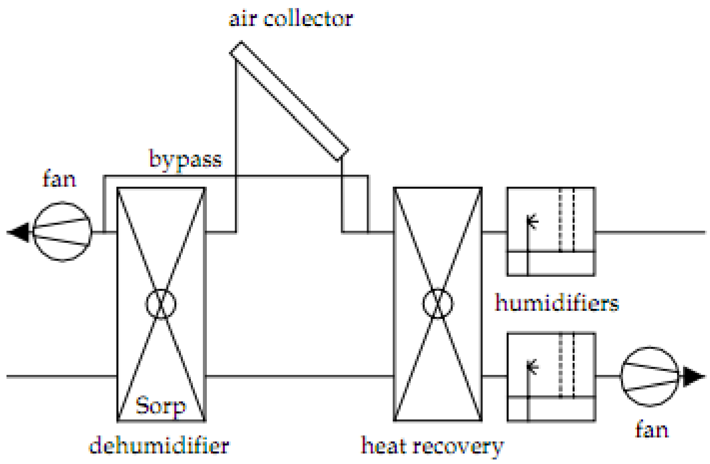


Figure 5 - 34 . Desiccant cooling/dehumidification system driven by a solar air collector, source Höfker

This principle is of limited application, as a cooling system by its own. However, it can be assisted by a mechanical cooling system to form a hybrid system, and thus may enable to increase the share of renewable cooling of air conditioning systems. It has also the ability to supply dehumification, which may reduce the cooling required at terminal units. As a counterpart, the pressure drop through the supplementary wheel may reach 200 Pa which cannot be neglected.
Robatherm sells air handling units built with desiccant systems. Daikin recently developed a VRF system with with a desiccant wheel (to be located in a AHU) which can be regenerated using the heat from the condenser of the cooling generator. Hence, dehumidified air is introduced into the buildings and latent loads are reduced.

Non available technologies

A number of alternative cooling principles do exist, amongst which:

- the magnetocaloric effect: magnetocaloric materials heat up in a magnetic field; the heat is extracted by a gas or liquid while the magnetic field is maintained ; then the material is insulated from the magnetic field, the dipoles divert from their previous direction extracting heat from the ambient to do so ; and cool down again when they are removed from it. It can then asorb the heat of a second fluid. Theoretically, magnetocaloric effect could become a competitor to vapour compression cycles, with theoretical efficiencies that are higher. But at the moment, the technology is not yet available.

- thermo-electric effect: this technology is based on the Peltier effect ; The Peltier effect is the presence of heat at an electrified junction of two different metals. When a current is made to flow through a junction made of materials A and B, heat is generated at the upper junction at T2, and absorbed at the lower junction at T1. The Peltier heat absorbed by the lower junction per unit time is proportional to the current multiplied by the coefficient ΠAB, which is the Peltier coefficient for the thermocouple composed of materials A and B and ΠA (ΠB) is the Peltier coefficient of material A (B). Π varies with the material's temperature and its specific composition. The principle is commonly used to cool electronics and in temperature sensors. But the scaling up to air conditioning appliances has not been a success until now.

- thermoacoustic refrigerators (figure below): “Thermoacoustic device consists, in essence, of a gas-filled tube containing a "stack" (top), a porous solid with many open channels through which the gas can pass. Resonating sound waves (created, for example, by a loudspeaker) force gas to move back and forth through openings in the stack. If the temperature gradient along the stack is modest (middle), gas shifted to one side (a) will be compressed and warmed so that a parcel of gas with dimensions that are roughly equal to the thermal penetration depth (δk) releases heat to the stack. When this same gas then shifts in the other direction (b), it expands and cools enough to absorb heat. Although an individual parcel carries heat just a small distance, the many parcels making up the gas form a "bucket brigade," which transfers heat from a cold region to a warm one and thus provides refrigeration."[19] This technology is still developing but shows greater promise than other alternatives.

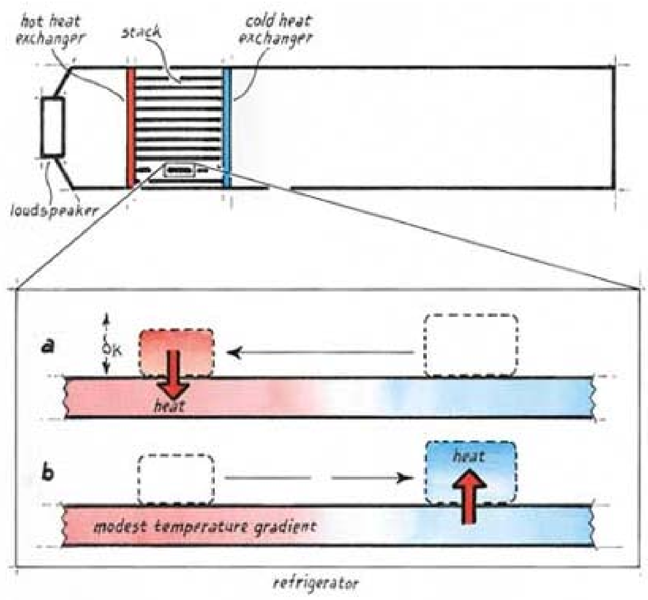


Figure 5 - 35 . Thermoacoustic refrigerator, source americanscientist

5.3. TERMINAL UNITS

The main type of terminal units identified in this study is the fan coil unit type. It has been shown in the task 4 report that their environmental impact was mainly due to their electric consumption. Energy efficiency is thus the main index to be used to reduce their environemental impact. It should be noticed that their aggregated impact is relatively low, the stock of fan coils in use is indeed estimated to consume about 2 TWh in Europe in 2010.

5.3.1. FAN COIL UNITS

Best available technologies

Best available technologies may enable to reach much higher energy efficiencies than available on the present markets. Amongst the technologies present in part 5.1, better fans, EC motors with variable speed drive and improved heat exchangers are the solutions to reach higher fan coil efficiencies. There may be also some lower potential regarding their low power mode consumption as controllers may consume 5 W or more continuously when equipment are on (e.g. at office working time all year long for reversible 2 pipe units).

The higher energy efficiency gains with the same product architecture can be hoped from the higher penetration of EC motors with VSD controllers and thus a part load efficiency metrics is clearly required in order to optimize the performances of fan coil units.

Energy efficiency metrics

[19] http://www.americanscientist.org

In order to establish the improvement potential of fan coils, it is necessary to crystallize the rules by which the products are evaluated. The study plans to adopt the efficiency characterization proposed by Eurovent (Eurovent, 2011) and reported below. Some points may require adaptation, which should be discussed in ad-hoc technical groups, this regards for instance:

- Part load weighting coefficients,
- Total equivalent numbers of hours,
- Low power mode inclusion or not, and if necessary with which equivalent hours …

The Eurovent proposal is reported below.

Definitions according to Eurovent Rating Standards 6/C/002 and 6/C/002A (Drafts 2011-04)

Scope

All Fan Coil Units (Ducted and Non Ducted) defined by Eurovent Rating Standards 6/C/002 and 6/C/002A (Drafts 2011-04).

• Non ducted units: Fan Coil Units with air flow lower than 0.7m3/s and a published external static duct pressure at 40 Pa maximum.
• Ducted units: Fan Coil Units up to 1m3/s airflow and 300 Pa available pressure.
A Fan Coil unit is a factory made assembly which provides the functions of cooling and/or heating air using chilled or hot water with air flow to the room ensured by one or more electrically driven fans. Fan Coil Units may be of the cabinet style, within a room, for free air delivery, or of the chassis style, concealed within the building structure with minimal ducting appropriately connected to the inlet and/or outlet of the unit.

The principal components are:
• one or more heat exchangers;
• one or more fans with electric motors;
• an appropriate enclosure;
• condensed water collecting facilities when cooling;
• air filter.

***Total Cooling Capacity**: Total heat energy removed from the air divided by the defined interval of time.*
***Sensible Cooling Capacity**: Sensible heat energy removed from the air divided by the defined interval of time.*
***Heating Capacity**: Total heat energy supplied to the air divided by the defined interval of time.*
***Fan Power Input**: Average electrical power input of the Fan Coil Unit within the defined interval of time.*
***Water Pressure Drop**: Difference between input and output water pressure.*
Sound Power: Total sound energy radiated by the Fan Coil Unit per unit time.
***A-weighted Sound Power**: A single figure on a specific scale which can be related to the subjective assessment of the loudness of a noise.*
***Air flow rate**: Volume air flow through the unit at standard conditions.*

FCEER and FCCOP

For each unit the participant shall select three speeds called high, medium and low speed. The Fan Coil Energy Efficiency Ratio (FCEER) and the Fan Coil Coefficient of Performance are defined as follows:

$$FCEER = \frac{5\% \cdot Pc_{high} + 30\% \cdot Pc_{med} + 65\% \cdot Pc_{low}}{5\% \cdot Pe(c)_{high} + 30\% \cdot Pe(c)_{med} + 65\% \cdot Pe(c)_{low}}$$

$$FCCOP = \frac{5\% \cdot Ph_{high} + 25\% \cdot Ph_{med} + 70\% \cdot Ph_{low}}{5\% \cdot Pe(h)_{high} + 25\% \cdot Pe(h)_{med} + 70\% \cdot Pe(h)_{low}}$$

With:

- *Pchigh, med, low,*
  total cooling capacity at high, medium and low speed respectively [kW];
- *Pe (c)high, med, low,*
  power input in cooling mode at high, medium and low speed respectively [kW];
- *Phhigh, med, low,*
  heating capacity at high, medium and low speed respectively [kW];
- *Pe (h) high, med, low,*
  power input in heating mode at high, medium and low speed respectively [kW].

Best available products

With this metrics, Eurovent developed the following energy efficiency classes and an associated label.

Energy Efficiency Classes in cooling and heating
A to G energy efficiency scale for Fan Coil units based on FCEER and FCCOP and defined as below.

- **Non-ducted Fan Coil units**

Class	Cooling mode	Heating mode
A	FCEER>=185	FCCOP>=265
B	185>FCEER>=120	265>FCCOP>=160
C	120>FCEER>=80	160>FCCOP>=100
D	80>FCEER>=55	100>FCCOP>=70
E	55>FCEER>=40	70>FCCOP>=50
F	40>FCEER>=30	50>FCCOP>=40
G	30>FCEER	40>FCCOP

- **Ducted fan Coil units**

Class	Cooling mode	Heating mode
A	FCEER>=85	FCCOP>=85
B	85>FCEER>=60	85>FCCOP>=60
C	60>FCEER>=40	60>FCCOP>=40
D	40>FCEER>=25	40>FCCOP>=25
E	25>FCEER>=15	25>FCCOP>=15
F	15>FCEER>=10	15>FCCOP>=10
G	10>FCEER	10>FCCOP

Eurovent evaluated the present products in their catalogue. This leads to the following product repartition by class.

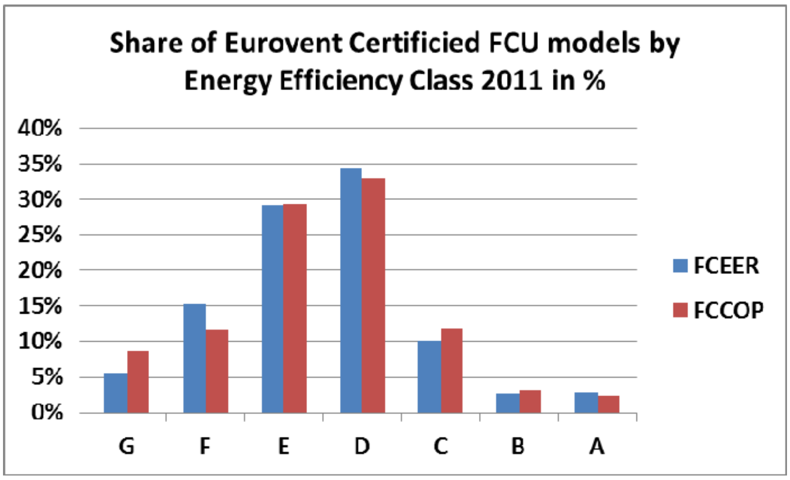


Figure 5 - 36 . Eurovent Certified FCU models by Energy Efficiency Class 2011, source Eurovent[20]

Based on these findings, Eurovent estimates that the products in class A and B are equipped with EC motors and as such they represent today less than 5 % of the listed models and consequently less than 5 % of the EU sales[21].

[20] Source: Eurovent Market Intelligence 2011, estimated 2010 volume in units sold for EU-27 plus Albania, Bosnia Herzegovina, Croatia, Macedonia, Montenegro, Norway, Serbia, Switzerland; Basis: reported unit sales of 26 major European suppliers
[21] Based on the estimate that the certified manufacturer sales represent about 75 % of the EU market and that the 25 % left have no access presently to the EC technology.

Such products may then decrease the energy consumption of fan coils by a factor of up to three as compared to standard class D model.

Higher efficiency than class A can be found on the market. For instance, best non ducted fan coils have declared cooling FCEER indices well above 200. Eurovent thus proposes to add supplementary classes above class A in the coming years. This shows an even higher potential for improvement and it seems that best available products could already enable to cut the consumption by a factor up to 4.

5.2.2. CHILLED BEAMS AND RADIATIVE COOLING

Chilled beams, either active or passive and other radiant cooling surfaces are seen as best available terminal unit technologies.

For active chilled beams, they require the use of a dedicated outdoor air system to ensure the central dehumidification of the air. The air is then mixed with the internal air with an induction rate of 1 to 8 and refreshed by the terminal coils in the beam. Chilled water temperature should not be below 14 °C in order to avoid condensation. With 3 % increase in efficiency by °C of chilled water temperature increase, this gives already close to 20 % efficiency gains. In addition, the system may enable the use of underground water in central and northern Europe in the summer to cool directly with natural water, i.e. without a chiller. When compared to typical fan coil unit systems, this requires central dehumidification in a air handling unit, generally with a dedicated coil on the water cooling system. This dehumidification thus occurs centrally instead of locally (fan coil units would deliver latent cooling capacity to ensure this dehumidification), and that should require less energy as room vapor emissions are not treated.
A comparison is done with a typical VAV system in the USA. It is there the more typical system for large buildings. The savings are computed to be more than 20 % upon a typical VAV system (ACEEE, 2009).

As the comparison with fan coil units not only depend on the product itself but on the treatment ensured for the air system, it makes the comparison complex and highly dependent on climate conditions and system operation. It is thus suggested not to attempt to rate these terminal units on a comparable basis with fan coils.

The same happens for passive chilled beams and other passive radiative cooling surface. For these beams and other radiative surfaces, the problem is the same but it should be added that their cooling capacity is limited and they cannot be used in all buildings. For a typical outdoor indoor condition at peak time of 35 – 24 or 11 K, the vailable cooling capacity is 50 W/m2 at most (ASHRAE, 2008). An efficient solution to overcome this, and this is also a fast developing solution, is to couple the radiative cooling solution with a displacement ventilation solution, or other distribution system in the room enabling to maintain a high level of stratification.

As these beams and radiative systems are developing, there is a need to characterize the efficiency of chillers for these temperature conditions. This could lead for instance to show performance characteristics of chillers at two or more chilled water temperature conditions, 7 °C for fan coils, 14 °C for active beams and 18 °C at least for radiative cooling terminals (as is planned in the French 2012 RT). This would be the parallel situation to the one of water based heat pumps in the latest version of the calculation method issued from ENER Lot 1.

5.4. HEAT REJECTION UNITS

According to stakeholders, dry coolers are majoritarily used in order to cool down the water temperature of new installed water cooled air conditioners and chillers, instead of open cooling towers in the past.

This is the base case situation for this study and this part focus on the potential offered to decrease the environnemental impact of dry coolers. The environmental impact of heat rejection units is estimated to be as low as 1 TWh for the stock products in use in 2010.

However, heat rejection has important consequences on the energy efficiency of the cooling generator it serves.

5.4.1. DRY COOLERS

Best available technologies and products

Best available technologies may enable to reach much higher energy efficiencies than base cases in this study.

Amongst the technologies presented in part 5.1, better fans, EC motors with variable speed drive for better part load performances and improved heat exchangers are the solutions to reach higher fan coil efficiencies. There may be also some lower potential regarding their low power mode consumption as controllers may consume 10 W or more continuously when equipment are on (e.g. at office working time all the cooling season).

The higher energy efficiency gains with the same product architecture can be hoped from the higher penetration of EC motors with VSD controllers and thus a part load efficiency metrics is clearly required in order to optimize the performances of fan coil units.

Energy efficiency metrics

The metric used in the USA to set requirements on the energy efficiency of heat rejection equipment is in gpm/hp, with 1 gpm (US) = 0.000158 m^3/h and 1 hp (US)=746 W. As inlet and outlet temperatures are set in the standard conditions, in fact, what the USA are using, are kWcooling / kWelec or the reverse of the efficiency of the units.

In Europe, we could then use, a EER metrics, by adopting the inverse of the ratio used in the USA, to remain coherent with already existing metrics.
EER = Pc / Pe(C)

This type of indicator was previously used by Eurovent with the index noted R, which can still be found in some manufacturers' technical brochures.

However, the energy efficiency gains which can be hoped from best available technologies are much more important at part load than at full load so that a seasonal performance metrics, as for fan coils should be preferred.

5.4.2. ALTERNATIVE HEAT REJECTION MEANS

Energy efficient heat rejection is first to obtain low temperature condensation

The more efficient process to extract the heat is clearly of the evaporative type, either an open cooling tower or an evaporatively cooled dry cooler or an open cooling tower. In that case indeed, the heat exchange benefits from the water vaporatisation and as such refers to the wet bulb temperature of the air, which is typically much lower than the dry bulb temperature of the air. As the chiller/air conditioner energy consumption is typically five times more important[22] than the one of the heat rejection unit, gaining 15 % on the air conditioner/chiller energy consumption gives higher gains than reducing the energy consumption of the heat rejection product by itself.

In the other hand, it appears that heat rejection units have non negligible consumptions as compared to the chillers with high potential for improvement, noticeably with the introduction of EC motors plus VSD control.

A common energy efficiency metrics for the different heat rejection types?

Thus, the question arises whether it is feasible to draw a single metrics for the different heat rejection unit types.

In fact, in order to reach the higher enenergy efficiency potential, the ideal solution would be design an energy efficiency metrics including not only the heat rejection or the air conditioner / chiiller, but the complete chilling plant with at least the heat rejection equipement and the chiller(s) and to compare the whole on a common load curve. Default efficiency and control scenarios could be used to compute the pumping energy of the condenser water loop when required.

In practice, however, the present situation is far from this optimum[23]. In a first step, it seems required to develop energy efficiency metrics for the different heat rejection types. In that direction, the USA can be used as example as they have developed standards and have already minimum requirements in the ASHRAE 90.1 for commercial buildings based upon these indices (see Task 1, part on the US legislation).

The study team thus suggests to adopt, at least temporarily, the USA approach, which is to rate separately heat rejection units, while including the different operating conditions representative of the different heat rejection types in the cooling generator standards (i.e. EN15218, EN14511 and prEN14825 standards).

[22] See Task 4, part 3.1
[23] Note that this statement is all theoretical as such experience is not available for this approach.

CONCLUSION

Best available technologies have been identified by product category, air conditioners, chillers, terminal unit and heat rejection units.

Air conditioners and chillers have still a large potential for improvement by using the best available technologies, which encompass better individual component like EC motors for fans, larger heat exchangers, better part load control and optimized part load designs. To maximize the potential benefits, the efficiency is to be judged on a seasonal performance standard, which is almost ready (prEN14825) for air conditioning products in this study. However, full load performance should not be forgotten as the externality linked to the management of the peak demand may be important in some EU countries. The potential alternative refrigerants have been screened. Regarding air conditioners, there is no perfect alternative for plsit and VRF air conditioners. Regarding chillers the choice is larger as they are operating as indirect systems. The identified alternative refrigerants will be considered as improvement options in the LCC analysis in Task 6.

Although they represent only a small part of the total energy consumption of air conditioning products, fan coils have a large potential for improvement which mainly coincides with the introduction of EC motors. The industry is proposing a seasonal based metrics which could help to foster the development of fan coils with EC motors and VSD controls.

Regarding heat rejection units, the gains that can be hoped first come from the choice of the heat rejection unit. The completion of the air conditioner/chiller standard should help the designers to make the best choices. In addition, and even if their own energy consumption is estimated to be low, heat rejection units may consume less energy by introducing more efficient EC motors and VSD control, as fan coils.

The overall potential for improvement could be much higher if comparing not only air cooled units amongst themselves but comparing the chilled water cooling plant as an extended product, together with its heat rejection means. This implies however a certain complexity which is thought to be at least a step forward as compared to the present situation.

Alternative cooling technologies have been described. Motor driven and heat driven absorption machines are available. They are of specific use, e.g. when waste heat is available or when supplying electricity ot the zone is too costly. The medium term development of solar and waste heat driven cooling system seems feasible. It may start with hybrid products, close to the present product architectures, with supplementary desiccant cooling or dehumidification. On the longer terms, thermo-acoustic and magnetic cooling still appear as possible alternative to the standard electric vapor compression cycles.

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LIST OF FIGURES

LIST OF TABLES

ACRONYMS

AC	1. Air Conditioning 2. Alternate Current
AHU	Air Handling Unit
CAV	Constant Air Volume
HVAC	Heating Ventilation and/or Air-Conditioning
IAQ	Indoor Air Quality
Pa	Pascal (SI-unit of pressure)
SFP	Specific Fan Power (in W per m³/s)
VAV	Variable Air Volume
VRF	Variable Refrigerant Flow
VSD	Variable Speed Drive
GEHP	Gas engine heat pump