Technological innovations in heat pump systems

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Abstract It is estimated that about 140 million heat pumps are working in the world. This evidences that the equipment conceived by Lord Kelvin back in 1852 has finally penetrated into the market. In the last 20 years the heat pump has improved both regarding the thermal exchange surfaces, the compressor, and the control and defrosting systems. Thus not only was the COP strongly improved but also seasonal performances. Further equipment improvements are aimed to better exploit the properties of new refrigerants, utilising even the pressure drop between condenser and evaporator, usually dissipated by throttling. Gas driven heat pumps were also improved with higher efficiency i.c. motors and much longer maintenance intervals. Absorption heat pumps are now available in many different models suitable to different applications. Probably most improvements took place with lower heating temperatures and the use of cold sources more suitable than the outside air, above all surface and underground water, the ground and the recovery in mechanical ventilation systems.

Keywords heat pump; innovation

1. A survey of the actual heat pump market

During the last ten years the heat pump market has spread greatly in the world. After developing in the US and Japan, it has spread towards new and unexpected markets, particularly in China. A recent evaluation gives an estimate of 130–140 millions of heat pumps in the world with an annual thermal production estimated at 1300 TWh for 2001. This energy is used for 57% in residential heating, 27% in commercial applications and 16% in industry.

The first promoting element of thermodynamic heating, as heat pump heating is sometimes called, was the reversible operation. An inversion valve on a summer air conditioner allows switching from summer operation with an evaporator inside to winter operation with a condenser inside (Fig. 1). This promoted firstly the selling of heat pumps in the US during the ’70 s (more than 100,000 units were already installed in 1972). Then, secondly, it favoured the spreading of heat pumps in Japan where at the beginning of the ’90 s more than one million units were sold every year. Even nowadays most heat pumps are installed in residential buildings in temperate and mild climates where summer cooling is requested and during winter outside air can be a very suitable heat source.

Other applications are, however, increasing from surface or ground water heat pumps to ground heat pumps. The latter can be favourably used even in cold climates and they are used increasingly in Northern countries such as Canada and Sweden.

Other heat pump developing fields are in heat recovery in building mechanical ventilation and in sanitary water heating. Yet one sector of the market which should

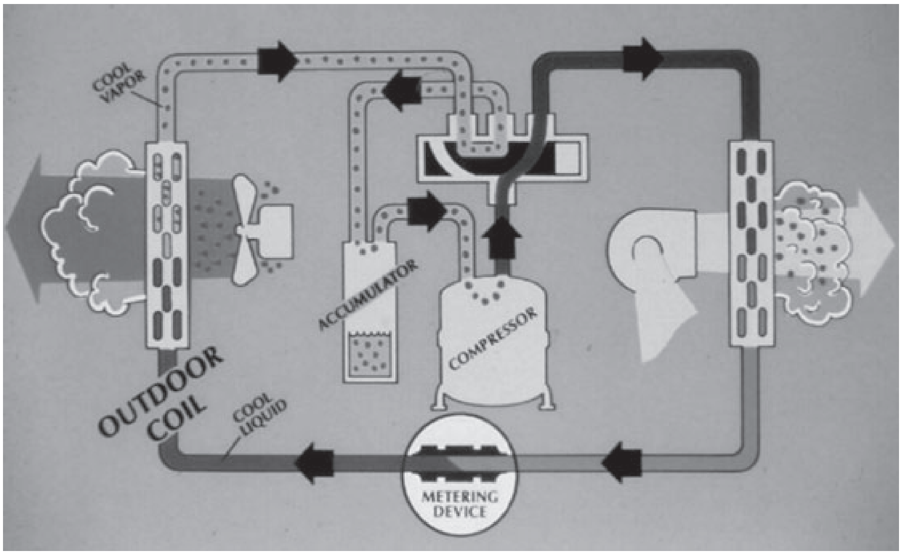

Figure 1. *Winter operation of a reversible heat pump (doc. Carrier).*

not be neglected is heat pumps driven directly by heat either by an i.c. engine (but recently also by an e.c.) or absorption heat pumps.

The market growth is particularly important in Europe where the annual increment has two digits in recent years, such as Germany, Austria, Switzerland and Sweden. Let us consider the example of Germany where the increase of the heat pump market was 30% in 2004 (Fig. 2).

The heating capacity of this equipment is very different: from a few kW for a sanitary water heat pump to more than 10 MW for some district heating heat pumps installed in Sweden.

From this survey of the market it appears to be of greater interest to understand the technological developments that the various manufacturers offer.

2. A short history of the heat pump

The heat pump is one of the few devices conceived in the theory before practice. Sadi Carnot's analysis back in 1824 primarily applies to the steam engine, but a fundamental role in his *Réflexions* is played by the concept of a reversible cycle that implies the possibility of realising a cooling machine. Passing from cooling machine to heat pump is rather simple for us: it is enough to change the roles of cold source and heat sink. The useful effect of the heat pump is the heat that the cooling machine must dissipate. However, theoretical passing required 28 years. William Thompson, the famous Lord Kelvin, wrote the report 'On the economy of the heating and cooling of buildings by means of currents of air'. It was written in 1852 and it contains the interesting technological proposal of using the air as a working fluid (Fig. 3). The air

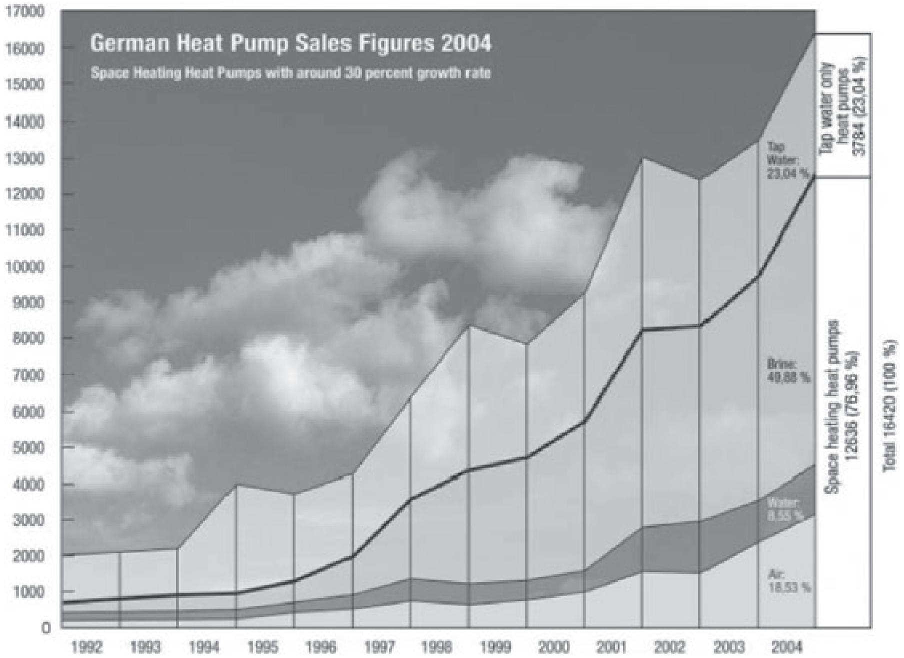

Figure 2. *German heat pump sales from 1992 to 2004.*

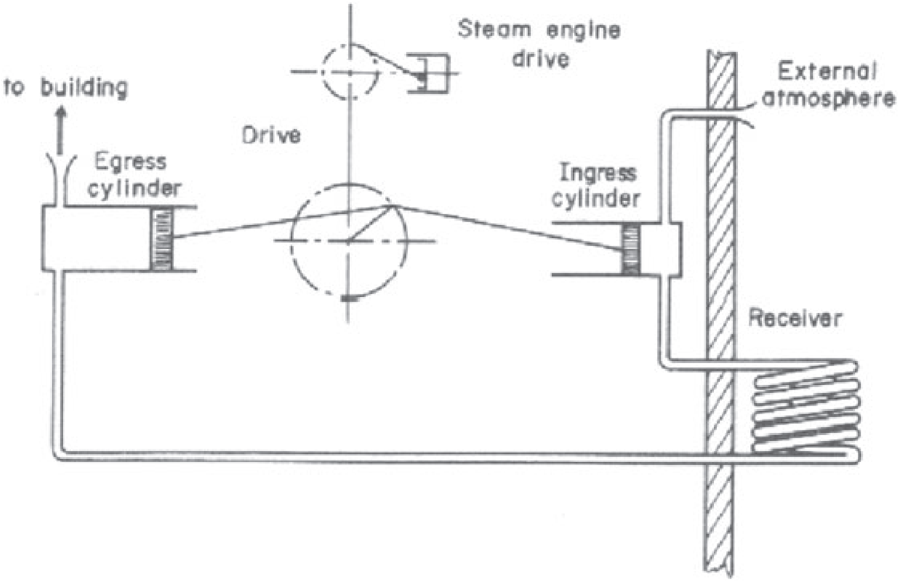

Figure 3. *Lord Kelvin's heat pump.*

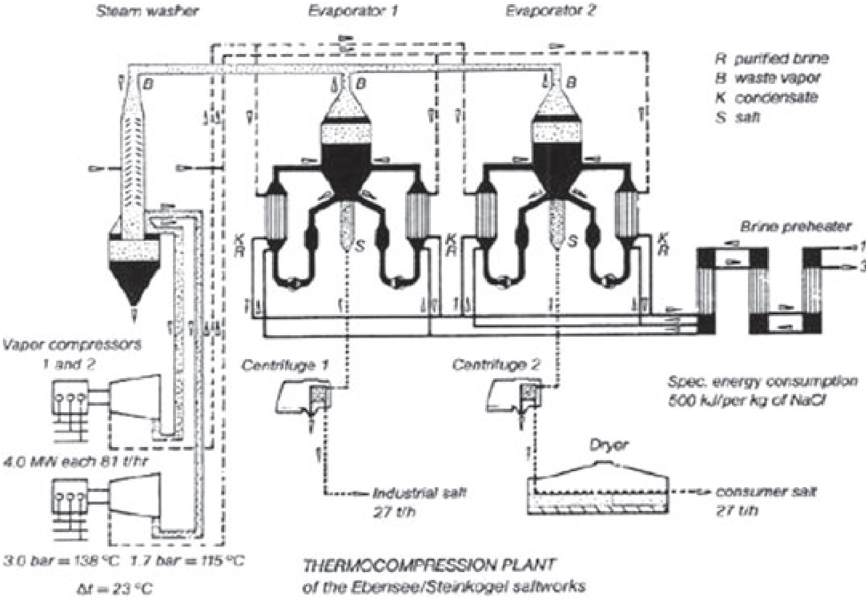

Figure 4. *Schematic of the open cycle mechanical vapour recompression heat pump realised by Peter Ritter von Rittinger in 1855.*

taken from outside is expanded: thus it is cooled well below the outside air temperature and it can be heated by outside air through a simple heat exchanger. Therefore the compression that follows the expansion heats the air to a temperature high enough to heat a room.

Some years after that famous writing, Lord Kelvin added the following note to his work:

> *The method of heating air described in the article remains unrealised today. When Niagara is set to work for the benefit of North America through electric conductors, it will no doubt be largely employed for the warming of houses over a considerable part of Canada and the United States. But it is possible that it will also have applications though less large in other cold countries to multiply the heat of coal and other fuel, and to utilise wind and water power for warming houses.* [1]

Meanwhile practical knowledge was also developed. In 1851 John Gorrie, a doctor and amateur technician, patented the first compression cooling machine, realised on the purpose of relieving his patients' suffering in Florida's hot climate. The very first heat pump was built in 1855 by the Austrian Peter Ritter von Rittinger who installed an open-cycle mechanical vapour recompression (MVR) unit, directly driven by hydro energy, in the saltern of *Ebensee*, *Upper Austria* (Fig. 4) [2]. The vapour developed in the aqueous solution concentrators at a temperature of 117ºC and a pressure of 170 kPa was compressed till 300 kPa. The condensation temperature at that pressure is 138ºC. Therefore vapour condensation allowed to develop a like vapour quantity at a COP higher than 10. The condensate before discharging preheated the salt diluted solution at the inlet of the concentrators.

During the nineteenth century the heat pump was not considered a useful device both for the eventual cost of the equipment and the lack of suitable refrigerants. Moreover, direct combustion either of wood or coal was much less expensive. Even the cooling equipment had difficulty in its competition with natural ice that was stored in winter for the following summer over the centuries. Organic refrigerants were developed in the '30 s with a progressive development of cooling equipment. Apart from isolated installations (the most in Switzerland and in Scotland and one, well known in Italy, in Milan), the first commercial heat pump spread took place in US during the '50 s. The utilisation of this equipment in North America's icy winters suddenly evidenced the weak points. The much higher pressure ratio gave rise to frequent compressor failures as this device was conceived for lower ratios.

Other problems (defrosting, compressor slugging and lack of lubrication) gave a low reliability reputation to heat pumps and produced a selling stagnation for a 10 year period (from 1963 to 1973). The heat pump rediscovery in the US happened during the first energy crisis when the various design and constructive faults were already faced and solved. However, the two energy crises in 1973 and in 1979 were not conclusive in promoting heat pump selling in spite of very favourable conditions, above all after 1979, with regard to the relative cost of electricity and fossil fuels.

The possible annual utilisation of the heat pump was conclusive for its wide spread use in Japan: 95% of the air conditioners in Japan can operate as heat pumps. The same applies to the recent spread in China, where about 60% of the 11 million heat pumps are reversible. Only recently public opinion and government politics began to accept high initial costs for heating devices against lower working costs and reduced pollution. This allowed not only technological progress in the classical electric heat pumps but also the growing and technological development of niche products such as gas heat pumps.

3. Heat pumps: field experiences

It seems useful to supply a short survey of a study of 237 less than 20 kW heat pump plants in the frame of a project promoted by the *Swiss Federal Office of Energy* [3] before analysing the technological development of heat pump machines and plants.

Failures, users' complaints and seasonal performances were recorded for these plants for the period from 1995 to 2002. About 45% of plants are equipped with air water heat pumps, whereas the rest are water-water heat pumps or brine-water ground source heat pumps. The surveyed *Seasonal Performance Factor* (SPF) was 3.4 for water-water heat pumps and 2.6 for air-water. The former number derived from a wide spreading of data probably due to the different sizing of the ground probes. The survey of the plants built up during the analysis period demonstrates the progressive equipment and plant improvement with an increase of 20% in SPF from 1995 to 2003 both for air source plants (Fig. 5) and ground or ground water systems (Fig. 6).

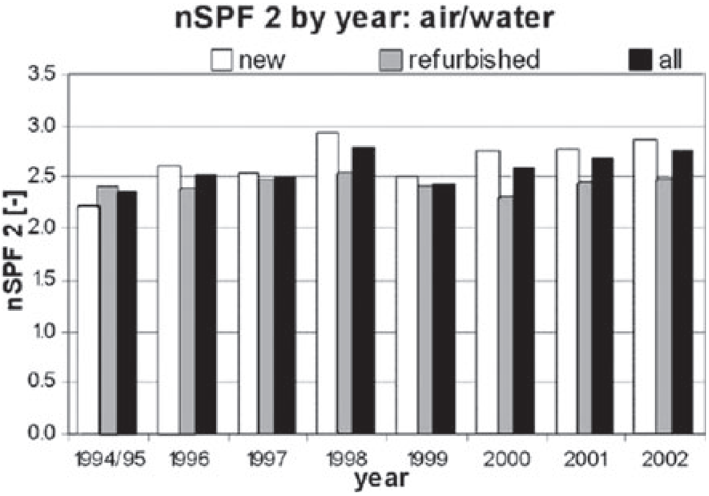

Figure 5. *SPF for air source heat pumps as a function of the year for new, refurbished or all devices.*

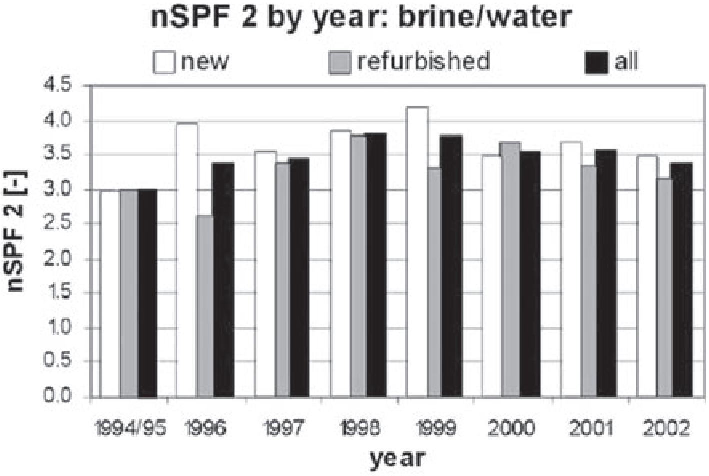

Figure 6. *SPF for ground or ground water source heat pumps as a function of the year for new, refurbished or all devices.*

Everybody is aware of the great importance of customer satisfaction in a market analysis: the study reveals 78% of very happy heat pump owners, 17% quite happy and only 5% are unhappy (a percent that seems physiological: it is also the minimum predicted percentage of dissatisfied in thermal comfort analysis). The considered working period in the survey was of 1,300,000 hours. During this time, 8500 hours of faults occurred. Over 70% of the plants operated without faults. The average

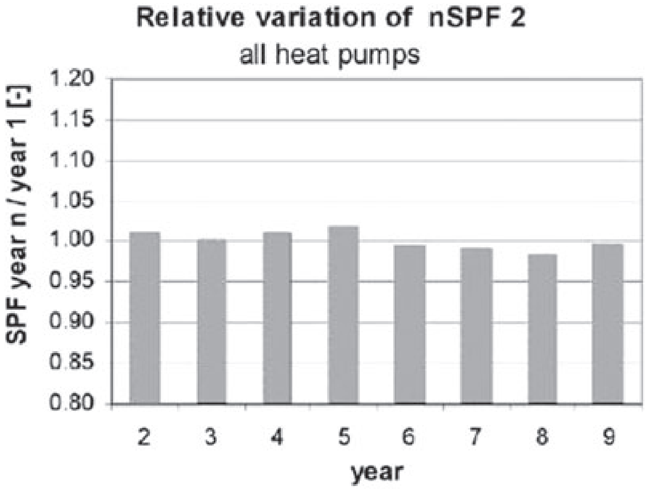

Figure 7. *Relative SPF variation starting from the first working year.*

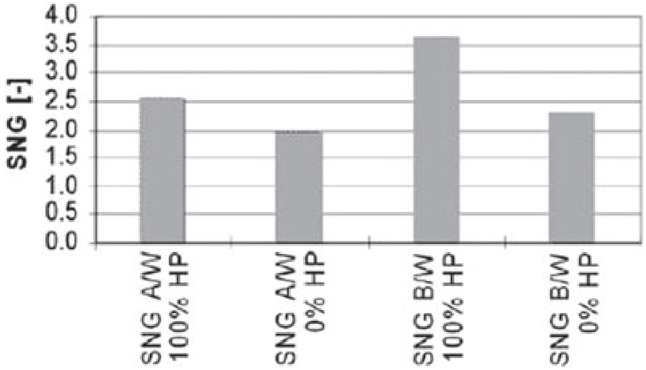

Figure 8. *SPF for air (A/W) and water (B/W) heat pumps according to the provision or not of domestic water heating service.*

duration of the fault (time from notice to rectification) was only six hours (!). Depending on the type of fault, out of service times ranged from about two hours to more than three days.

SPF survey during the years did not show appreciable variations (Fig. 7) with very stable performance. Domestic water heating is very important to improve SPF of the plants with regard to the only ambient heating for both kinds of heat pump systems (Fig. 8).

A final interesting datum regarding ground source heat pumps is where the average circulating fluid temperature was 5ºC, a much higher temperature than

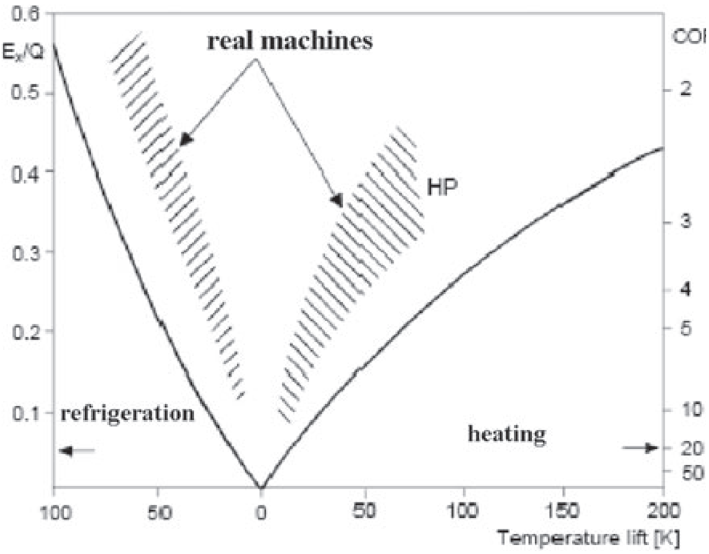

Figure 9. *COP (scale on the right) or requested exergy per thermal unit (on the left) for ideal or real cycles as a function of temperature lift.*

expected, but coming out of a great data sample. Many plants were largely oversized with practically no auxiliary heating operation.

4. Electric heat pumps

When considering the possible use of a heat pump, one should always be aware of the strong sensibility of performance to the low and high temperatures of the cycle, or as otherwise stated, the temperature lift above the cold source temperature.

Performance means both COP and capacity. As far as COP is concerned the maximum performance is given by the reverse *Carnot* cycle:

$$COP = \frac{T_c}{T_c - T_e}$$

Possible COP variations are represented in Fig. 9 as a function of the temperature lift. The lowest values are usually due to small capacity machines whereas the highest are for large size machines. A similar trend strongly influenced by the temperature lift can be found for the capacity.

This suggests at least two important considerations:

1) before every effort to look for higher level heat sources, one should select carefully the heating system in order to lower the heat supply temperature. It is meaningless to operate with systems which need the highest temperature level attainable by the machine (usually around 60ºC), when systems are at hand

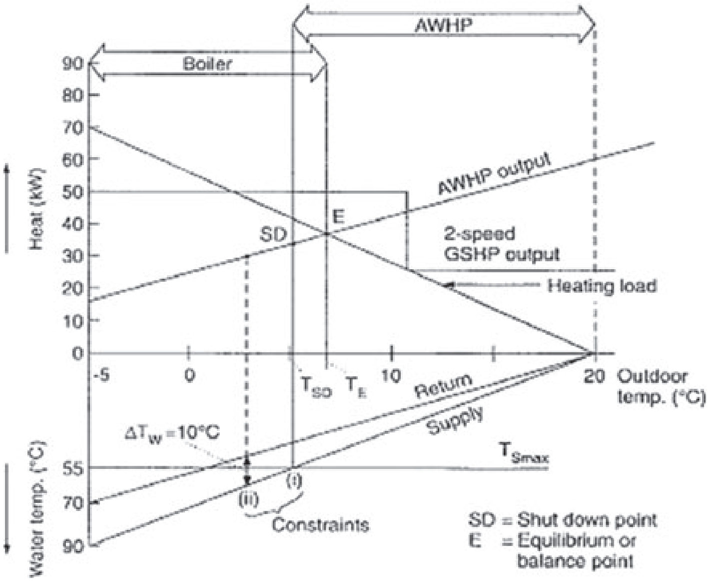

Figure 10. *Capacity line of an air-water heat pump compared with a building load as a function of outside temperature for a conventional radiator heating system.*

which can distribute the heat at temperatures lower than 35ºC, such as heating panels or all air systems.

2) the outside air, the most common heat pump source, is thermodynamically the least favourable source since the thermal load is increasing when its level is decreasing so that both heat pump COP and capacity decrease.

Let us consider in Fig. 10 the heating load as a function of the outside air temperature compared with an air water heat pump (AWHP) output [4]. Capacities find their balance point in E. On the left, heat pump output is no longer sufficient to compensate the heat load and an auxiliary boiler must provide the left part. The downside in the figure is that the trend of the water temperature is represented for supply and return to a radiator system with a control on the supply temperature when the outside air temperature varies. The choice of the balance point is fundamental for the plant economy. Too much '*on the right*' means lower initial costs as the machine capacity is lower but also the annual contribution to the heating requirements is lower. Too much '*on the left*' means a higher initial cost as the capacity is higher but the device might operate at reduced load for a long period. The outside air temperature distribution must be known for a correct sizing.

Once the balance point is fixed, the operation can be either in parallel with the auxiliary heater for lower temperature or in series with the boiler after the heat pump. Heat pump operations suffer some constraints. One is the reaching of the maximum heat pump working temperature. In the figure this is reached when a 55ºC temperature is demanded for an outside air temperature of 5ºC. The other constraint is due to the heat pump contribution when set in series with the auxiliary. Again this happens in the figure at 55ºC supply for an outside air temperature of 1ºC.

This example was considered at length because it underlines some important points. Firstly, the described situation would be much more favourable with a low temperature heating system such as a radiant floor or ceiling with supply temperatures as low as 30–40ºC. The heat pump would have developed a higher heating capacity with a higher COP. The same machine had allowed a balance point more '*on the left*' with a longer utilisation period of the heat pump without the working constraints described before. Secondly similar and cumulative benefits could be obtained with a constant temperature cold source, as reported in the figure by the two capacity steps of a double speed ground source heat pump. The balance point shifts to lower outside temperatures, though the heat pump COP keeps high.

The electrical heat pump is still the most widespread type in the different applications. The most outstanding technological innovations were produced at the beginning of the '90 s using screw compressors for medium capacity and scroll or rotary compressors for lower capacities. At that time the partial load operation was already carefully considered with the first variable speed inverter systems.

Moreover a careful design was devoted to the air side heat exchangers with louvered or corrugated fins and finned or ribbed tubes. The use of enhanced heat transfer surfaces (produced by specialised firms such as *Wieland* or *Hitachi*) began just then. The purpose was to reduce the temperature shift between the outside fluids and the refrigerant both in evaporation and condensation (Fig. 11).

This effort has continued recently with microchannel heat exchange surfaces (Fig. 12) where the refrigerant charge can be reduced (by more than 40%) and performance improved keeping the same volume of heat transfer surface. Other efforts

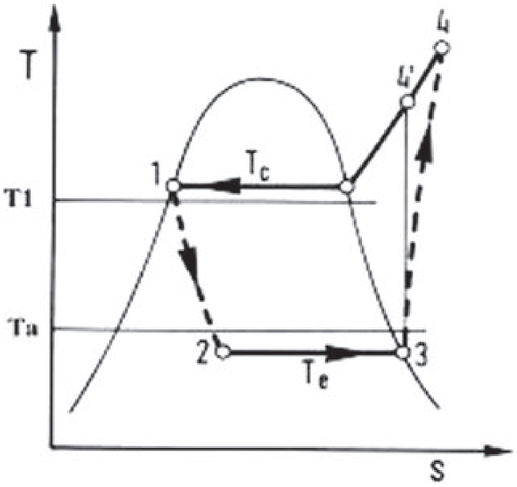

Figure 11. *Temperature differences in a heat pump cycle between outside fluids and refrigerant.*

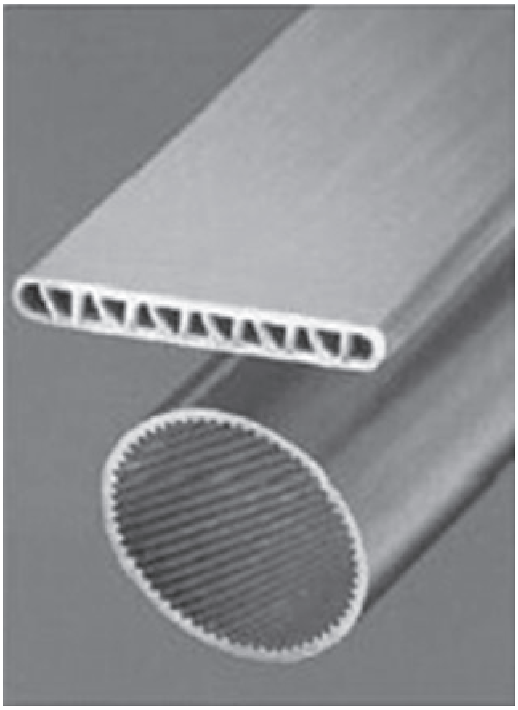

Figure 12. *Comparison between a microchannel heat exchanger and a conventional tube.*

were devoted to the CFC substitutes both toward fluids such as R410A and natural refrigerants, particularly propane, ammonia and above all carbon dioxide.

Perhaps the most outstanding innovations were related to compression and expansion processes from condenser and evaporator (and vice versa). As far as compression is concerned, a two stage process was realised, particularly for fluid characterised by high pressure lift such as carbon dioxide. A possibility was the injection of liquid refrigerant during the compression. This is possible both in screw and centrifugal compressors (Fig. 13).

For operation at partial loads a system less expensive than inverter speed modulation was conceived. It operates on scroll compressors under the commercial name of digital scroll. A simple lifting of 1 mm of the fixed scroll stops the gas supply that is bypassed between inlet and outlet of the compressor (Fig. 14). The lifting allows modulating alternating period of full compressor load with periods of by pass by 20 seconds cycles. The full compressor load can extend from 20 seconds (maximum capacity) to 10 seconds (half capacity) till 2 seconds (20% capacity) thus modulating in a simple and reliable way.

Regarding the expansion, usually achieved by throttling, prototypes have been developed to exploit the refrigerant expansion to produce a fraction of the compression work. The benefit can be appreciable in transcritical cycles such as with carbon

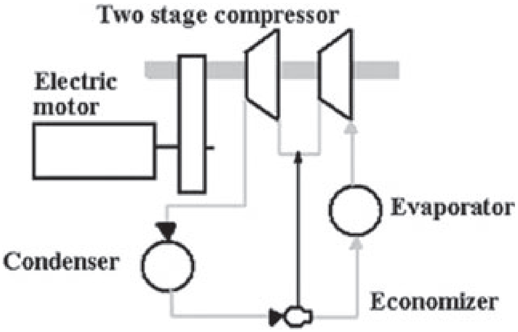

Figure 13. *Two stage compression.*

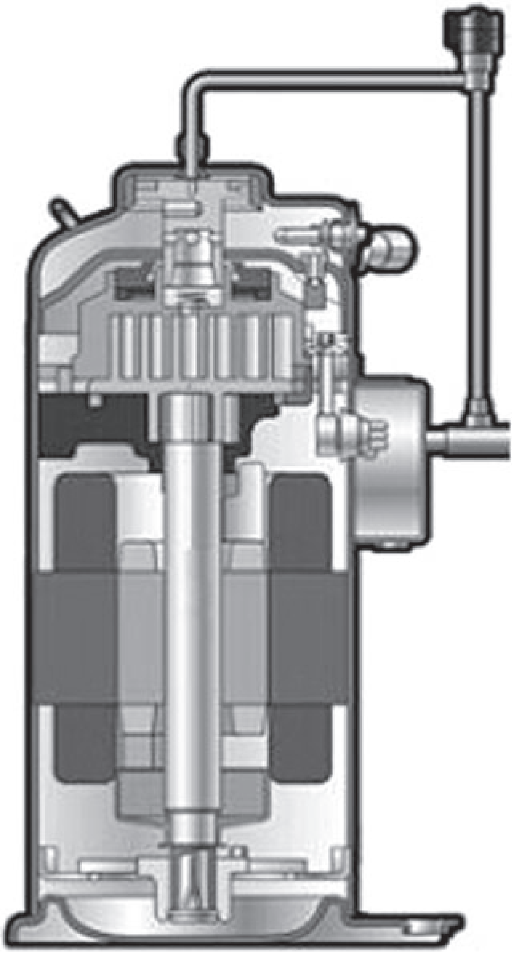

Figure 14. *Schematic of digital scroll compressor.*

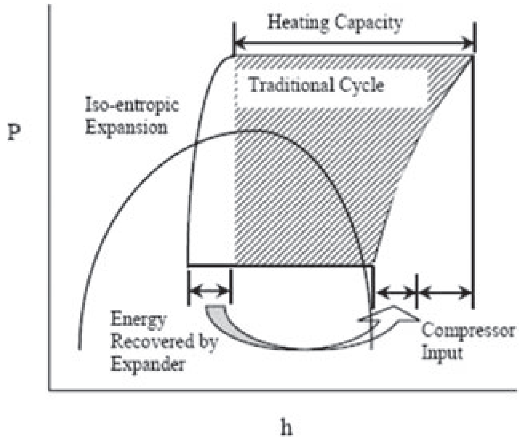

Figure 15. *Transcritical cycle for a carbon dioxide heat pump equipped with an expander device.*

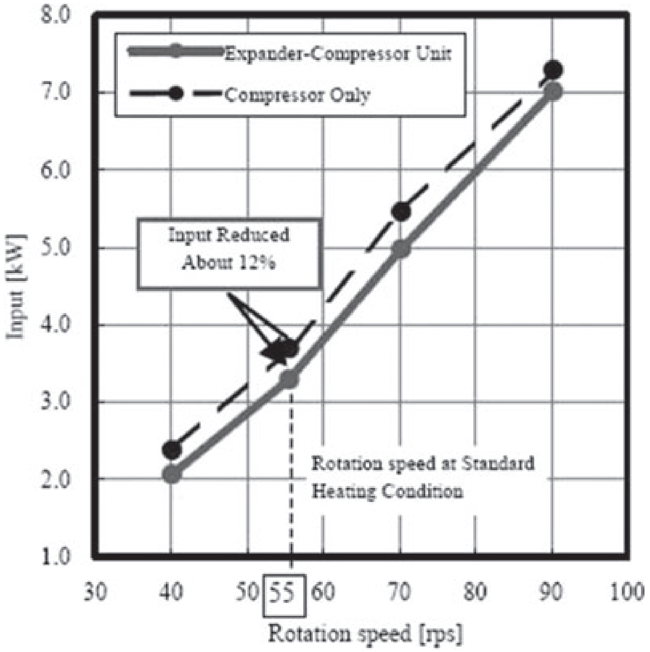

Figure 16. *Comparison between the energy input required by a heat pump with or without the expander.*

dioxide [5]. A specific cooling effect increment is obtained (Fig. 15). Moreover the produced useful work can lower the compression work by more than 10% according to the first experimental results, even if the expander efficiency is only 59% (Fig. 16).

Table 1. *Development specification (Microturbo S series)*

Type	MTSH175	
Operation mode	Cooling	Heating
Compressor type	MCM50S	
Capacity	175 kW	175 kW
Temp. of chilled and hot water	7ºC out/12ºC in	40ºC in/45ºC out
Outside air temperature	35ºC db/24ºC wb	7ºC db/6ºC wb
Power consumption	35 kW	43.75 kW
COP	5.0	4.0
COP of conventional machine	(2.7)	(3.2)
Control system	Variable speed control, inlet guide van control, fan control, sprinkling control	

For higher capacity applications the novelty of turbochillers operating with R134a must be considered. These were applied at first for the high capacities of district cooling with very high COPs, over 6. A recent application was magnetic levitation of oil free shaft supports. These innovative compressors were applied also to heat pumps with a power consumption of 35–45 kW and a capacity of 175 kW both for cooling and heating. Table 1 reports the performance both in cooling and in heating mode [6].

Problems regarding defrosting must be taken into account when dealing with air source heat pumps. As it is well known, the air temperature at the surface of the outside evaporator usually drops below dew point with liquid formation. When the temperature drops below 0ºC and the condensate is plentiful, frost is formed increasingly on the heat exchange battery. The frost operates a kind of thermal insulation for the heat exchange, moreover it thickens till clogging of the air passage occurs. It is then of paramount importance to identify the condition of defrosting. Hot gas is passed inside the external battery either by cycle reversing or by bleeding of hot gas from the compressor. Defrosting goes on for some minutes. In this period a cooling effect is produced in the room inside in the case of cycle reversing (the inside battery operates as evaporator) or no heating is produced. Furthermore, defrosting penalises heat pump efficiency: when COP is represented as a function of outside temperature a typical drop is found for outside conditions that require defrosting (Fig. 17). Therefore the accurate identification of defrosting requirement and its achievement is particularly important. The maximum frost formation takes place for outside temperatures between 5ºC and 0ºC. For lower temperatures the air specific humidity is small and it does not produce appreciable ice formations.

The required energy to operate defrosting is not negligible. Fig. 18 reports the percentage of energy required for defrosting for different working conditions for the outside air and the heated water temperature [7]. The bars refer respectively to defrosting by hot gas injection or cycle reversing. About 10% of the energy required

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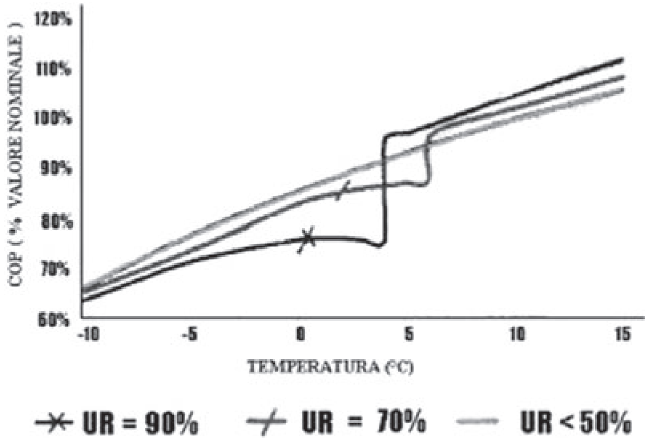

Figure 17. *COP ratio nominal Eurovent value (Tair = 7ºC, relative humidity 70%) keeping into account defrosting.*

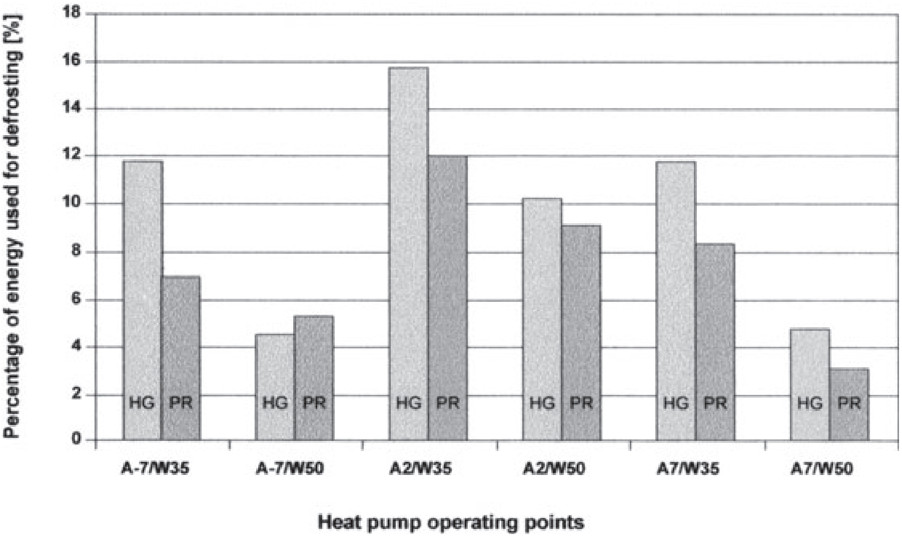

Figure 18. *Percentage of total electrical energy consumption of heat pumps required for defrosting (mean values for the heat pumps measured).*
Legend:
HG = hot-gas defrosting
PR = defrosting using cycle reversal
A-7/W35 = Air temperature of minus 7ºC and water temperature of 35ºC.

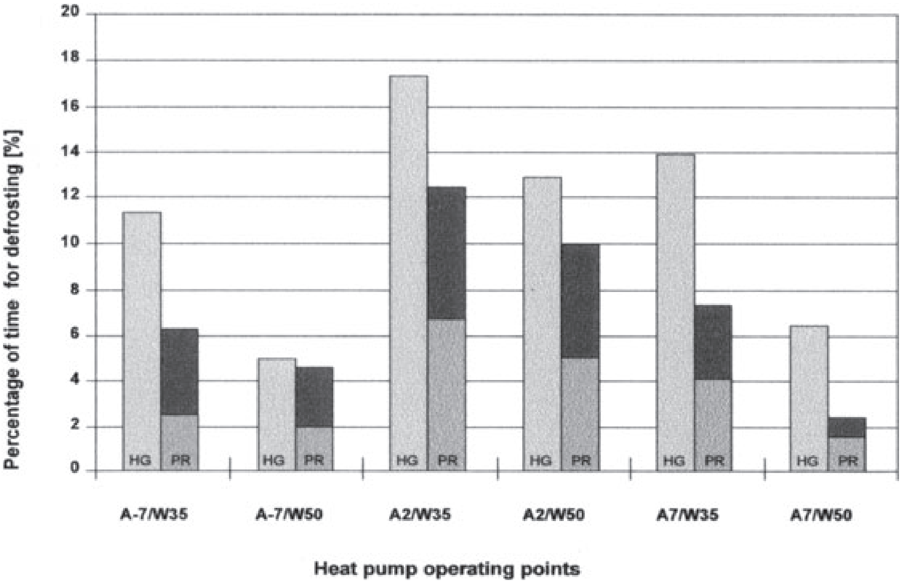

Figure 19. *Percentage of total effective heat pump operating hours used for defrosting (mean values for the heat pumps measured; HG = hot-gas defrosting, PR = defrosting using cycle reversal; for PR: bottom part is for actual defrosting process, and upper part is for heating up the heating system again).*

by the heat pump is due to defrosting with worse performance of the hot gas injection mode that is, however, less penalising for thermal comfort.

The analysis also shows the required time to provide defrosting (Fig. 19). It is a function of outside air conditions and it sometimes exceeds 10% of the heat pump total working period. In the case of cycle reversing mode the upper bar gives the time fraction required for heating up the heating system again.

A recent proposal to mitigate the energy request and thermal discomfort due to defrosting was produced by *Oak Ridge National Laboratory* [8]. The starting point is the survey that the frost formation is not significant if the surface of the outside battery is kept over −2.8ºC when the outside temperature is between 0ºC and 5ºC. A small thermal power is provided to the liquid accumulator before the compressor. For a heat pump whose capacity is between 10–15 kW the required thermal power is of about 1 kW. This heat (that returns as room heating effect) allows increased suction pressure and temperature with a delay in frost formation. Laboratory tests revealed that for the same working conditions frost develops after 100 minutes in the traditional system, whereas the proposed system requires defrosting after about twice as long, that is after around 3 hours. Moreover in the traditional system defrosting is 10 minutes long while the heat pump supplies about 10 kW (with an electric power of about 3.5 kW), whereas the proposed system defrosting is 12 minutes long but with a required electric power of only 1 kW.

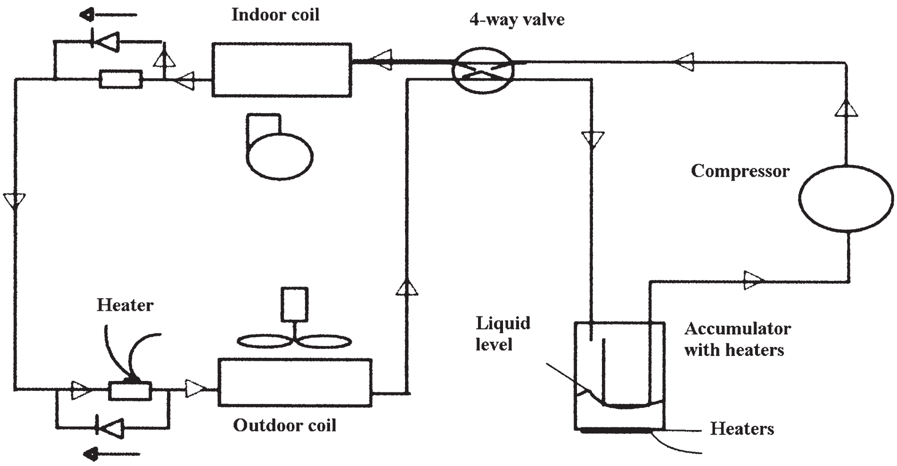

Figure 20. *Innovative system to delay the defrosting by a moderate heating of the liquid accumulator.*

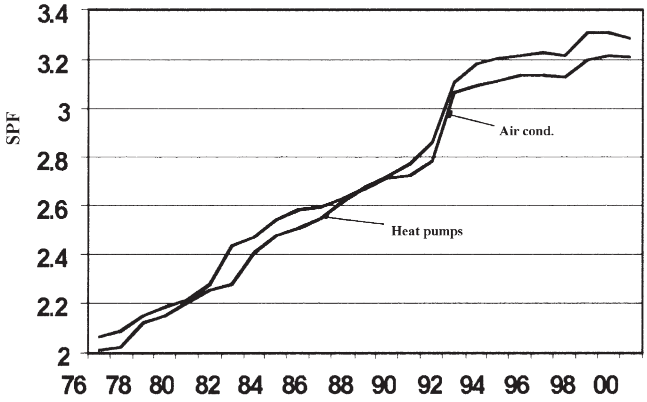

Figure 21. *SPF improvements during the last 25 years in heat pumps and air conditioners.*

A schematic of the circuit is represented in Fig. 20 where an electric heater is provided just before the evaporator and, as presented in the paper, in the liquid accumulator. Seasonal performance data that reports the heater operations to keep the evaporator over –2.8 ºC are not available so that seasonal performance advantages cannot yet be claimed, but the system appears really promising.

Beyond the different technological innovations described above, a picture is reported that illustrates the continuous SPF improvement both of heat pumps and air conditioners (Fig. 21). This improvement is due of course not only to the innovations of the equipment but also to the more favourable plant conditions.

5. Gas driven heat pumps

For a country like Italy where the electric installed capacity is scarce it is worthwhile considering the developments in the field of gas driven heat pumps, taking into account that natural gas is plentiful in summer. Gas driven heat pump can be classified as i.c. motor driven heat pumps with heat recovery and absorption heat pumps. Both allow high *Primary Energy Ratios* (PER), often much higher than for electric driven heat pumps (at least if electricity is generated with traditional thermoelectric plants with efficiency slightly higher than 40%).

As far as motor driven heat pumps are concerned, small capacity machines are available, with about 30 kW capacity. Motors are less than one litre displacement at 3 or 4 cylinders, water cooled with thermal recovery on the exhaust and the cooling water, mechanical power of about 7–8 kW. The great novelty is due to the maintenance intervals that arrive at 10,000 hours. The same interval is claimed for the lubricating oil change. This extension is surprising: it would be like a car changing the oil every 300,000 or more km!

Some improvements were made to the motor to increase its efficiency. Some manufacturers applied the *Miller* cycle that already some Japanese car producers use. The *Miller* cycle is used in supercharged motors and it is similar to the *Otto* cycle, except for the intake phase. This terminates at the *Bottom Dead Center* (BDC) in *Otto* cycle. Intake valve duration is from two degrees before *Top Dead Center* (TDC) until 70 degrees after BDC in *Miller* cycle, that is the intake valves remain open for around an additional 30º of crankshaft rotation beyond 'normal'. This delays the true compression phase of the mixture and a fraction of the intake charge is blown back out the intake valves. The charge loss and the volumetric efficiency reduction is more than compensated by the supercharging, usually obtained by a volumetric compressor (not a turbocharger): a screw compressor is usually used. Compression phase takes place after intake valve shut. The effective compression ratio is reduced from 10 : 1 to about 8 : 1. Also piston compression work is reduced. Motor specific power loss is low, while its operation is smoother with a lower production of NOx. Combustion process is better, so much the more because the charge is hotter than an intercooled supercharge.

The main point is that for the higher compression efficiency of a screw compressor compared to the piston compression the compression work is reduced by 10 to 15% with an improvement in overall motor efficiency of a few percent [9].

Improvements have also been made to the motor control system. The motor can operate at a variable speed so that heat pump capacity is modulated according to the demand. Defrosting is no longer necessary as outside air batteries are set just over the motor (Fig. 22). Finally the noise is comparable to a common boiler (55 dB(A) at 1 m). Some data can suggest the possible PER obtainable: a small capacity model gives 26.5 kW heat with a natural gas demand of 19.4 kW (LHV) and electric power of 0.7 kW for the auxiliaries. PER is then 1.32. The same device can operate as an air conditioner with a cooling effect of 22.4 kW (gas input 17.8 kW) with a cooling PER of 1.21.

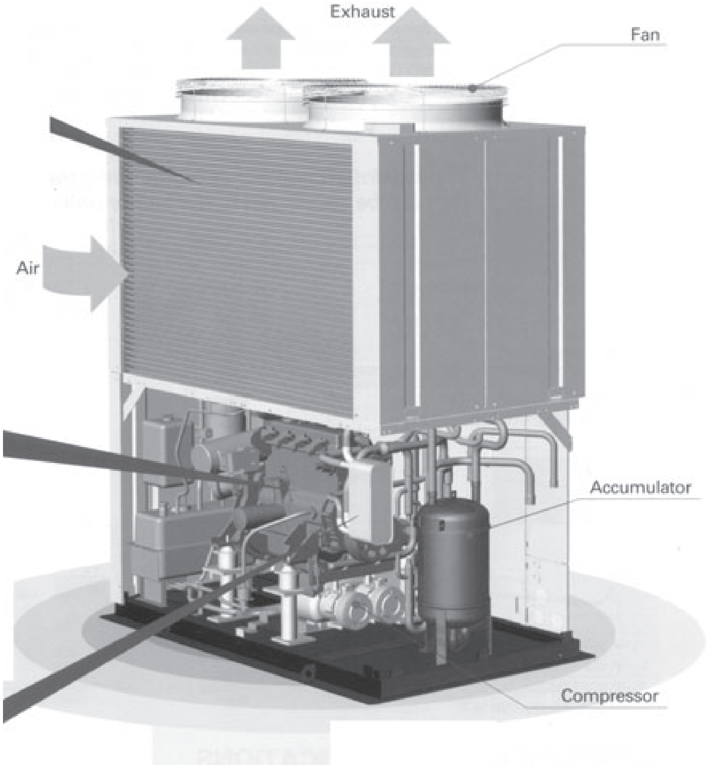

Figure 22. *Air to water motor driven heat pump (doc. Sanyo).*

Absorption heat pumps are the other group of gas driven heat pumps. Till now the absorption machine has been used above all as an air conditioner, though its operation is more suitable to heat pump mode. In fact even if only a block diagram of the cycle is considered (Fig. 23), it is easy to perceive that the energy taken from the evaporator can produce a useful effect during absorption. The energy cost is essentially a heat supply to the generator to separate the refrigerant from the solution. A second useful effect is then obtainable at the condenser, so that a PER of 2 could be theoretically obtained. Technically achieved PERs are of about 1.3–1.4, that is as high as in motor driven heat pumps. The advantage of the absorption heat pump lies in the few moving parts, then in its reliable and silent operation. Moreover, according to the thermal balance just presented, the thermal request to the heat pump cold source is well below (even at the same PER) than a conventional electric heat pump. This allows a smaller size of the outside air batteries and lower costs for cold source which are particularly expensive like the ground.

Two kinds of machines are on the market: water-lithium bromide and ammonia-water. The latter are more suitable to heat pump mode as evaporation can drop below 0ºC so that outside air is a possible cold source.

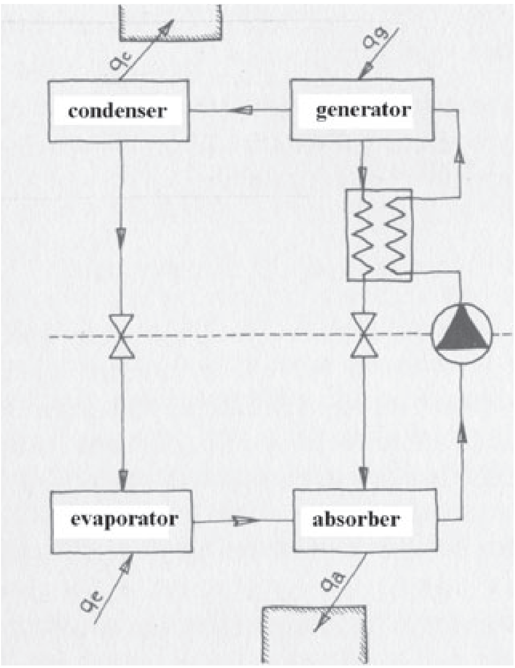

Figure 23. *Simplified scheme of an absorption heat pump.*

Moreover higher temperatures can be reached arriving even at more than 60ºC. In the past very few models were on the market. Long ago an absorption cooling machine was modified in Germany to operate as a water to water heat pump under the *Rekord* brand, but with bad results. Demonstration devices of higher capacity (250 kW) were also developed as the Dutch *Colibrì* heat pump that installed an absorption heat pump to heat a big public building at *Maastricht* in 1993, using as the cold source the near *Maas* river.

The major novelty in this field is the appearance on the market of a series of new absorption heat pumps in the range of 30–40 kW of the air to water type with summer-winter operation and water to water type with possible contemporary heating and cooling.

A schematic of the air to water heat pump is represented in Fig. 24. The presentation (much more effective in the animations in the company website [10]) can start from the generator, set on the left that can be identified from the side burner which develops ammonia vapours from the solution. The vapours are rectified and then passed to the condenser (set on the right) with a useful thermal effect. Liquid ammonia evaporates in the air battery all around the machine, taking thermal energy from the outside air. The vapour is directed by the 6 way inversion valve (set just the condenser aside) toward the absorber generator (in the middle of the figure). Absorption produces a strong thermal effect in a wide temperature range owing to

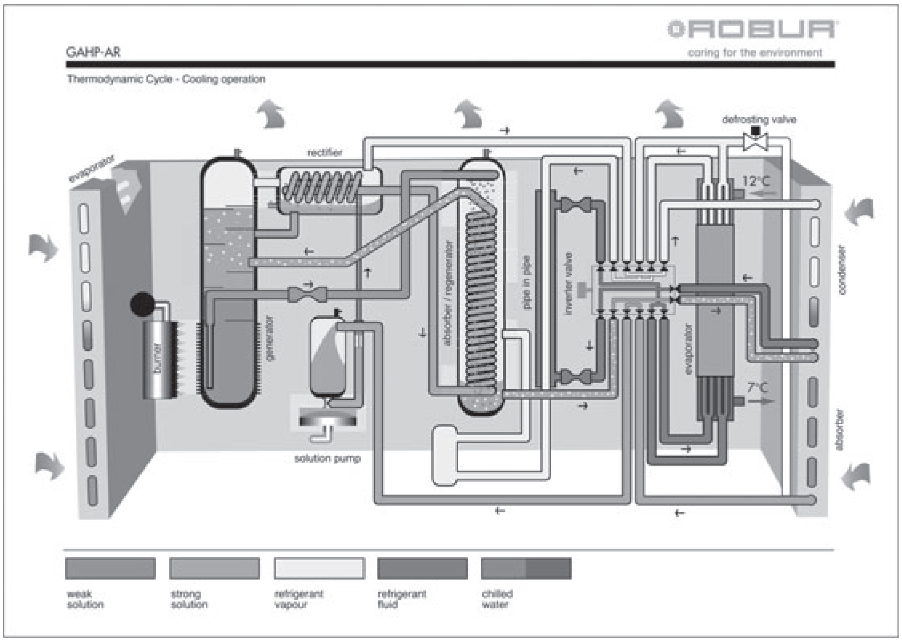

Figure 24. *GAX air to water absorption heat pump (doc. Robur).*

the strong concentration difference between rich and poor solution. This thermal effect lowers the energy supply the burner must provide. In fact the rich solution flows in the coil (the absorber inside where it is heated) before passing to the generator (GAX cycle – *Generator Absorber Heat Exchanger*). Final refrigerant absorption supplies a further thermal effect in the second absorber (the condenser aside). The ratio between the useful thermal effect and the LHV of the requested natural gas is of about 1.4, whereas the electric power for the solution pump and control panel is of 0.9 kW.

One of the novelties of the machine is the 6 way inversion valve. It allows the machine to operate also as an air conditioner with a cooling capacity of 17 kW and a 0.67 COP. The air battery can also be easily defrosted, directing the refrigerant vapour from the generator directly to the battery instead of to the condenser when defrosting is demanded.

The schematic of the water to water machine is represented in Fig. 25. This device can operate with cold source different from the outside air, like surface or ground water or the ground itself. Moreover the possible double heating and cooling effect could be exploited. Instead of the outside battery a heat exchanger is provided between the evaporating refrigerant and a hydronic circuit where refrigerated water for contemporary cooling and heating or cold water from one of the previously listed cold sources can circulate.

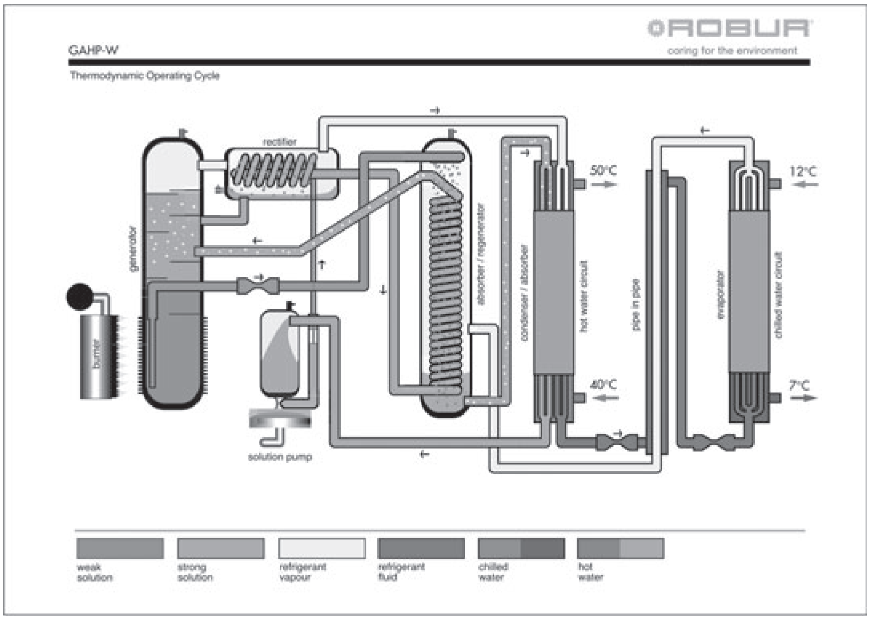

Figure 25. *GAX water to water absorption heat pump (doc. Robur).*

6. Heat pump systems progress

Probably the most important technological innovations in heat pumps concern the systems. Firstly low temperature heating systems are often realised, usually radiant systems that can lower to something more than 30ºC the heating fluid temperature. The lowering of the temperature has improved the COP well before other technological innovations. Secondly, cold sources different from outside air were selected. Four main possibilities should be mentioned:

- heat recovery on mechanical ventilation systems;
- water loop systems;
- surface or ground water systems;
- ground source heat pumps.

A short description of these possibilities follows.

As far as the heat recovery on the exhaust air is concerned, it can be operated with heat exchangers that can recover only a fraction of the available enthalpy, especially if the heat exchanger is only for sensible heat. Cold source is at an excellent temperature level and it is perfectly phased with the fresh air heating demand.

A contemporary heating and cooling demand in the same building gives excellent possibilities of energy savings: this happens more often than one can imagine, particularly in well insulated commercial buildings. A water loop supplies to many reversible heat pumps installed in the different zones of the building moderate

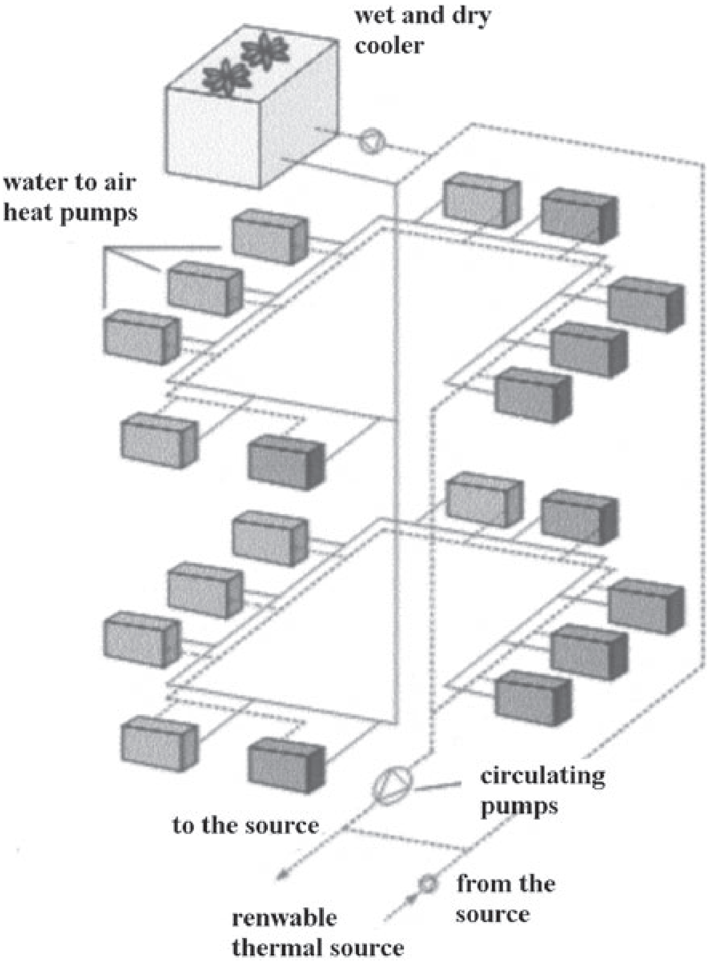

Figure 26. *Water loop plant with decentralised heat pumps operating both in heating and cooling mode. The ring is thermally balanced by an air cooling system and a thermal source that could be ground or surface or ground water.*

temperature water which is a good heat pump cold source but also a heat sink for the air conditioner (Fig. 26). Of course heating and cooling requirements are not always balanced: a water loop sometimes dissipates the excess heat (obviously in summer) and is sometimes lacking thermal energy: this must be supplied if possible by a free and renewable source such as the ground. Systems operating on surface or ground water are an excellent possibility both as regards constancy of the temperature levels during the year and of the temperature values. In the case of surface water utilisation it is not always necessary to directly pump the water: surface water heat exchangers can be used in the form of long and cheap HDPE coils as represented in Fig. 27.

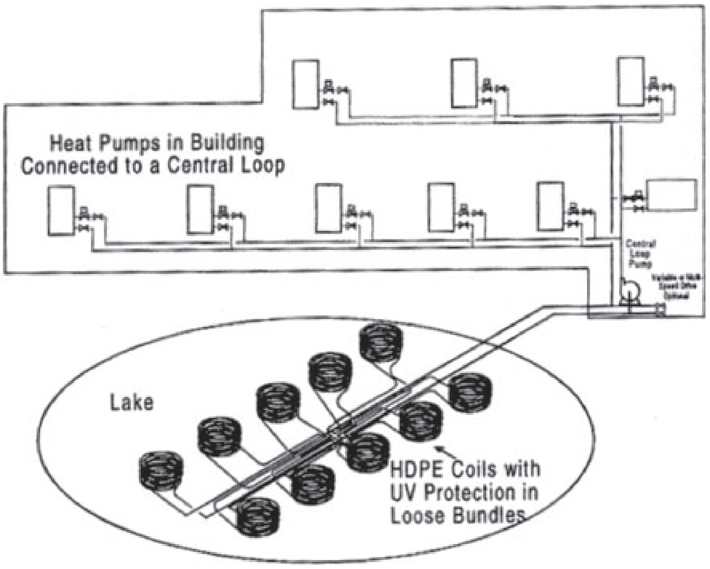

Figure 27. *Heat exchange system for surface water (a lake) and heat pumps.*

Finally ground coupled heat pump systems must be considered. The cold source is more and more often obtained by vertical tube heat exchangers that sometimes exceed 200 m depth. This subject is outside the purpose of this paper. It is important only to recall the thermal stability of the ground at higher depths and the possibility of using the system as a kind of a seasonal storage. For a size of 50 W/m of tube for the probes, a SPF (*Seasonal Performance Factor*) of 4.5 can be reached. This value can be improved by reducing the temperature drops from the ground to the heat pump. A lot of systems use a secondary fluid to take heat from the ground to the heat pump. Few systems (widespread in Austria) use direct expansion [11]. Recently a very innovative proposal provides vertical heat pipes, where the fluid is carbon dioxide (Fig. 28). This would eliminate the energy cost of the brine circulation and the hazard of the fluid leakage in the direct expansion systems. A hundred plants have been built up till now with a heat pipe that sometimes exceeds 65 m with a temperature drop between the two pipe sides of only 2ºC. A pipe as deep as 100 m is being experimented. This diversified picture of the state of the art heat pipe equipment and systems terminates describing a really innovative plant built up in Holland. The plant was set into operation in 2003: nine blocks with 382 40-year-old apartments were retrofitted for higher energy efficiency. The first action was better insulation regarding both the walls and the windows. Secondly a mechanical ventilation system with heat recovery was installed in each apartment. Thus heating demand was lowered, so that the existing heating radiator plant became oversized and it could operate at lower temperatures. The domestic hot water heating system of each apartment had to be replaced for safety reasons. A centralised domestic hot water

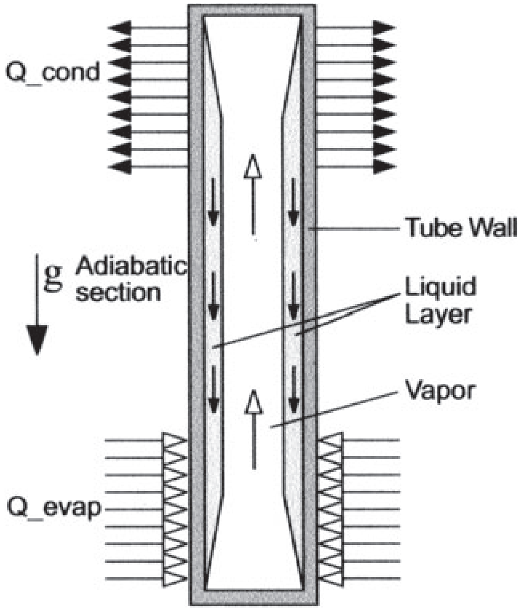

Figure 28. *Heat pipe ground probe.*

Figure 29. *Solar collector modules on the roofs of the apartment blocks.*

heating plant was selected for each block. The most striking part of the plant is an area of 2850 m^2 of flat plate solar collectors set on the roof of the 9 blocks, that is an area of 7.6 m^2 for each apartment (Fig. 29).

Solar collectors satisfy almost completely summer hot water demand, storing the excess heat in a central aquifer storage set at a depth of 115 m, connected to the plant by two wells. In winter the stored heat is used both for preheating sanitary hot

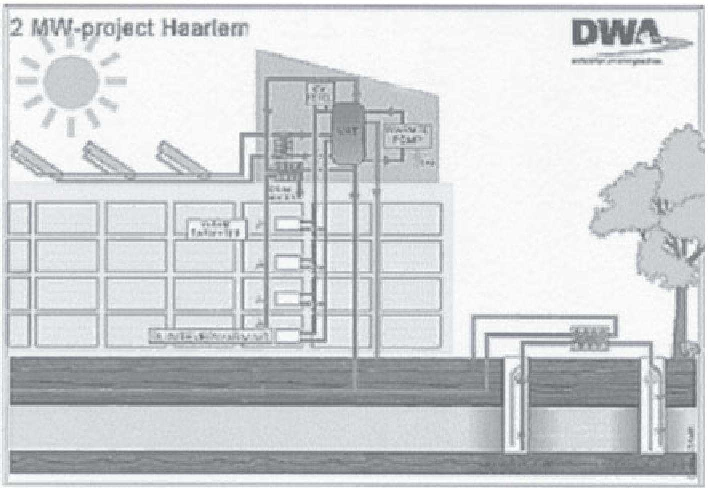

Figure 30. *Simplified scheme of the energy system.*

water and as the heat pump cold source. Each apartment block (about 40 apartments) is heated by two absorption heat pumps with an overall heating capacity of 76 kW. The water taken from the aquifer is the heat pump cold source. The storage can supply up to 45 m^3/h at a starting temperature of 45ºC at the beginning of the heating season.

A short term storage is also provided with a volume of 9.5 m^3 in each block. During sunny winter days, solar heat can be stored at a rather high temperature, which makes the heat usable for space heating and pre-heating of domestic hot water. A scheme of the energy system is represented in Fig. 30 with the two wells for connection with the aquifer on the right; for a single block solar section and short term storage are also represented. A 70% energy saving is anticipated. This project is expected to be the first in a series of similar projects in existing housing districts in Holland [12].

References

[1] G. C. Groff, ‘Heat pumps – who uses them and why?’, 8th IEA Heat Pump Conference, Las Vegas, 2005.

[2] H. Halozan, ‘Heat pumps and the environment’, 8th IEA Heat Pump Conference, Las Vegas, 2005.

[3] P. Hubacher, ‘Field Analysis of Heat Pump Installations – the FAWA Project’, *IEA Heat Pump Centre Newsletter*, 22(2) (2004), 15–18.

[4] I. Kilikis, ‘Advantages of combining heat pumps with radiant panel heating and cooling systems’, *IEA Heat Pump Centre Newsletter*, 11(4) (1993), 28–31.

[5] K. Sakitani, M. Moriwaki, M. Okamoto, E. Kumakura and T. Okamoto, ‘Development of a two-phase flow expander for CO_2 heat pumps and air conditioners’, 8th IEA Heat Pump Conference, Las Vegas, 2005.

[6] W. Seki et al., 'Development of a high performance turbo-chiller', 8th IEA Heat Pump Conference, Las Vegas, 2005.

[7] M. Zogg, 'Study of energy requirements for air-source heat pump evaporators: alternatives and efficiencies', *IEA Heat Pump Centre Newsletter*, 21(2) (2003), 19–21.

[8] V. C. Mei, Z. Gao and J. J. Tomlinson, 'The frost-less heat pump: new method of controlling the wintertime performance of air-to-air heat pumps', *IEA Heat Pump Centre Newsletter*, 23(1) (2005), 22–24.

[9] http://www.mazda.com.au/articleZone.asp?articleZoneID=92.

[10] http://www.robur.it/pag_prodotto_doc.jsp?idl=1&tipo_doc_prod_id=18&idp=10.

[11] H. Halozan, R. Rieberer, 'Direct evaporation ground coupled heat pump in Austria', *IEA Heat Pump Centre Newsletter*, 20(3) (2002), 28–32.

[12] H. Buitenhuis, '2 MegaWatt project. The most energy efficient heating system in a retrofit housing project', *IEA Heat Pump Centre Newsletter*, 21(4) (2003), 18–20.