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

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TECHNICAL ANALYSIS OF BEST AVAILABLE TECHNOLOGY

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6 TECHNICAL ANALYSIS OF BAT

Introduction

Following the MEEuP, this task *"entails a technical analysis not of current products on the market but on currently available technology, expected to be introduced at product level within 2-3 years. It provides part of the input for the identification of part of the improvement potential (task 7), i.e. the part that relates especially to the best available technology."*

Best available products

Since main players on the European market are international companies, we have to look for best available technologies not on the European market but on the best air conditioning markets, as Japan or USA, as shown in task 1. The diffusion of efficient appliances is not a problem of technology but more likely of market conditions and is to be treated as such within the following tasks. Thus, it makes little sense to separate available technologies inside and outside the EU.

In the USA, amongst highest efficient unitary air conditioners are Japanese reversible mini-split with US SEER of 21 (in SI units, 6.2) and US HSPF of 3.2[1]; some air conditioners rank higher in SEER (23 or 6,74) but the design conditions may be different (not forcedly reversible units, different indoor air flow rate, …).

So, for split units we should focus on most efficient Japanese material, small capacity reversible split units. Other types of air conditioning appliances, larger split units, window/wall appliances and single duct units have limited efficiencies mainly because of size and/or cost limitations. Best available products and technologies are used in the split segment; technology could be applied to all types of air to air conditioners, removing these technical or economical barriers.

Energy consumption and refrigerant as the main focus

It has been established in the previous tasks that the main contributor to the environmental impact of air conditioner products was energy consumption. Because of the particular importance given to GHG emissions, refrigerant is also of particular interest for these products. As a consequence, we will look for best available technologies keeping these targets in view.

Energy consumption is presented for the base case with the segmentation adopted previously in task 4 and 5. Not only the cooling and heating mode best technologies have to be assessed but also the "parasitic" mode ones.

Table 6-1: Average EU 27 total energy consumption per unit (task 4)

Energy consumption per unit / year KWh/y		Cooling only single split [3,5kW]	Reversible single split [3,5kW]	Cooling only single split [7,1kW]	Reversible single split [7,1kW]	Single duct [2,2 kW]
Cooling mode	Compressor on	377,5	374,0	882,6	839,7	311,1
	Thermostat off	23,2	18,1	42,5	33,2	47,0
	Stand-by	13,7	13,7	13,7	13,7	13,7
	Off mode	30,5	0,0	30,5	0,0	0,0

[1] Source: ARI directory, high sales models, November 2007.

	Compressor on + electric resistance		957,9		1997,5	
	Thermostat off		100,7		184,6	
	Stand-by		0		0	
Heating mode	**Crankcase heater**		34,9		81,4	
TOTAL per unit in kWh/y		445	1499	969	3150	372

In order to assess the potential to cut "parasitic" energy consumption, the different power mode functionalities of the air conditioners are reported in the following table. It is to be noticed that the preheating function have been separated to compute the yearly energy consumption.

Table 6-2: Air conditioners, power modes and functions of reversible air conditioners

	Function	**Current supplied to**
Thermostat on	Cooling Heating Dehumidification Humidification Ventilation (single duct) LCD display and lighting Functions activation by remocon Condensate evacuation	Compressor Fans PCB (outdoor and indoor units) LCD screen (single duct) LED (sensor for remocon) Drain pump
Thermostat off	Mixing indoor air to measure indoor air temperature LCD display and lighting Oil heater Functions activation by remote control Interface Condensate evacuation	Indoor fan, possibly at low speed PCB (outdoor and indoor units) Crk case resistance or stator coil LED (sensor for remocon) LCD screen (single duct) Drain pump[2]
Standby mode	Screen display and lighting (single duct and some split) Crankcase heater Reactivation function, timer clock, status indications	PCB (outdoor and indoor units) LCD screen (single duct) LED (sensor for remocon) Crk case resistance or stator coil
Indoor unit hard switched off	Crankcase heater	PCB (outdoor and/or indoor units) Crk case resistance or stator coil

[2] The **condensate pump** evacuation can be or not part of the product. Whether it is included, it is accounted in the performance of the unit in T1 conditions and should be on only part of the time when the compressor is running.

The proposed part load EU[3] standard does not include standby and off mode consumption. In the case of split units, the power may be supplied by outdoor unit or indoor unit. In general, power is supplied in single or 3 phase current to outdoor unit. Hence, when the end-user uses the hard switch off of the indoor unit, the outdoor unit may still consume energy. For reversible units with a crankcase heater, the PCB remains on in order to supply power to the crankcase heater. This function may explain why there is little difference between standby and off mode. For cooling only split air conditioners, there should be no remaining consumption, but it is not what is observed from Australian data presented in task 3.

6.1 Thermal comfort indoors

The improvement potential is to be studied from the point of view of the product that cannot be summarized only in terms of energy efficiency. Its controls and more generally the system interactions may largely affect the heating and cooling requirements.

It was supposed for the base case computation that a timer was available to the end user; in other cases, this could be supplied also as a separate part since most models do not enable to program the set point according to a weekly or more sophisticated schedules as boilers do. The computation of heating needs shows that implementing a heating set back corresponds to a massive cut always larger than 22 % in shops (set back at 12 °C) and 28 % in residences (set back at 15 °C); average values suggest a 50 % heating energy cut.

Another simple option would be to control the set point in cooling mode following the “adaptive law” formulated in the EU comfort standard (CEN, 2006) and that specifies a set point indoor variable with outdoor air temperature. It has not been found an air conditioner proposing this type of option yet despite several brands propose also sophisticated control like restricting the cooled area to people and not serving the whole room, with the same idea of cutting the needs to consume less.

Through the wall air conditioners in the USA and some window units in Europe are typically able to use cold air from outdoor in order to free cool when it is possible. This type of free cooling option is today available on larger units for instance on VRV systems but rarely primary air introduction is controlled by the cooling equipment. In task 4, the load calculation focused on buildings where openings were not available to the end users, as it is typically the case in cities and in the tertiary sector. The single duct makes use of this free cooling effect but as a counter part introduces hot air in the warm season. Split air conditioners able to use fresh air in order to free cool inside air and thus economizing compressor power consumption could help dividing cooling energy consumption by a factor 2. Barriers are on the cost of the installation but also on the air treatment part that can be necessary to introduce polluted fresh air and noise conduction from outside to inside.

A last point of importance is that increasing energy efficiency can be done while maintaining the evaporating temperature low enough in order to ensure dehumidification capability is maintained or without this constraint, the first option is typical in the USA and the second one in Japan. For Japanese best available products, dehumidification now appears as a separate option. Since dehumidification requires lower evaporating temperatures than necessary for sensible cooling and that consequently cooling and dehumidifying at the same time leads to less efficient cooling performance, manufacturers now propose intermittent operation in a dehumidifying mode operating with very low evaporating temperature for a short time. This can be done thanks to the addition of a second and dedicated expansion valve. It means that the air conditioning load is decreased of its latent part.

[3] Revision of the CEN TS14825 standard to be released for enquiry in 2008.

6.2 Cooling and heating functions

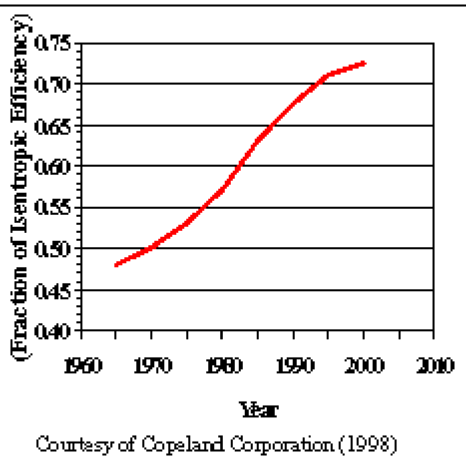
In that part, options to reduce the energy consumption of the units for hours when the compressor is operating (even only at par load) are investigated.

6.2.1 Compressor

Compressor efficiency at rated capacity

The first place to look at for energy efficiency gains is the compressor. Present compressor efficiencies have been identified in the task 4 report and range from 2.8 to 3.2 (ARI rating conditions for compressors) for rotary vane compressors and up to 3.2 for scroll compressors. Hence, the replacement of a rotary vane compressor by a scroll one may be considered as an efficiency option for smaller capacity units (ECCJ, 2006). For scroll compressors, perspective of improvement are limited at rating conditions (Figure 6-1).

Figure 6-1: Compressor efficiency limit, (DOE, 2001)



Concerning rotary compressors, progresses have been made to reduce friction losses. The Swing® compressor of Daikin in which the roller is solidarized with the blade; this avoids leakages between high and low pressure sides and for this compressor the global efficiency is 15 % higher, but this figure includes also an improved DC motor. Other manufacturers (Toshiba, Fujitsu, Sanyo …) have adopted a rotary compressor with two stages or twin rotary (Bensafi, 2006) that enables to improve the compressor efficiency by 10 % according to (Bensafi, 2006).

(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 74.4 %, already competitive with larger and most efficient scroll compressors.

Very high efficiency DC motors and scroll optimized compressors are the best available technology for the compression process (ECCJ, 2006), even for 2.8 and 3.5 kW cooling units. Despite, best performing split units in Japan (JRAIA, 2006) on the [0-12 kW] capacity range use rotary compressors, scroll units being reserved to higher capacity units.

Compressor motors

Conventional permanent split capacitor (PSC) motors operate at typically 60 percent efficiency (DOE, 2001). Small capacity motor efficiency levels presented in lot 11 are reported below.

Figure 6-2: Efficiency of motors according to lot 11

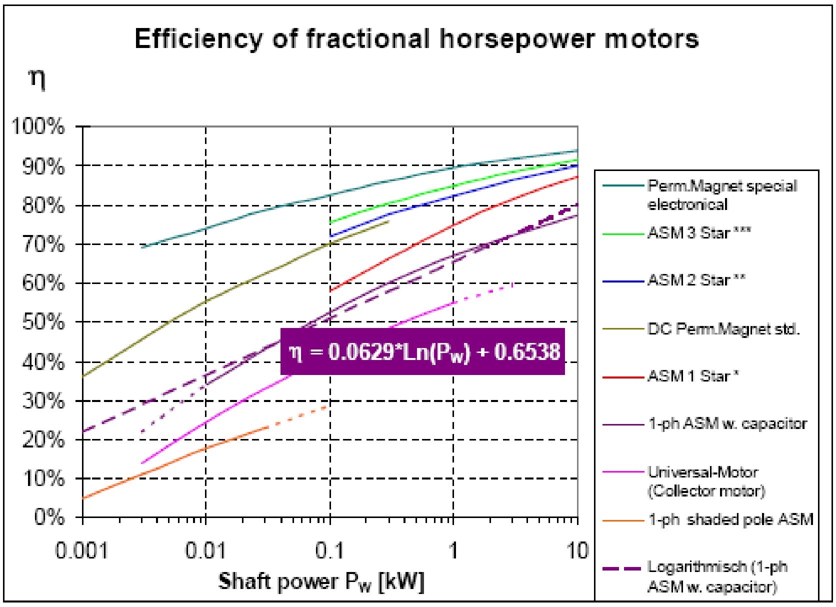
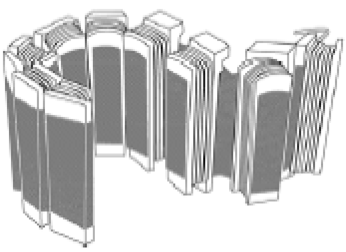


Figure 138: Efficiency of fractional horsepower motors [Nipkow, 2007]

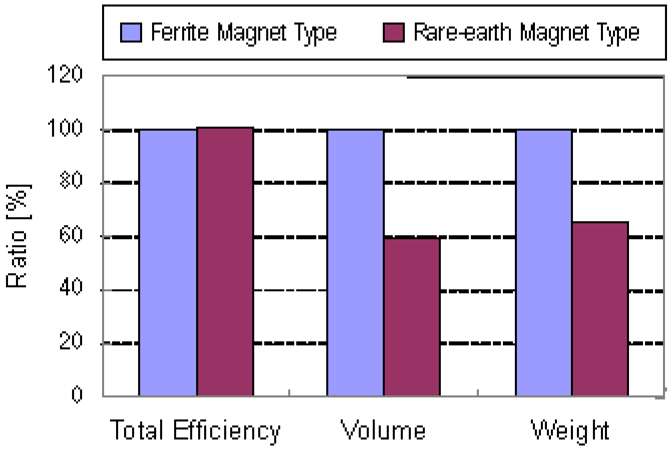
Standard direct current motors (DC motors) enable to reach 80 % efficiency. (ECCJ, 2006) notices supplementary efforts to increase the motor efficiency, either by replacing the ferrite of conventional motors by neodymium magnet that has higher magnetic flux density, or by reduction of the copper losses with improved coil winding geometry (Figure 6-3), by limitation of iron losses with the adoption of thin laminated steel sheets. Special magnet DC motors can reach efficiency levels up to 90 % in the 1 kW power range and slightly more for 2 kW. Specific efforts by Japanese manufacturers proved it was possible to reach 95 % motor efficiency in the 1 kW power range (ECCJ, 2006).

Figure 6-3: Reduction of copper losses by change in the winding geometry



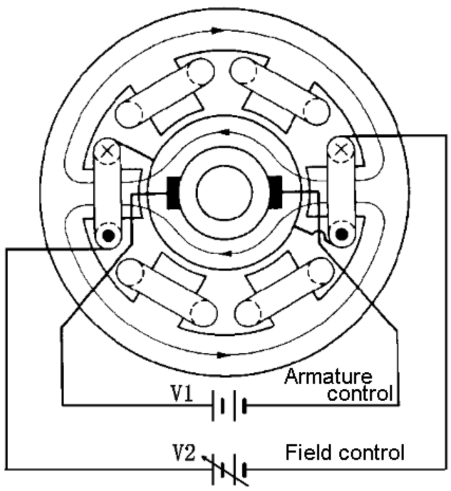
The use of neodymium enables to decrease significantly the volume of the DC motors for equivalent function and efficiency, as reported by (Mitsubishi Electric, 2006).

Figure 6-4: Reduction of material for motors using rare-earth materials (Mitsubishi Electric, 2006)



DC motors can also be improved by shifting to a 6-pole motor. Not only efficiency can be increased but also noise and volume can be reduced. The motor volume can be reduced by 30% and the motor losses can be reduced by 20% as compared to a typical 4 pole DC motor (Daikin).

Figure 6-5: 6 pole DC motor scheme



Recent developments for DC motors with variable speed include the improvement of the form of the wave signal of the current delivered to the motor with a shift from square signals to sine wave current forms (ECCJ, 2006).

Capacity staged and variable speed compressors

Following the load calculation results of task 4, the adoption of a technology that enables to adapt the capacity of the unit to the required cooling load can make a large difference in terms of seasonal performance but also at rated conditions. Seasonal performances are increased because the performance improves at reduced refrigerant flow rates as opposed to a on-off unit that cycles on and off. Also, inverter driven units let the freedom to manufacturers to declare more or less capacity and less or more EER. What really occurs on field depends on how the unit is controlled. In cooling mode, it is generally stated that inverter units will operate only a very limited time at high speeds, in a “booster mode” and will then get back to frequencies closer from the rated frequency.
Important progresses have been made in terms of efficiency of the compressor itself when operated at variable frequency. The adoption of high efficiency DC motors enabled significant gains at rated speed and above all at reduced speed. As a consequence, the range of operation of inverter controlled

compressor has been increased greatly and efficiency is maintained or even improved as compared to full load performances (JARN, 2003).
DC inverter rotary compressors can now operate between 8 and 120 Hz with peak efficiency at 30 Hz, while scroll compressors may be operated between 40 Hz and 180 Hz. Small units with inverter compressors operating at low frequency represent a challenge to standardization whose present test capability are limited to 1 kW cooling capacity measurements with acceptable measurement uncertainties.

The possibility of high speed operation in heating mode enables to maintain the capacity until –15 °C (Daikin, 2007) by increasing the frequency when the outdoor air temperature decreases. This is a definitive advantage for a seasonal performance index rating, as in field operations. Efficiency may be low but still higher than with resistive heating added.

Compressor performance improvement perspectives

Linear compressors are now available for refrigeration, in small sizes (LG). Developments of linear compressors for air conditioners were said to be able to further decrease motor losses and increase compression efficiencies, in particular at reduced frequency. With present DC motor compressor efficiencies, it seems difficult to improve further motor efficiencies. Moreover, no linear compressor is presently commercially available for air conditioners.

Despite centrifugal compressor size is on the decrease for several years, capacities are still far to reach power levels required for air conditioners. Other developments on oil free compressors and magnetic bearings could allow further compressor efficiency increase in the future (Clodic, 2008).

6.2.2 Heat exchangers

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 that can be reduced by improving the heat exchanger performance. The compressor will then operate more efficiently than at rated conditions.

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.

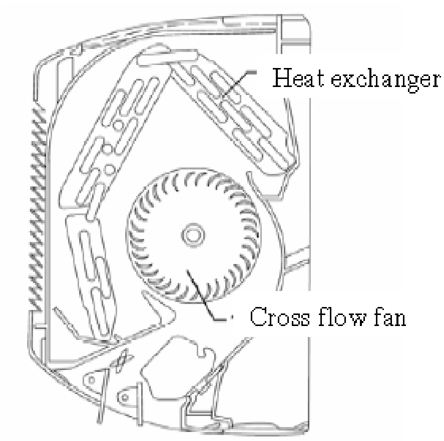
(Perrotin, 2004) figures out the performance increase with larger heat exchangers for a standard 2.8 kW unit similar to our base case split unit. The improvement is studied at constant cooling capacity, by downsizing (or reducing the frequency) of the compressor. For an 80 % increase in the heat exchange 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. These figures will be adapted for the base cases in task 7 by taking into account fan power corrections and isentropic efficiency variation with compression ratio. Nevertheless, the analysis shows that for the split unit tested, the main difference with an average unit was made by the larger heat exchangers, highly efficient fans and electronic control.

There are limitations to the possibility offered by this technique. Manufacturers generally use from 2 to 5 cabinet sizes for outdoor units in the 0-12 kW range. Indoor units are generally of a single size for wall mounted units in the 0-4 kW range while for cassette type indoor units, the size may be constrained by the size of ceiling panels it will be inserted into.

Too large heat exchanger surface area may necessitate to change the outdoor and/or indoor cabinet size with significant manufacturing overcost. This explains that smaller units in the same product range (for instance 2.8, 3.5 and 4 kW have often the same cabinet sizes indoor and outdoor) may have higher efficiency levels. 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.

A typical high efficiency indoor unit is presented on Figure 6-6. 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.

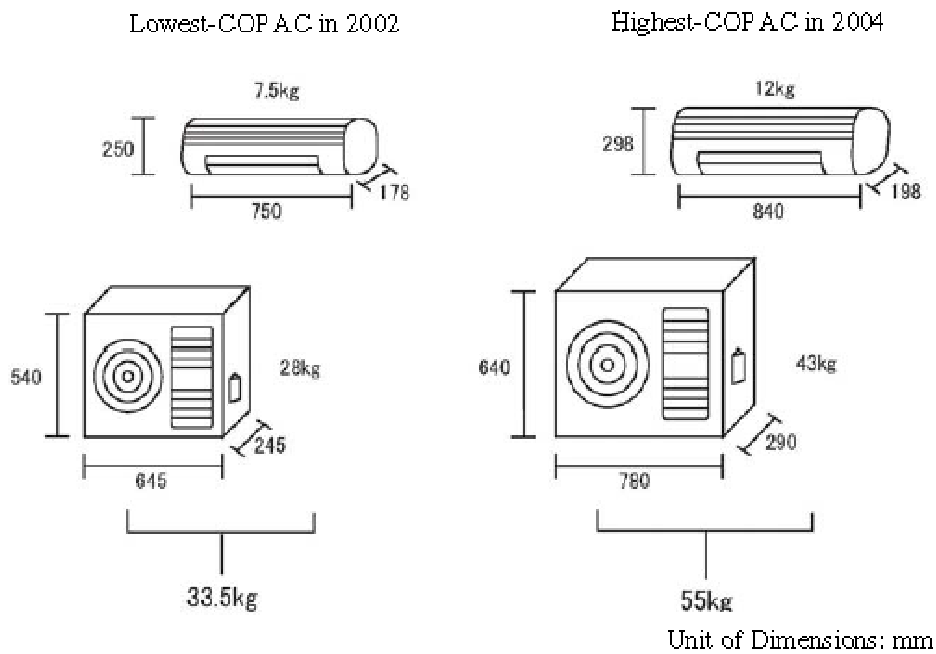
Figure 6-6 – 2.8 kW high efficiency indoor unit (ECCJ, 2006)



The figure below 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 %. 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.

Figure 6-7: 2.8 kW high units dimensions and mass evolution (ECCJ, 2006)

Dimensions and Mass of Highest-COP and Lowest-COP Air Conditioners



Given the energy efficiency increase associated to heat exchanger surface area (Perrotin, 2004), the heat exchanger area increase could by itself explain more than the half of the increase in efficiency.

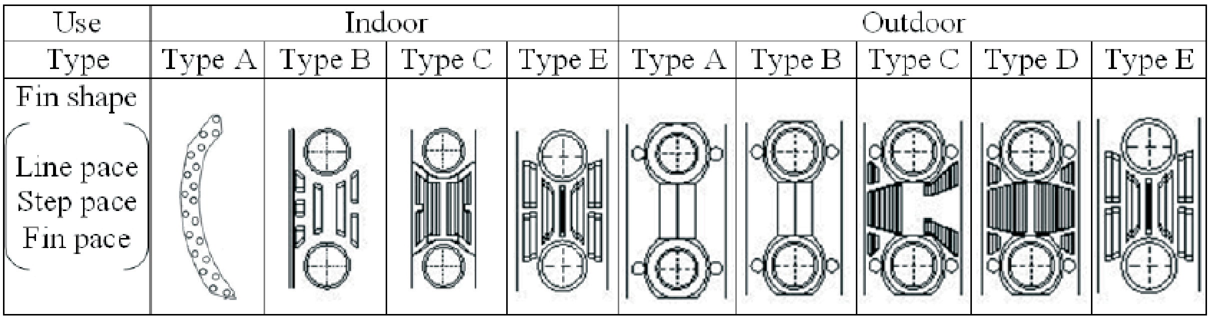
The heat exchanger area can also be enhanced by increasing the fin density. For reversible units, there is a trade-off between increased performance in cooling mode and decrease in performance in frost conditions in the heating mode. Also in cooling mode, too high fin densities may lead to premature 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.

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.

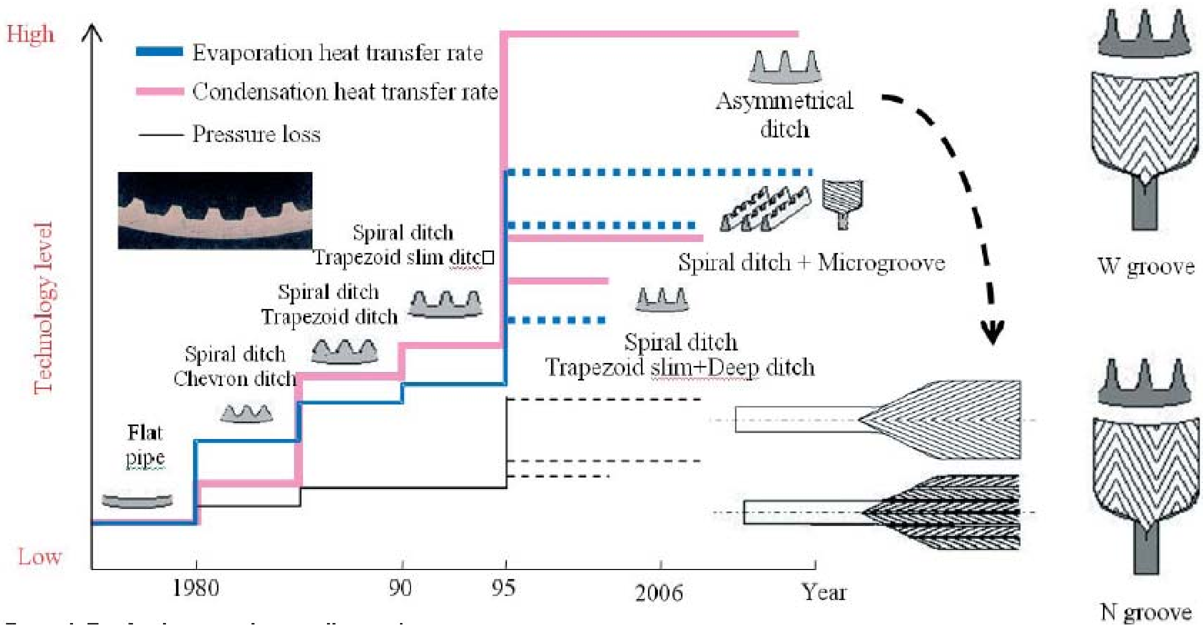
Evolution in the fin pattern was already presented in task 4. Heat transfer rates increase from smooth to louvered, and interrupted surfaces. Evaluation of potential gains on the air conditioner performance suggest that the efficiency could be improved by about 10 % when using slit fins instead of plain fins (Perrotin, 2006). 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.

Figure 6-8: Evolution of the fin pattern (Daikin, 2007)



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). 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 6-9: Evolution of inner refrigerant copper tube design (Daikin, 2007)

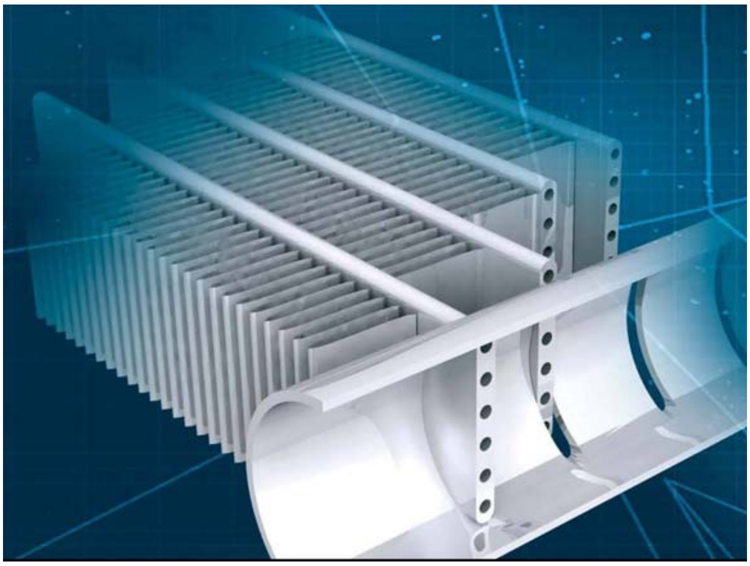


Microchannel heat exchangers

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. One of the problem for the adaptation of microchannel heat exchangers to reversible split products is that it prevents condensate coil drainage and consequently may be a concern for the heat pump evaporator.

Figure 6-10: 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.

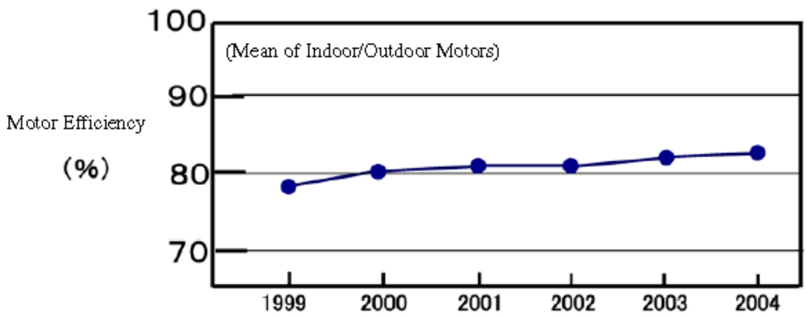
Evaporatively cooled condensers

Spraying water on the outdoor air / refrigerant heat exchanger enables to increase the performance of the heat exchanger in not too humid climates. The efficiency of this option is much dependent on the outdoor air temperature and humidity content. This is an option already available for single duct air conditioners on the EU market (Delonghi, 2007). Water consumption is of 2 L/h for a 2.5 kW unit (Delonghi, 2007). Data are missing for a more complete evaluation of the efficiency of the improved unit. In absence of a complete set of data, it seems difficult to evaluate this option.
(Hu, 2005) improves by 10 % a 3.1 residential split air conditioners by including a cooling tower and a water to air heat exchanger. Efficiency can be improved by 10 % at peak condition for Taiwan design wet bulb conditions. Water consumption needed to save 1 kWh is about 3 m^3/h and is of the order of magnitude of what is needed to produce 1 kWh with a standard coal plant.

6.2.3 Fans

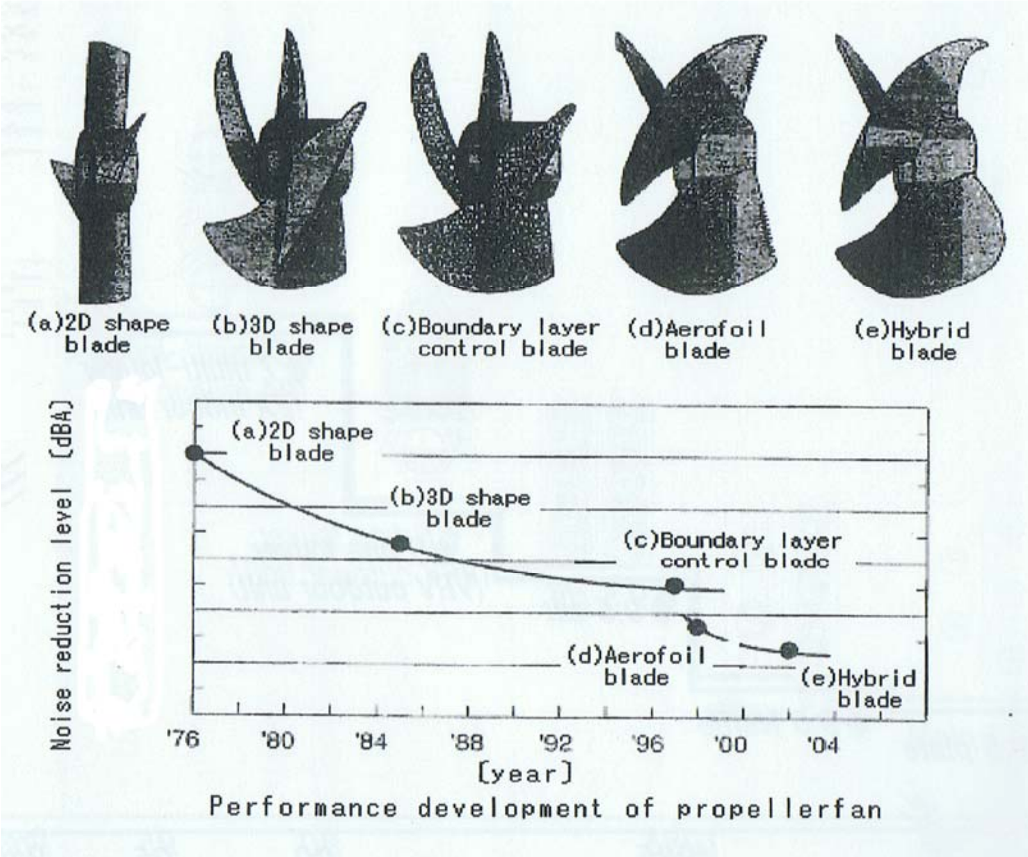
Fan motors and drive
The same progress observed for compressor motors can be applied to fan motors. Because of lower power ranges, efficiency ranges are lower. Following lot 11 references, previously reported on Figure 6-2, for a 100 W motor, standard motor efficiency would be of 18 %, 55 % for a DC motor and until 75 % for a DC fan motor with advanced features as 6 or 8 poles instead of 4, or rare-earth magnet types. For a 20 W fan motors, figures would respectively be 10 %, 40 % and 70 %. Best available technology is reported by (ECCJ, 2006) to reach more than 80 % motor efficiency as motors for compressors.

Figure 6-11: Air conditioner fan motors best available efficiencies (ECCJ, 2006)



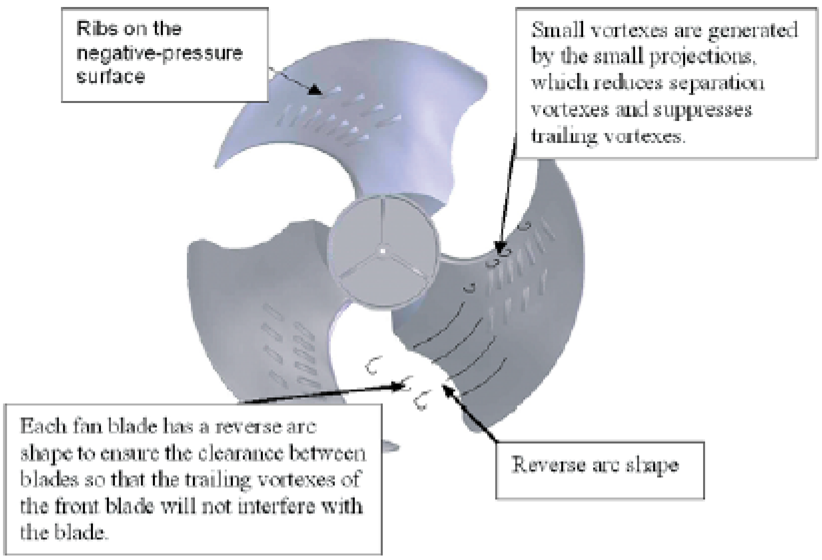
Axial or propeller fans
Propeller fans have known important evolutions, previously made of processed metals, it is now made of plastics (ECCJ, 2006). The shape has evolved progressively to increase the volumetric efficiency and decrease the noise level. The figure below presents some of the evolutions as reported by (Daikin, 2006).

Figure 6-12: Evolution of axial fan shape in air conditioner condensers (Daikin, 2006)



Further developments have been reported recently for operation in frost conditions, and consequently higher static pressure conditions when the coil is frozen.

Figure 6-13: High static pressure axial fan shape in air conditioner condensers (Daikin, 2007)

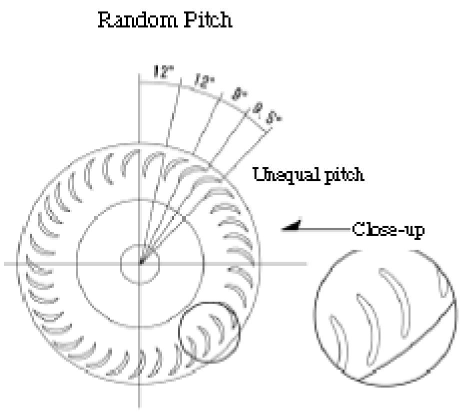


Cross flow fans

Mechanical efficiency tends to be half the one of axial fans in present split air conditioners.

(ECCJ, 2006) "Although a cross flow fan was composed of blades that were processed metal sheets in the past, an attempt to increase air volume has been made through introduction of plastic blades having a wing-shaped section and growing size of fan diameter, while controlling noise. The layout and molding of a fan and blades have also been improved, by having random spacing between blades, angling a fan shaft, etc."

Figure 6-14: Evolution of cross flow flan blade design (ECCJ, 2006)



Centrifugal fans

From forward blades with a lateral fix with a 40 % mechanical efficiency centrifugal fans can be improved from 40 % to 60 % with central fix rotors.

Reduced pressure losses

There is a competition between the increase in heat exchanger performance with higher compacity and consequently higher air side pressure drops and fan power consumption.

6.2.4 Expansion devices

In standard equipment, the expansion between condenser and evaporator is made by incurring a pressure loss by a diameter restriction of the piping, either by an orifice of constant diameter or a capillary tube of a given length. These cheap solutions enable to control the superheat at a given value for design conditions. Because the superheat is uncontrolled outside the design point, it is necessary to design the product for relatively high superheat values (typically 5 to 10 K) in order to avoid harmful liquid suction at the compressor inlet.

As explained in task 4, several steps can improve the performance of the expansion device. The largest part of the gains occurs outside the design conditions.

Thermostatic expansion valve (TXV) enables to achieve almost constant superheat whatever the operating conditions may be. Design superheat can be reduced to 4 to 7 K. The superheat is controlled by the equilibrium of 3 pressures, the condenser side pressure, the evaporating pressure and the pressure related to the superheat temperature. This may lead to little variations around the set point depending on the evaporating pressure. In addition, at very low load, there are stability problems that may lead to hunting of the valve and important energy losses. In addition, TXV enables to block the refrigerant migration when cycling and thus gain at part load. Cd coefficient decreases from typical 0.2 to 0.14 or lower (Dougherty, 2002).

Further gains can be obtained with an electronic expansion valve. Design superheat can be as low as 2 K and C_D coefficient decreases to 0.10 or lower.

6.2.5 Noise

In Japan, as in the EU, the customer is very sensitive to noise, outdoor as indoor. The simplest way to reduce noise is to reduce the air flow speed at the heat exchangers, but that comes with an important energy efficiency loss because of the low air flow rates generally observed for these units. Best available products also get very low noise levels as low as 45 to 47 dBA pressure levels outdoor and lower than 40 at full speed indoors, up to 21 dBA at lower speeds. The efficiency of the product is not declared by manufacturers at lower fan speeds despite the end-user may prefer the quiet mode operation at use time. The speed fan reduction may decrease the efficiency of the unit by as much as 15 to 30 %.

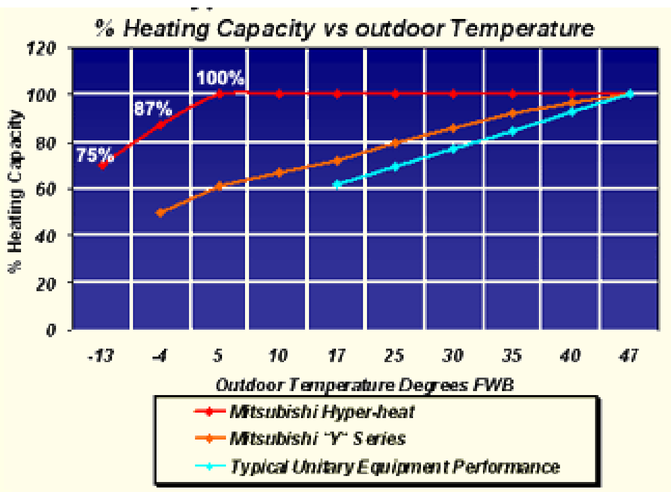
A number of technological evolutions have occurred in order to reduce the noise of air conditioners. For fans, larger diameters and lower speeds are used in addition to specific fan shapes, previously mentioned in the fan improvement part. The optimization of air flow path and pressure loss minimization also helps in reducing the noise level. The transition to DC motors and the use of variable speed compressors also enable a gain in noise levels, that is however difficult to evaluate because of the absence of published data on this subject.

6.2.6 Improvement of the thermodynamic cycle

Improved operation at low outdoor air temperature

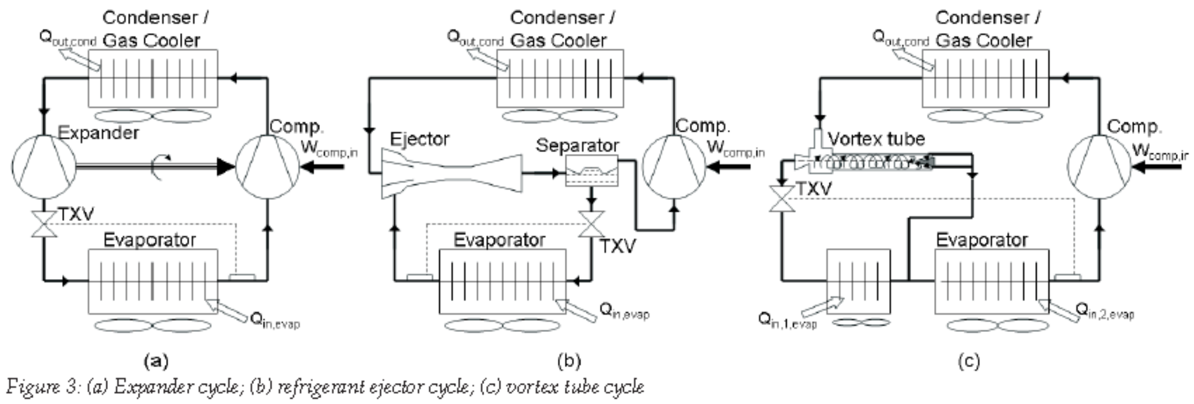
Cycle improvements are envisaged to improve the performance of air to air heat pumps in the heating mode at low ambient. In the range of air based heat pumps, an injection of vapor at an intermediate suction port in the compressor may enable to supply more heating capacity at lower ambient with a slight COP increase in these conditions (Ding, 2004). Mitsubishi proposes an air based split heat pump with the following advertised characteristics (Figure 6-19) thanks to a combination of inverter control (higher frequency at lower ambient temperature) and scroll vapor injection as described above. Capacity can be maintained until –15 °C with a COP increase likely to lie between 2 % at 2 °C and 15 % below – 7 °C (Ding, 2004) as compared to operations at low indoor air temperature and high frequency.

Figure 6-15: Variation of heating capacity with outdoor air temperature (°F), (Mitsubishi, 2007)



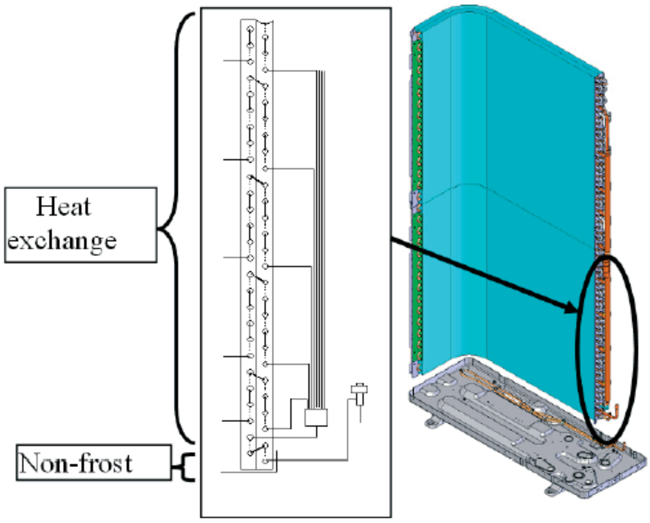
(Cremaschi, 2007) reports other technologies that can help to improve the performances of heat pumps at low outdoor air temperature. Expanders can improve the cycle performance by as much as 30 %. Ejector that use the flash kinetic energy to lower the compressor lift are also a promising technology. These cycles are still at the prototype level.

Figure 6-16: Heat pump cycle improvement for low ambiant, (Cremaschi, 2007)



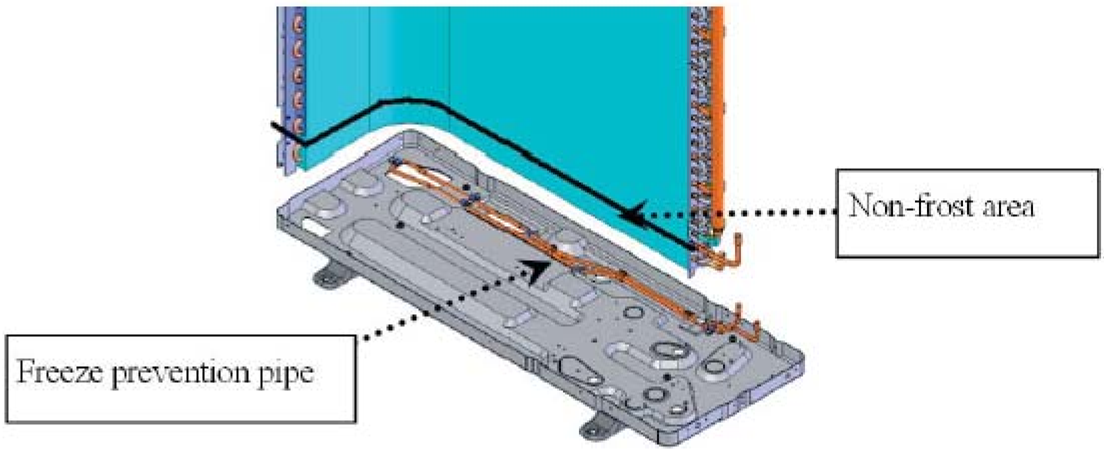
(Daikin, 2007) depicts how the outdoor air coil of VRV systems is improved to gain efficiency in frost conditions and at low outdoor air temperature. “The heat exchanger employed in this air-conditioner type has a frontal area about 15 % larger than the normal type to ensure heat exchange performance at a low temperature. The heat exchanger tubing is modified in order lower tubes serve as a protection against frost for colder temperatures. The heat exchanger has larger refrigerant paths than in the normal type in this class, in order to reduce the pressure loss during heating operation, as well as to ensure a no-frost area at the bottom where temperature is controlled in order to prevent frost formation during heating operation.”

Figure 6-17: Modification of the outdoor piping to increase performances in frost conditions, (Daikin, 2007)



In addition, water removed by defrost operation is likely to freeze at the bottom of the unit. This is likely to ease the frosting of the coil. (Daikin, 2007) also proposes a solution to this problem.

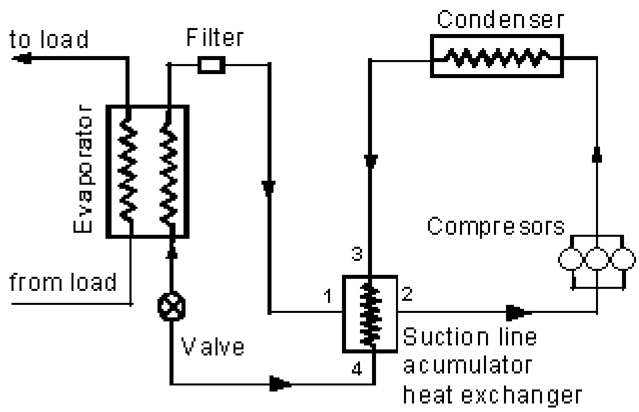
Figure 6-18: Anti-freezing outdoor unit bottom pipe, (Daikin, 2007)



Other cycles improvement

Internal heat recovery by liquid (subcooled refrigerant) and vapor suction heat exchanger may be an option to improve the cycle performance at low temperature (Klein, 2000). However, it would be necessary to short cut the heat exchanger at higher outdoor air temperatures and in cooling mode in order not to loose any gains or even decrease the global performance of the air conditioner.

Figure 6-19: Scheme of a modified cycle including liquid vapor heat exchanger (Klein, 2000)



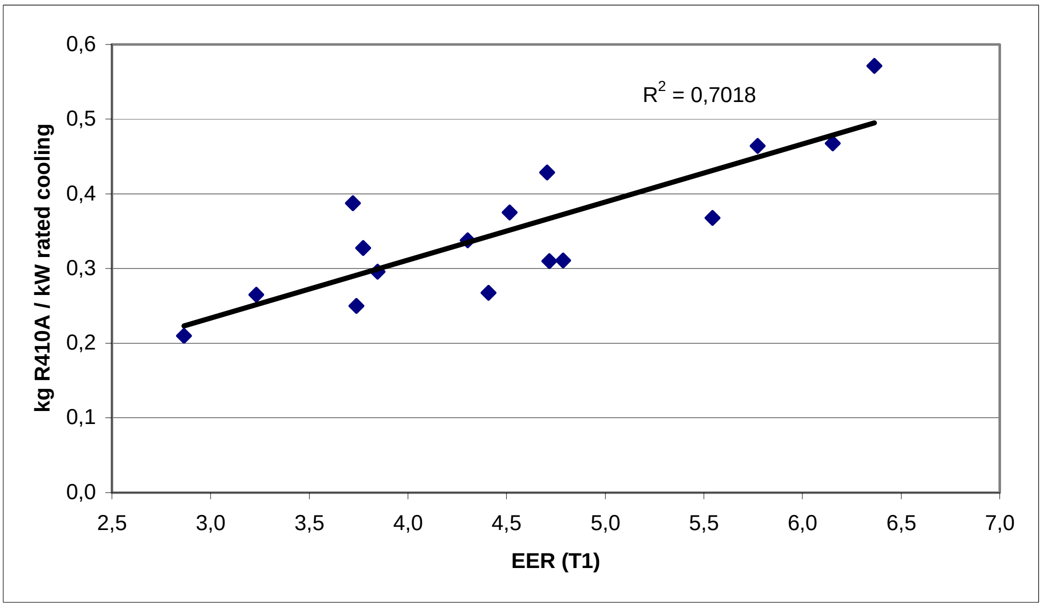
As for the expansion turbine, it makes it an improbable option for all fluids except for CO2 where either one of the two options could be used to improve the basic cycle efficiency.

6.2.7 Refrigerant fluid

Concerning **refrigerant**, more efficient air conditioners, in Europe as in Japan, have moved to R410A that offers a significant gain in compactness and thus potential to increase the heat exchanger area and consequently the efficiency.

R410A is taking the lead nowadays for air to air appliances in all main air conditioning markets. In task 4, a standard value of 0,31 kg/kW of cooling capacity was identified for R410A split air conditioners. The graph below presents the ratio of kg of R410A per kW cooling at rated conditions following data supplied by JRAIA. The relative increase of the heat exchanger size for high efficiency models as compared to the rated capacity of the unit has the direct consequence of the increase of the refrigerant mass per kW cooling. This graph suggests, despite important dispersion, that doubling the efficiency is also doubling the refrigerant charge of the unit.

Figure 6-20: Relationship between efficiency and refrigerant charge per kW cooling, source (JRAIA, 2006)



The graph suggests also there is an important dispersion. Refrigerant charge decrease of 40 % at lower efficiencies and 15 % at higher efficiencies may be reached with present available technologies; the set of available data is too restricted to extract more precise figures.

As explained before, in the section on heat exchangers, the reduction of the tube diameters of standard coil technology, or the transition to micro-channel technology can help to limit the refrigerant charge of the products. Micro-channel heat exchangers have the potential to reduce the refrigerant charge as far as 20 to 40 %.

Some manufacturers propose as an option on their product a sensor to detect potential leakage.

Alternative refrigerants

Alternative known refrigerant fluids are propane, ammonia, R32, and CO2. Ammonia (high toxicity), R32 and propane (flammable) present a risk that at the moment prevents their use in split air conditioners in Europe.
Propane is used in small capacity single duct air conditioners. For propane to be an option for split units, according to (UNEP, 2001), split systems should then be replaced by compact indirect water based systems that may not be as efficient as air to air conditioners. A TEWI analysis of the implementation of propane as a refrigerant is proposed in part 6.6 for single ducts.

Refrigerant fluids are being developed for the mobile air conditioning (air conditioning for cars) that aim to reach a GWP lower than 150 with a negligible ODP. Those fluids should have about the same thermal performances as R134a but are not yet commercially available. They could be good candidates to replace R410A for air to air stationary equipment as well.

Thus 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.

A TEWI analysis of the implementation of CO2 as a refrigerant is proposed in part 6.6 and uses conservative hypothesis of (Ortiz, 2003).

6.2.8 Best available products

(JARN, 2007) reports the energy efficiency current values for different Japanese manufacturers for smaller capacity sizes of air conditioners.

Highest efficient 2.2 kW unit reported
APF : 6,6
COPc : 6.38
COPh : 6.85
COPave : 6.62
APF Target 2010 for units with free dimensions : 6,6

Highest efficient 2.8 kW unit reported
APF : 6.6
COPc (EER) : 6.38
COPh (COP) : 6.67
COPave ((EER+COP)/2) : 6.56
APF Target 2010 for units with free dimensions : 6,6

Size does matter because units with larger capacities do not rank so high in efficiency; above 4 kW, efficiency drops and comes back to levels that are closer to the average values observed on the EU market. Nevertheless, this is again a matter to be developed later in tasks 7 and 8; physically, those levels can be reached for all sizes of units.

Because the T1 and H1 ISO testing points and tolerances are the same in Europe and in Japan, EER and COP are directly comparable. **Best reported EER is 6,4 and best reported COP is 6,9**.

For a number of air conditioners that were sold in Japan between 1996 and 2006, with cooling capacity of 2.8 and 4 kW, Japanese manufacturers made available the 5 testing points of the APF. With these points and the default modeling hypothesis that are supposed to represent an average air conditioner in Japan (JRA:4046 standard), it is possible to compute seasonal efficiency figures in cooling and in heating mode (8 points index to avoid any bias) for the EU seasonal indexes (without auxiliary mode consumption) reported in task 4. For cycling, the Japanese APF hypothesis of a linear degradation with decreasing load ratio is kept and Cd equals 0.25 (see task 4.5 for more details). The results are presented in the graph below.

Figure 6-21: Ranking of Japanese reversible inverter air conditioners with CSPE (JRA4046, 2004) and European SEERon, capacity 2.8 and 4 kW

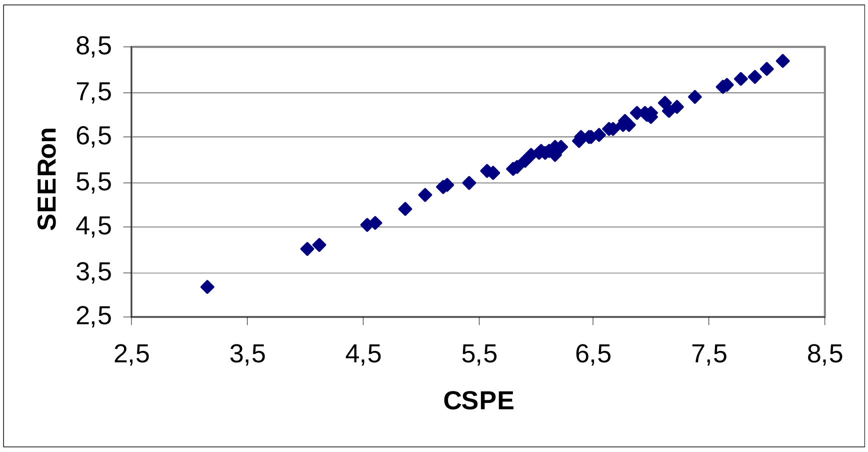
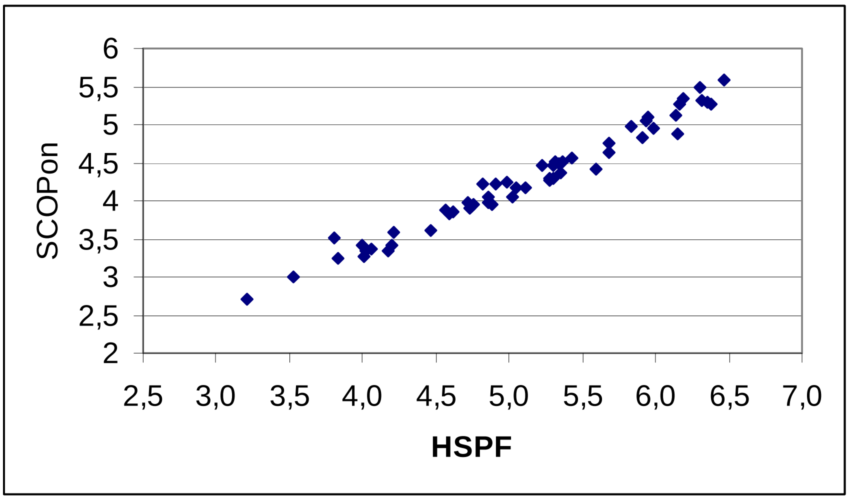


Figure 6-22: ranking of Japanese reversible inverter air conditioners with HSPF (JRA4046, 2004) and European SCOPon, capacity 2.8 and 4 kW



In terms of seasonal cooling performance, the best product for cooling mode efficiency has a CSPE (Japanese SEER) of 8.1 and a SEERon of 8.2 with the default JRA:4046 hypothesis to model the performance of the unit.

In terms of seasonal heating performances, the best product for heating mode efficiency has a HSPF (Japanese SCOP) of 6.5 which would equal for that specific product a SCOPon of 5.5.

6.3 Preheating function

Crankcase heater consumption is the second largest consumption cause for reversible air conditioners with the assumption that the heater is cut at 10 °C outdoor. This technology is simple and is thought to be common for most sold reversible split models on the EU market. Energy consumption levels identified in task 4 are twice this value for a control at 20 °C and 4 times without temperature control. Going to 0 °C would enable to divide present consumption by a factor of ten.

As explained before, the oil heater enables to maintain the oil temperature higher than in the other parts of the system to avoid too high refrigerant concentration levels in the oil. (Danfoss, 2008) gives useful indications to how far the crankcase heater consumption may be reduced. It is necessary to maintain the compressor crank temperature 18 °F (10 °C) higher than the refrigerant temperature in the other parts of the system. Hence by keeping track of representative temperatures of the oil sump and of the outdoor unit, it is possible to cut energy consumption by 50 % (Daikin, 2006) by using this control in addition to an outdoor air 12 °C control. The drawback is that it necessitates again more the CPU of the units for that control so that standby and off mode power is likely to increase with present electric circuiting.
In addition, (Danfoss, 2008) explains that the crankcase is only necessary before start-up, but the time is not given because dependent on each system. But this could lead to further decrease in energy consumption.

By installing a solenoid valve before the expansion valve or using an electronic expansion valve that can close and remain refrigerant tight, it is possible to pump down the evaporator before the compressor is cut (by a BP switch). Nevertheless, there may be other energy drawbacks to this method since the compressor is to be started in case the valve leaks.

Since the oil heater is generally located outside the compressor shell, most part of the energy supplied may not serve the purpose. Compressor and equipment manufacturers use electrified stator coils that enables to reduce the power required to heat the oil sump. Best available technologies, following information supplied by (JRAIA, 2006) is about 10 W for 2.8 kW and 25 W for 8 kW split units respectively equipped with rotary vane single and double stage compressors, or about 3.5 W by kW cooling.

Different BAT technologies can be kept for the control of the crankcase heater. Either the control temperature is lowered, or (and) the control is improved by heating in function of the difference between oil sump temperature and indoor refrigerant temperature, or (and) the crankcase heater is used only to preheat before operation, that can be done with no comfort decrease for the end user whether its operation has been programmed, which should become the standard situation in heating mode, if we take as reference other heating equipment than vapor compression cycles. The combination of these options should lead to virtually null energy consumption of this function.

Best available technology would be to use the crankcase heater only before cooling or heating function and thus it should disappear from off mode functions and partly from the standby energy functions.

Starting from the present 10 °C outdoor air temperature control, the energy consumption of the crankcase heater:
- can be divided by 3 with an electrified stator coil,
- can be divided by 2 by improving control in function of real temperature differences inside the refrigerant circuit (information supplied by JRAIA also.

Then best available technologies enable to divide the preheating function power consumption by a factor 6.

In addition, it could be considered to limit the use made of the preheating function to hours before operation of the compressor when the indoor unit (or one of the indoor unit for split systems) is activated. This cannot be figured out within the frame of the calculations led up to now because it has been supposed that reversible split systems remained on all the year long either for cooling or heating operation. This is only an average case that cannot represent all situations. And this option would avoid a end-user wants to cut completely the air conditioner energy consumption and the equipment still draws power for the preheating function.

6.4 Standby and off mode functions

Best performing reversible split products in **standby** consumes **0.7 W**. This is for units with current supplied by outdoor unit, meaning that current flows via the outdoor unit to the indoor unit by the communication line in order to enable the reactivation and status display functions. This is now common in Japan for units with cooling capacity inferior to 4 kW.

It has to be kept in mind that power measurement is made at 35 °C outside. It means that all these units (that a priori have a preheating function for the oil sump) have a crankcase heater controlled for a lower outdoor air temperature than 35 °C. It also means that the fact that the unit is ready to activate the preheating equipment (and correspondingly senses temperature(s)) is included in this power consumption.

According to EuP Lot 6 Task 7, standby could be reduced to 0.3 W for the indoor unit by switching off other than the reactivation function (the supply is inferior to 10 W) and for the outdoor unit it depends on the size of the crankcase heater. Whether the crankcase heater is of more than 45 W, 1.25 W could be reached while 0.4 W would be feasible under 45 W.

For larger capacity units which could have several indoor units, the same technology that is used for small split systems could enable to reach 1 W or 1.85 W depending on the crankcase heater size so that BAT would be of **1 or 1.85 W for bi-split units and 1.4 W or 2.25 W for tri-split units depending on the capacity size of the unit and the efficiency of crankcase heaters**.

For split air conditioners without a crankcase heater, 0 W off mode should be the general situation (when the indoor unit is hard switched off). However, because most of the time current is supplied by the outdoor unit and of the remote control sensing, off mode power equals standby power. According to EuP Lot 6 Task 7, standby could be reduced to 0.3 W for the indoor unit by switching off other than the reactivation function (the supply is inferior to 10 W) with a dedicated PCB and 0.4 W for the outdoor unit. Standby could hence be cut to 0.7 W as for reversible split units. However, for this type of unit, a hard off switch could help to reach 0 W off mode.
Hence, BAT for cooling only split units is 0.7 W in standby mode and 0 W off mode.

For **moveable air conditioners**, standby mode would include LCD display and remote sensing and best available products consume in that case **1.24 W** (Delonghi, 2007). Moreover, the unit can be hard switched off, leading to 0 W power for off mode.
For standby, best available technology reaches far lower levels, under the condition it can be accepted not to display information on the LCD screen. Even in the case a timer program is used, there are options available to drastically reduce these values (EuP Lot 6, 2007) to **0.3 W and lower**.

6.5 Thermostat off

For hours without cooling or heating needs but when the end-user still requires the temperature to be maintained at a given set point, the air conditioner electric consumption is the sum of the indoor fan electric consumption and of the electronic controller. At that time, the indoor room air temperature is checked by the temperature sensor (and possibly humidistat) located in the indoor side heat exchanger. In order to get a correct representation of the room conditions, the fan is generally working at minimum speed (less noise and electric consumption) and then still draws power.

Whether the fan may represent a small fraction of the power demand when the air conditioner is on, this number of hours can get very high for instance if the end-user requires a set point for a long period. For thermostat off functions, the energy consumption will be the sum of the fan consumption plus the PCB of both units. For single duct units with only one fan, the power is greater since the fan power should cover evaporator and condenser flows.

Best available technologies of fan at low speed (between 5 and 10 W), CPU 2 W (EuP Lot 1 Task 6 report, 2007) and 1 W for information display would lead to 13 W for 3.5 kW units.

Best available technology level is then of 13 W for split 3.5 kW and single duct units and 23 W for split 7.1 kW units. In addition to the gain on the thermostat-off hours with no cooling load, there is second bonus on the part load performance for operation in thermostat-off mode with non null cooling load.

6.6 Best available technologies

6.6.1 Energy efficiency potential

To supply the same function, which is cooling or heating indoor air temperature at a given set point, the best available product corresponds to the small size split air conditioners presently available on other than the European markets (Japan, USA). Best available technologies of these products are gathered below and give an idea of the BAT levels for both heating and cooling electric consumption (SEER 8.2 and SCOP 5.5).

Table 6-3: Energy consumption of best available technology, 3.5 kW cooling reversible split

Energy consumption per unit / year		BAT kWh/unit/an
Cooling mode	Compressor on	133
	Thermostat off	6
	Stand-by hours	2
	Off mode	0
Heating mode	Compressor on + electric resistance	532
	Thermostat off	37
	Stand-by hours	0
	Crankcase heater	5
TOTAL per unit in kWh/y		**717**

The values in the table above correspond to BAT SCOPon, SEERon and parasitic energy consumption levels shown previously in task 6.

The table below shows the equivalent BAT levels computed for each type of unit and compared with the base case. Potential energy efficiency gains are quite high. Results suggest that BAT technologies could already enable to cut 52 to 76 % of the electric consumption of air conditioners whether applied to all products.

Energy consumption per unit / year		CO split [3,5 kW]	BAT CO split [3,5 kW]	Reversible single split [3,5kW]	BAT Reversible single split [3,5kW]	Cooling only single split [7,1kW]	BAT Cooling only single split [7,1kW]	Reversible single split [7,1kW]	BAT Reversible single split [7,1kW]	Single duct [2,2 kW]	BAT Single duct [2,2 kW]
Cooling mode	**Compressor on**	377,5	133,4	374	133,4	882,6	270,7	839,7	212,3	311,1	83,9
	Thermostat off	23,2	5,9	18,1	5,9	42,5	14,8	33,2	11,8	47	5,9
	Stand-by	13,7	2,3	13,7	1,6	13,7	2,3	13,7	2,3	13,7	0,7
	Off mode	30,5	3,6	0	0,0	30,5	5,1	0	0	0	0
Heating mode	**Compressor on + electric resistance**			957,9	532,4	0		1997,5	1080,0	0	
	Thermostat off			100,7	37,2	0		184,6	75,5	0	
	Stand-by			0	0,0	0		0	0,0	0	
	Crankcase heater			34,9	5,4	0		81,4	11,0	0	
TOTAL per unit in kWh/y		445	**145**	1499	**716**	969	**293**	3150	**1393**	372	**90**
Potential gain per unit in kWh/y		299,7		783,3		676,4		1757,2		281,3	
Potential gain per unit in %		67%		52%		70%		56%		76%	

Table 6-4: Energy efficiency potential by type of unit

6.6.2 TEWI analysis

The detailed environmental impact of the different options were not done using the Ecoreport tool since the main cause for environmental impact was identified to be the energy consumption. As a consequence, our analysis focused on energy efficiency in task 6.

However, some bills of material of best Japanese performers was made available by JRAIA manufacturers. This material was presented previously in task 4. It appears the composition does not vary much from average units to best performers. However, the size of the unit and its total weight are increasing as presented before on Figure 6.7. We have found different evolutions of efficiency and weight of the units, that greatly depends on which type of option is implemented to increase the energy efficiency. However, in general the heat exchanger size increases and despite efforts made to reduce the tube diameters and consequently the refrigerant charge, the refrigerant charge is increasing with performance.

CO2 as a refrigerant for split units

For the 3.5 kW unit, and using the hypothesis derived from the study by (Ortiz, 2003), the comparison has been made between the base case and BAT technologies in order to see whether a CO2 system could compete with R410A air conditioners on a TEWI basis.

The kWh CO2 content is as stated in the EuP Methodology the average EU values of 430 g / kWh. For R410A, hypothesis are the following ones: 3 % leakage rate by year over a 12 year life time, 5 % recovery loss at the end of life. In addition, the refrigerant charge increases with efficiency following the relationship found for Japanese units but BAT units are supposed to be equipped with micro-channel heat exchangers and refrigerant charge of BAT units is then 45 % less.

For CO2 systems, following (Ortiz, 2003), the baseline (noted as CO2_baseline) system is approximately 60 % as efficient as our base case while the best technology available (CO2_best) is 115 % more efficient than our base case.

While most efficient units are available with R410A, it is not the case for CO2, whose figures by (Ortiz, 2003) are issued from models based on prototype testing.

Hypothesis are gathered in the tables below. Results are presented on the two following figures, respectively for cooling only air conditioners and reversible.

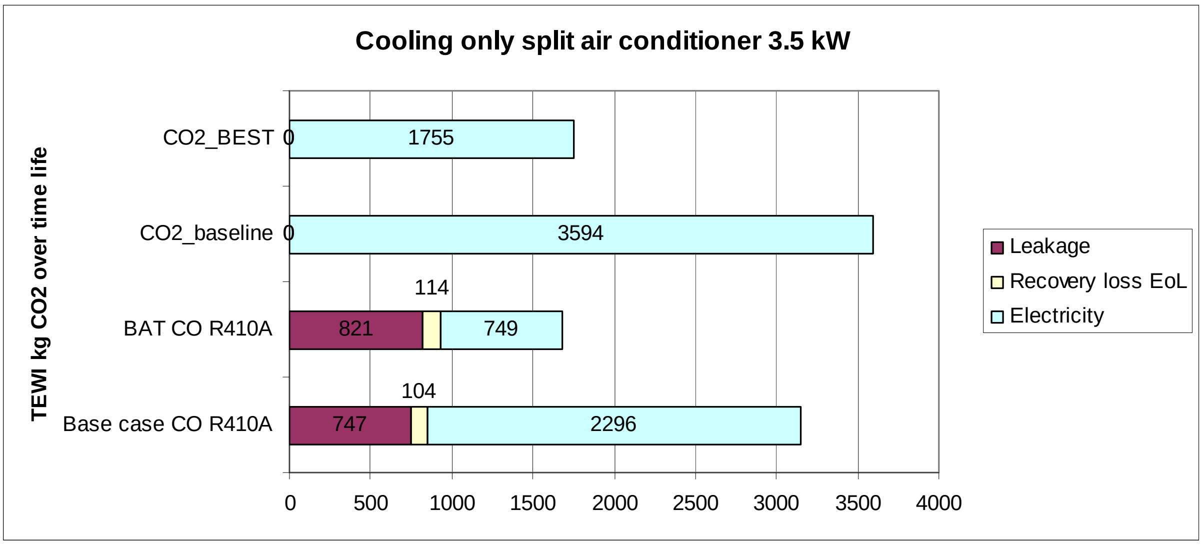


Figure 6-23:TEWI of split cooling only 3.5 kW base case and BAT, R410A and CO2

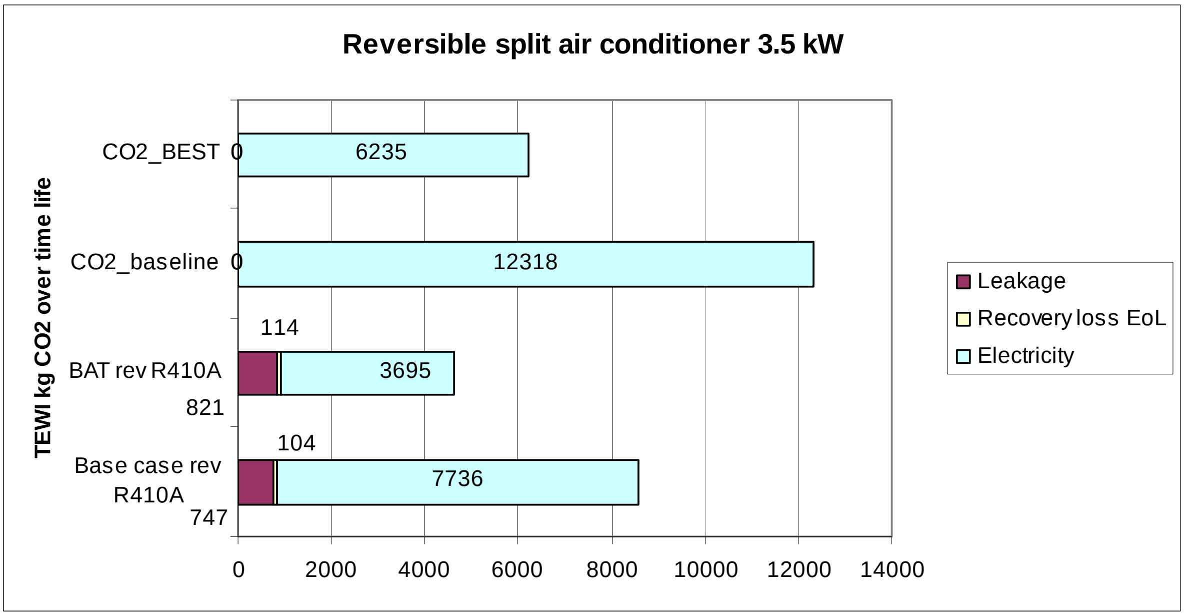


Figure 6-24:TEWI of split reversible 3.5 kW base case and BAT, R410A and CO2

Type	Base case CO R410A	BAT CO R410A	CO2_baseline	CO2_BEST
GWP	1975	1975	1	1
Refrigerant charge kg / kW	0,3	0,33	0,3	0,15
Refrigerant charge kg	1,05	1,155	1,05	0,525
Leak rate yearly	3,0%	3,0%	3,0%	3,0%
Refrigerant leak life time kg	0,378	0,4158	0,378	0,189
Direct emission leak life time kg CO2	747	821	0	0
End of life recovery rate	95%	95%	95%	95%
Recovery refrigerant loss kg	0,05	0,06	0,05	0,03
Direct emission EoL kg CO2	104	114	0	0
El cons heat and cool kWh/year	377,5	133,4	629,2	328,3
El cons parasitic losses kWh/year	67,4	11,8	67,4	11,8
Life time	12	12	12	12

Global electric kWh	5339	1743	8359	4080
Emission rate kg CO2/kWh	0,43	0,43	0,43	0,43
Indirect emissions kg CO2	2296	749	3594	1755
TEWI kg CO2	**3146**	**1685**	**3595**	**1755**
Leakage	24%	49%	0%	0%
Recovery loss EoL	3%	7%	0%	0%
Electricity	73%	44%	100%	100%
Leakage	747	821	0	0
Recovery loss EoL	104	114	0	0
Electricity	2296	749	3594	1755

Table 6-4: TEWI calculation and hypothesis of cooling only split 3.5 kW units

Type	Base case rev R410A	BAT rev R410A	CO2_baseline	CO2_BEST
GWP	1975	1975	1	1
Refrigerant charge kg / kW	0,3	0,33	0,3	0,33
Refrigerant charge kg	1,05	1,155	1,05	1,155
Leak rate yearly	3,0%	3,0%	3,0%	3,0%
Refrigerant leak life time kg	0,378	0,4158	0,378	0,4158
Direct emission leak life time kg CO2	747	821	0	0
End of life recovery rate	95%	95%	95%	95%
Recovery refrigerant loss kg	0,05	0,06	0,05	0,06
Direct emission EoL kg CO2	104	114	0	0
El cons heat and cool kWh/year	1331,9	665,8	2219,8	1158,2
El cons parasitic losses kWh/year	167,4	50,2	167,4	50,2
Life time	12	12	12	12
Global electric kWh	17992	8592	28647	14500

Emission rate kg CO2/kWh	0,43	0,43	0,43	0,43
Indirect emissions kg CO2	7736	3695	12318	6235
TEWI kg CO2	**8587**	**4630**	**12319**	**6236**
Leakage	9%	18%	0%	0%
Recovery loss EoL	1%	2%	0%	0%
Electricity	90%	80%	100%	100%
Leakage	747	821	0	0
Recovery loss EoL	104	114	0	0
Electricity	7736	3695	12318	6235

Table 6-5: TEWI calculation and hypothesis of cooling only split 3.5 kW units

Given the figures of (Ortiz, 2003) for CO2 units and the hypothesis depicted hereabove, it seems that whether CO2 was used instead of R410A, it could reach equivalent savings in terms of global warming for cooling only units but not for reversible ones. Without the microchannel option for R410A units, CO2_BEST unit (cooling only) would emit less CO2 than BAT R410A unit.

Of course, in case refrigerant would be lost at the end of life and not recovered, the picture would be different: CO2 emissions of CO2 cooling only units would be lower whatever the efficiency of the R410A cooling only units. Reversible units situation would then look like the present situation of cooling only units, offering a large playing field for CO2 improved units.

This type of analysis gives a rather clear signal to the necessity to decrease the charge, leaks and recovery losses for HFC units and of the importance of the EC regulation on HFC refrigerant, but precise comparison is a problem because of very few data regarding refrigerant leaks and recovery at the end of life in the present situation.

Adopting CO2 as a refrigerant would be at the cost of a much lower energy efficiency following (Ortiz, 2003). However, according to (Jakobsen, 2006), split air conditioner with CO2 could be as efficient as R410A ones.

It should be added that with the perspective of a greener electricity for the years to come in Europe, the efficiency required for CO2 units to emit less GHG than R410A units on a TEWI basis is likely to increase.

The direct CO2 emission shares would evolve from about 25 % with present base case to 55 % for BAT R410A cooling only products and respectively from less than 10 % to about 20 % for reversible units.

As a conclusion, it should also be added that proven BAT split products not only have the potential to reduce energy consumption of 50 to 75 % but also total CO2 emissions from 40 to 50 %.

Propane as a natural refrigerant for single duct air conditioners

The TEWI evaluation is also done for single duct units hereafter.

In this task BAT technology regarding energy efficiency has been set to split energy efficiency level (noted BAT Task 6). Nevertheless, improvement to the single duct base case have been done in task 7 and enable to reach a SEERon of about 7 using R410A. Given that it is possible to reach these values with propane as a refrigerant, TEWI analysis is detailed for 6 units, the base case single duct, the base case single duct with propane instead of R410A, the base case R410A cooling only split unit (with 2.2 kW capacity), the BAT split unit with R410A, the BAT (Task 7) single duct unit with R410A and the BAT (task 7) unit with propane. Propane, in addition of gains on direct emissions also increases the performances in cooling mode by 7 %.

Type	Base case SD R410A	Base case SD Propane	Base case split R410A [2.2 kW]	BAT (Task 6) split R410A [2.2 kW]	BAT SD (Task 7) R410A	BAT SD (Task 7) Propane
GWP	1975	3	1975	1975	1975	3
Refrigerant charge kg / kW	0,30	0,15	0,16	0,21	0,33	0,17
Refrigerant charge kg	0,66	0,33	0,35	0,46	0,73	0,36
Leak rate yearly	1,0%	1,0%	3,0%	3,0%	1,0%	1,0%
Refrigerant leak life time kg	0,08	0,04	0,12	0,16	0,09	0,04
Direct emission leak life time kg CO2	156	0	622	821	172	0
End of life recovery rate	95%	95%	95%	95%	95%	95%
Recovery refrigerant loss kg	0,03	0,02	0,04	0,06	0,04	0,02
Direct emission EoL kg CO2	65	0	86	114	72	0
El cons cool kWh/year	311	291	237	84	101,9	95,2
El cons parasitic losses kWh/year	46,9	46,9	67,4	11,8	6,6	6,6
Life time	12	12	12	12	12	12
Global electric kWh	4296	4052	3656	1148	1302	1222
Emission rate kg CO2/kWh	0,43	0,43	0,43	0,43	0,43	0,43
Indirect emissions kg CO2	1847	1742	1572	494	560	525
TEWI kg CO2	**2069**	**1742**	**2281**	**1429**	**803**	**525**
Leakage	8%	0%	27%	57%	21%	0%
Recovery loss EoL	3%	0%	4%	8%	9%	0%
Electricity	89%	100%	69%	35%	70%	100%
Leakage	156	0	622	821	172	0
Recovery loss EoL	65	0	86	114	72	0
Electricity	1847	1742	1572	494	560	525

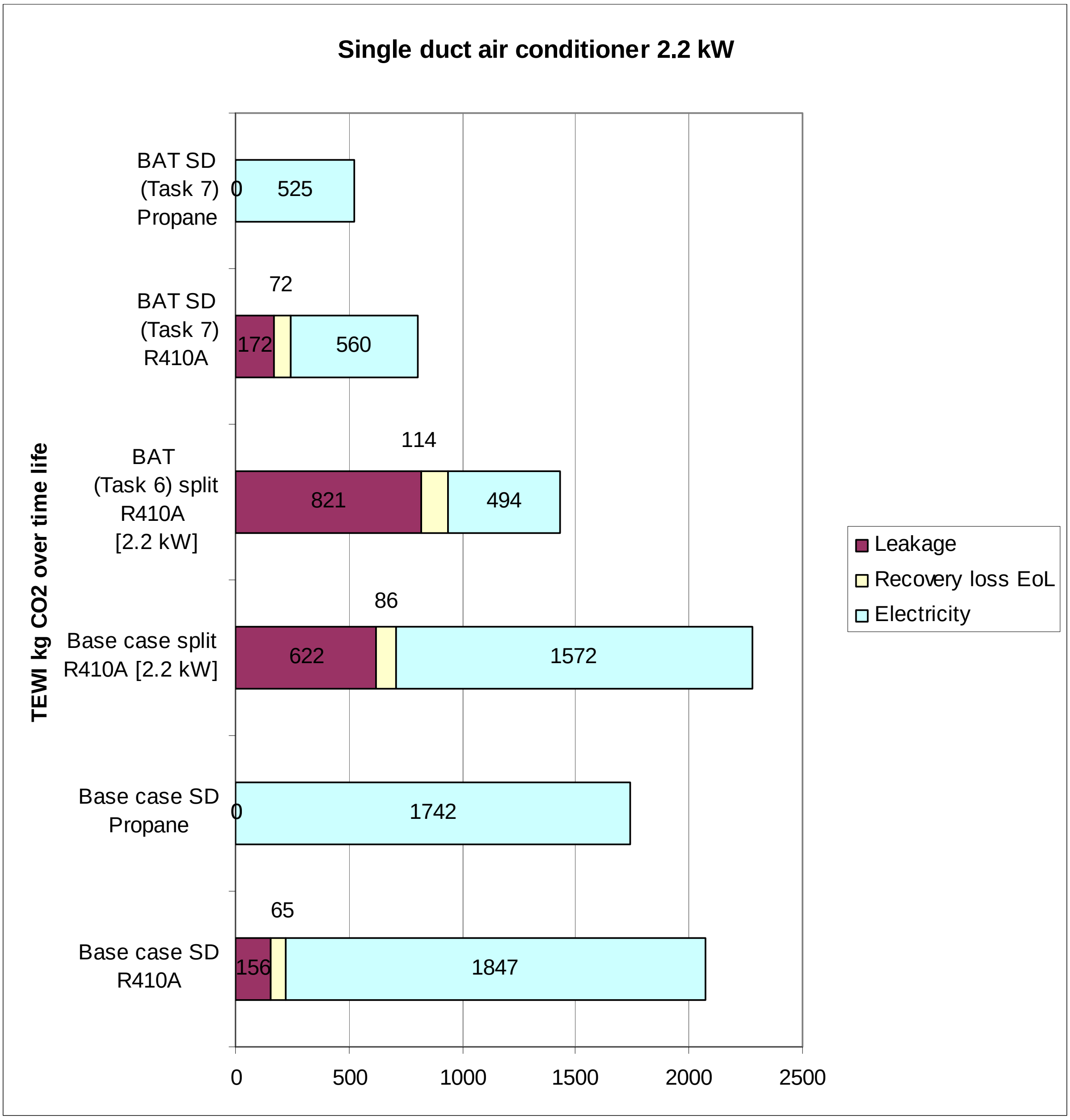
Table 6-6: TEWI calculation and hypothesis of single duct 2.2 kW base case, propane and BAT units

Results are presented on the graph below. A single duct base case unit (with the same energy efficiency level) can save about 10 % of direct emissions. When considering the best available technologies applied to the product (this leads in task 7 to a SEER about 7 with R410A as a refrigerant), propane can help decreasing total emissions by as much as 25 %.

Despite higher energy consumption, BAT single duct with R410A would have 30 % lower TEWI values as compared to a BAT split cooling only unit because of lower leak levels and BAT single duct products would have a TEWI of less than 40 % of the one of the BAT split of equivalent cooling capacity.

Hence propane appears as an important advantage regarding CO2 emissions for single duct units.

Figure 6-25:TEWI of single duct, split 2.2 kW, base case and best available technologies, with R410A and propane – for single duct units only



Conclusion

The main environmental impact being energy consumption, available technologies have been screened that could help cut the energy consumption of the products. This analysis has been led at cycle and component level.

Not only the efficiency could be improved but products could help to cut the cooling load by better interaction with their environment and with the end-user.

The different power modes have been reviewed and potential energy gains appear to be very high, superior to 50 % for all products when the BAT product is defined as the best air conditioner sold in Japan.

Finally, a TEWI analysis enables to compare total global warming impact of base case and their BAT levels as opposed to adopt natural refrigerant fluids in order to limit the emissions.

It appears that cooling only CO_2 units could be efficient enough to lower the total emissions linked to the R410A base case and reach similar total equivalent CO_2 emissions as BAT R410A cooling only units, whereas it is not the case for reversible units. The adoption of micro-channel heat exchangers would be an important advantage for R410A units that can be more efficient and emit less than CO_2 units both for cooling and reversible units.

Anyway, in both cases, adopting CO_2 as a refrigerant would be at the cost of an important energy efficiency loss following (Ortiz, 2003) whereas other authors indicate that on a seasonal performance basis, CO_2 units could be competitive (Jakobsen, 2006). Without present information, it seems rather difficult to conclude.

For single duct units, propane appears as an important advantage that could compensate the lower energy efficiency levels reachable by these products as compared to split when CO_2 emissions are considered.

REFERENCES

R6-Technical Analysis of Best Available technology

Bensafi, 2006, Evolution des technologies en climatisation au Japon, Revue pratique du Froid, 2006.

Carrier, 2007, Commercial documentation on microchannel heat exchangers, www.carrier.com.

CEN, 2006, Draft standard prEN 15251, Criteria for the indoor environment including themal, indoor air quality, light and noise, 2006

Clodic, 2007, The latest refrigeration technologies: how to lower the environmental impacts, Communication à : "XII European conference on the latest technologies in refrigeration and air conditioning", Milano, 8-9 June 2007

Cremaschi, 2007, HPC, 2007, Heat pump Center, Newsletter n°3, 2007.

Daikin, 2006, Personnal communication.

Daikin, 2007, HPC, 2007, Heat pump Center, Newsletter n°3, 2007.

Danfoss, 2008, Refrigerant Transfer and Compressor Damage, 11 January 2008, www.danfoss.com.

Delonghi, 2007, CEN TC 113/WG7 presentation A Andriani

Delonghi, 2007, Personal communication.

ECCJ, 2006, Final Summary Report by Air Conditioner Evaluation Standard Subcommittee, Energy Efficiency Standards Subcommittee of the Advisory Committee for Natural Resources and Energy; TAKANORI SAITO et al]

EuP Lot 6, Final report, 2007.

Hu, 2005, Study of a high efficiency residential split water-cooled air conditioner S.S. Hu, B.J. Huang, Received 20 August 2004; accepted 20 November 2004. Applied Thermal Engineering.

Jakobsen, 2006, Djakobsen, Skiple, Neksa, Wachenfeldt, Stene & Skaugen, Norway, Development of reversible residential air conditioners and heat pumps using CO2 as working fluid, IEA Heat Pump Center Newsletter, Volume 24 – No. 3/2006.

JARN, 2007, Oct 2007.

Mitsubishi Electric, 2006, Mamoru Kawakubo and Masaaki Yabe, Technologies of Power Electronics for Air Conditioner and Household Electric Appliances, ADVANCE Technical Reports, March 2006, and web site.

Norland, 2006, How High Will SEER Go? by Jim Norland, August 7, 2006 http://www.achrnews.com

Ortiz, 2003, Thomas M. Ortiz, Daqing Li and Eckhard A. Groll, EVALUATION OF THE PERFORMANCE POTENTIAL OF CO2 AS A REFRIGERANT IN AIR-TO-AIR AIR CONDITIONERS AND HEAT PUMPS: SYSTEM MODELING AND ANALYSIS, ARTI-21CR/610-10030.

Perrotin, 2004, Thèse de Doctorat en Énergétique de l'EMP, Etude côté air d'échangeurs à haute efficacité énergétique utilisés dans les systèmes de conditionnement d'air résidentiels - Air-side investigation of high efficiency heat exchangers used in room air-conditioners.

UNEP, 2002, REPORT OF THE REFRIGERATION, AIR CONDITIONING AND HEAT PUMPS TECHNICAL OPTIONS COMMITTEE, 2002 Assessment.

USDOE, 2001, Energy efficiency standards for consumer products: residential central air conditioners and heat pumps, 2001.