The Swiss Retrofit Heat Pump Programme

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Abstract

Replacing boilers by a combination of cogeneration units with electric heat pumps can double the energy efficiency in producing low temperature heat. Even when replacing boilers in older dwellings by heat pumps and cogeneration units, a total energy performance factor of about 1.5 can be attained. However, this requires heat pumps with a smaller drop of heating capacity at higher temperature lifts, a lower compressor outlet temperature and a higher energy efficiency.

In order to meet these requirements the development of a new type of retrofit heat pump has been initiated by the Swiss Federal Office of Energy. This development leads to new cycles for small heat pumps: two stages with two compressors, vapour injection with an economizer, vapour injection with an economizer and a suction gas superheater, and a separate heat pump loop for the subcooling of the condensate. The experimental results of these cycles are presented and compared with those of a conventional heat pump cycle.

For HFC refrigerants a promising retrofit heat pump with vapour injection is now in the phase of field tests. A retrofit heat pump with ammonia as refrigerant and an adsorptive ammonia trap is still in the stage of a bench system.

Introduction

For reasons of energy saving and emission reduction, Switzerland has been promoting heat pump heating since 1937. As a result of coordinative efforts among manufacturers, installers, and customers, today 38% of the heating systems installed in new single-family houses are heat pump heating systems. However, in the much larger retrofit market the share of heat pumps represents about 2.5% only. In other countries of Western Europe the retrofit heat pump share is even much lower. This indicates a very large potential in the retrofit market. Every new domestic boiler that is installed to replace an old one represents a missed opportunity. In Western Europe alone, this happens a million times a year. Consequently higher CO_2 emissions are accepted than the state-of-the-art technology would produce. If only 10% of all the new gas or oil-fired boilers installed were replaced by heat pumps, additional primary energy savings equivalent to more than 84,000 tons of oil could be realised annually!

What are the obstacles? The older hydronic heating systems are characterised by high supply temperatures. Conventional heat pumps reach their limits if they have to provide the high temperature lifts and the high supply temperatures required in the retrofit market. Therefore, since 1998, the development of a retrofit heat pump, which meets the requirements of an older hydronic heat distribution system at a competitive price, has been the main priority of the Swiss Federal Office of Energy's research programme on the utilisation of ambient heat.

The basic concept for an efficient production of low temperature heat

In Switzerland a properly designed residential building requires normally no summer cooling. But there is a large heat demand for the space heating in wintertime. During the coldest days the outside temperature gets down to about–12°C in the most populated regions. More than 50% of the total Swiss end user energy is required for space heating. 86% of this heat demand is produced by boilers (57% oil, 26% natural gas, 3% wood). The heat distribution is almost exclusively done by hydronic systems with baseboard heating in newer buildings or with radiators in older ones.

A boiler always has some losses. Therefore a boiler produces less than 100% utilizable heat of 100% oil, gas or biomass energy input. This conventional solution of producing heat by boilers is a waste of exergy (availability, maximum theoretical work obtainable). In the flame of a boiler with a temperature of 1100 °C and with an assumed room temperature of 20°C the exergy rate is around 78.7 %. A boiler does not take profit of this high quality energy. If the supply temperature of a boiler is 40 °C the used rate of exergy is only 6.4 %. The boiler destroys 91.9% of the exergy! This "stone age" principle has to be replaced by the combination of cogeneration units and heat pumps (Figure 1). The cogeneration unit is usually more efficient and more economic if it is installed in larger buildings, such as schools, office buildings or hospitals. There the produced heat is used for space heating and tap water heating. All the electricity produced by the cogeneration units is transported by the public electric power grid to smaller residential buildings in the vicinity. There it is consumed by heat pumps, which have to work at the same time. These heat pumps produce the heat for space heating and hot tap water heating by using a large part of the ambient heat.

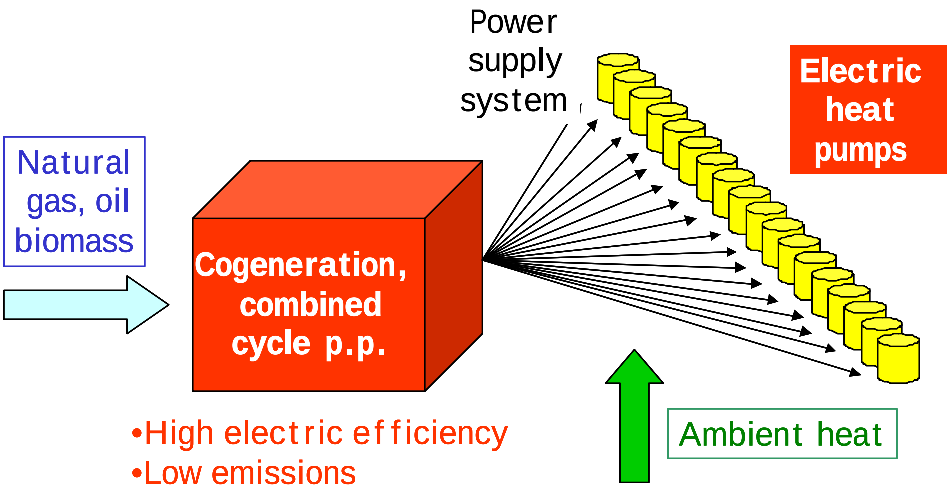

Figure 1: Basic concept of an efficient space heating system by the combination of cogeneration units and heat pumps (CG-HP-system).

With the same input of oil, gas or biomass, this solution leads to a much higher heat output than a conventional boiler, due to the utilization of the ambient heat. With a fuel energy input of 100% this arrangement produces between 150 % and up to 200% (or even more) of utilizable heat (total energy performance factor of 1.5 … 2.0), which is illustrated by a numeric example in Figure 2. By the way – about the same efficiency is reached by producing the electricity with a modern combined cycle power plant (without utilizing the waste heat). Of course in the future conventional cogeneration units with combustion engines can be replaced by fuel cell modules.

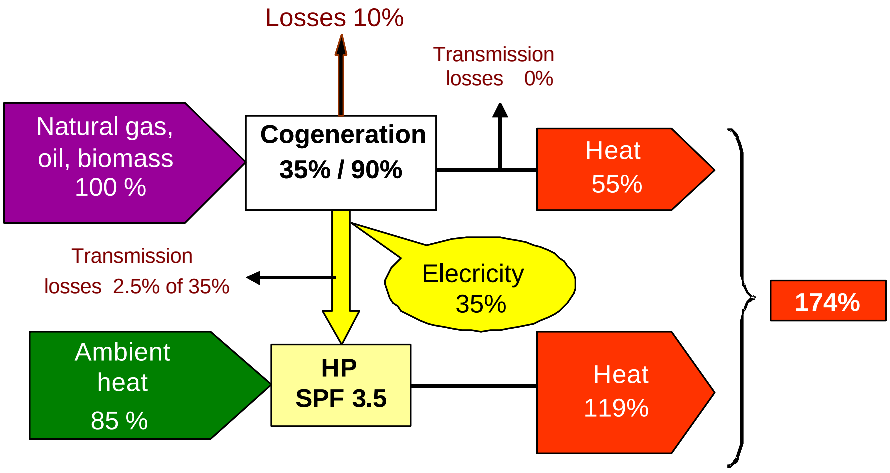

Figure 2: Example for the high efficiency of the combination of cogeneration units and heat pumps (CG-HP-system). Assumptions: Cogeneration with an electric efficiency of 35% and a total efficiency of 90%, transmission losses 2.5% of the electric energy, seasonal performance factor of the heat pump 3.5

The almost doubled efficiency of the CG-HP-system, compared to conventional boilers, is the reason why Switzerland makes a strong effort to replace boilers by this system [Zogg 1998, 2000]. This policy has been very successful in new buildings. As a result of coordinative efforts among manufacturers, installers and customers, 38% of the heating systems installed today in new single-family houses are heat pump heating systems. However, in the much larger retrofit market this number is as low as about 2.5%. In other countries of Western Europe, the heat pump share in the retrofit market is even much lower.

What are the obstacles for heat pumps?

The retrofit market heating systems are characterised by high supply temperatures and low thermal inertia of the existing heat distribution systems. In comparison to a conventional heat pump, a successful retrofit heat pump must have a smaller drop of heating capacity, a lower compressor outlet temperature, and a higher performance factor at higher temperature lifts. Furthermore a more sophisticated control strategy of the hat pump heating system is needed in order to minimize the size of the heat storage tank or to avoid it completely if possible. This has to be based on a dynamic model of the whole heat pump heating system including the heat source, the heat pump, the heat storage tank, the hydronic heat distribution and the building. This is dealt with in a separate paper [Shafai 2002].

Regarding the smaller heat pumps (with heating power up to 25 kW), sold during the year 2000 in Switzerland, 55% use air, 39% use vertical borehole heat exchangers, and 5% use water as a heat source. Drilling boreholes in a well-tended garden seldom elicits much enthusiasm. Therefore, for retrofit heat pumps air remains in most cases the only possible heat source. Furthermore heat pumps are only accepted if they provide comfortable room temperatures even in the coldest days without any backup heating system. The consequences in terms of efficiency can be taken from Figure 3. The second law efficiency η of a heat pump can be defined as the ratio of the coefficient of performance (COP) achieved in reality to the coefficient of performance of an ideal heat pump ("Carnot heat pump"). For an air-to-water heat pump with an inlet temperature of the ambient air T_i and the delivered supply temperature T_s this results in

$$\eta = \frac{COP}{\frac{T_s}{T_s - T_i}} \qquad (1)$$

When including all losses (fan, defrosting, control, additional pressure loss in the condenser compared to a boiler), the best commercial air-to-water heat pumps tested achieve a second law efficiency of about 0.4.

In the most populated lower areas of Switzerland the minimal temperatures can decrease to about –12°C. Unfortunately this happens when the heat pump has do deliver the highest supply temperature and the highest heating power. At these conditions modern baseboard heating systems require a supply temperature of about 35°C only. But older hydronic heat distribution systems with radiators require much higher supply temperatures. In order to be successful in the retrofit market, the heat pump has to guarantee a supply temperature of at least 60 °C at an ambient temperature of –12°C. The resulting temperature lift of 72 °C is quite a challenge for heat pumps. First of all in terms of energy efficiency. Figure 3 shows the dependence of the coefficient of performance (COP) on the supply temperature for a heat pump with a second law efficiency of 0.40. The worst case for –12°C/60°C with a COP of only 1.85 is marked with WC. The average outside temperature in the most populated areas of Switzerland is about 2 °C. Assumed a linear characteristic of the hydronic heat distribution system this gives an average supply temperature of 40°C. According to Figure 3, this operation point A (2°C/40°C) has a COP of 3.3.

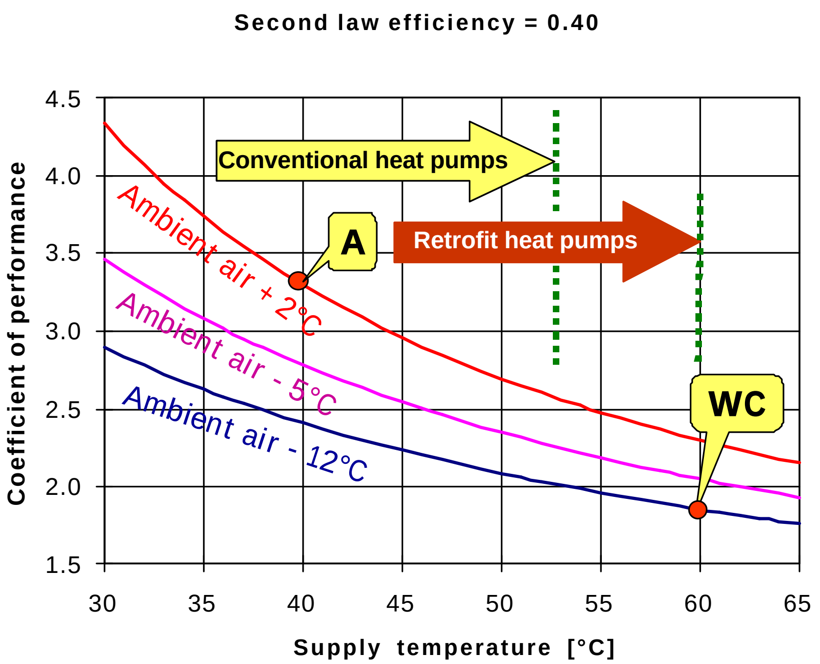

Figure 3: Coefficient of performance of an air-to-water heat pump as a function of the supply temperature for ambient air inlet temperatures of –12°C, -5 °C and +2 °C. Second law efficiency = 0.4. WC: worst case in wintertime, A: average operation.

For average weather conditions such a retrofit heat pump achieves a seasonal performance factor of about 2.7. This is a much lower value than the 3.5 supposed in Figure 3. But according to Figure 4, in the combination with the same cogeneration unit, it still leads to a total energy performance factor of 1.45. Even if the seasonal performance factor of the heat pump would decrease to 2.5, the CG-HP-system would still result in an energy performance factor of 1.34. This value is much higher than the one of a boiler which is about 0.95. Compared to a boiler the CG-HP-system would even in this worst case result in 30% fuel savings and 30% less CO_2 emissions. This is true for a state of the art technique. The potential for improvements is high, on the cogeneration siede as well as on the heat pump side.

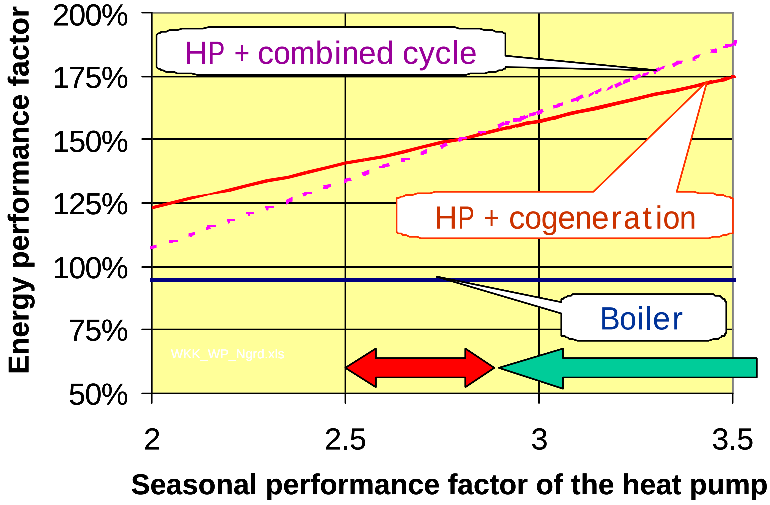

Figure 4: Dependence of the total energy performance factor on the seasonal performance factor of the heat pump. Data of the cogeneration and transmission as in Figure 2. For comparison (dashed line): production of electricity with a combined cycle plant with an electric efficiency of 58% and 7.5% electric transmission losses.

The second problem to get over is the dramatic drop of heating power with an increasing temperature lift of standard heat pumps. A heat pump without backup heating has to be dimensioned for the coldest days with the maximum heating power demand. This leads to a considerable oversizing for the average operation regime. Reducing this drop of heating power is essential for a retrofit heat pump. It results not only in cost reductions but also in longer periods of operation under part load conditions, and thus in a higher efficiency.

Last but not least in a simple one stage heat pump a high temperature lift leads to intolerable high compressor outlet temperatures for the majority of the interesting refrigerants. These outlet temperatures have to be reduced considerably.

The *Swiss Retrofit Heat Pump* project

While larger retrofit heat pumps are already available on the market (for example with economizer and screw compressors), optimal solutions do not yet exist for heating powers below 25 kW. In order to change this situation the *Swiss Federal Office of Energy* launched a competition “Swiss retrofit heat pump” in 1998 in order to develop a new type of retrofit heat pumps which would meet the following requirements:

1. Ambient air has to be used as heat source.

2. The heat pump must provide the required heating power at an ambient air temperature of –12°C and a supply temperature of the hydronic heating system of 60°C without any backup heating systems.

3. The retrofit heat pump has to produce hot tap water at 55°C as well.

4. For high temperature lifts the retrofit heat pump needs to have a considerable smaller drop in heating power than the heat pumps tested so far.

5. For the highest temperature lifts from –12 °C ambient temperature up to 60 °C supply temperature the compressor outlet temperature has to be kept below 85 °C.

6. For ambient air as heat source the second law efficiency with all losses included (according to the standard EN 255) has to be maintained above 0.375 under all operational conditions and above 0.425 for the test point at air 2°C to water 50°C.

7. Natural refrigerants are preferred.[1]

8. The liquid refrigerant hold-up has to be kept at a minimum.

9. The control system has to assure a minimal volume of the heat storage tank.

10. The heat pump has to fulfil the requirements of the DACH quality label as well as all European standards.

To assist the Swiss manufacturers of heat pumps a number of research projects were initiated by the *Swiss Federal Office of Energy*. These were aimed at thermodynamic challenges and control issues. They have been worked out in close cooperation with the manufacturers, several university institutes and the Swiss Heat Pump Centre. Some problems are still under investigation.

The topics of the research projects were new cycles for heat pumps with a heating power below 25 kW in order to fulfil the special requirements for retrofit heat pumps with a high temperature lift as mentioned above: smaller drop in heating power, lower compressor outlet temperature and higher efficiency. Figure 5 gives an overview of the cycles optimised by computer simulation, built, measured in the laboratory and finally – the most promising – tested in real installations.

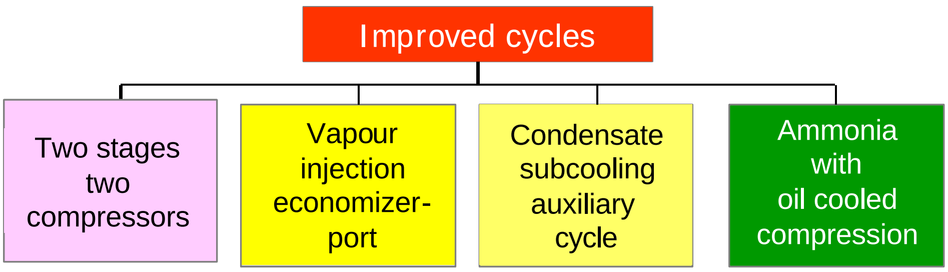

Figure 5: Heat pump cycles for small retrofit heat pumps (< 25 kW heating power) investigated in the *Swiss Retrofit Heat Pump* programme

The thermodynamically most promising solution to the retrofit problem is a two stage heat pump with two compressors as shown in Figure 6. This cycle has been built and investigated [Zehnder et al. 1999]. Compared to a simple one stage cycle it attained a 50% increase in heating power and a 14% increase of the COP at the highest temperature lifts. But it turned out that the oil migration in the circuit prevented a proper lubrication of both compressors after a few hours of op-

[1] Originally HFC were excluded. However severe restrictions by important producers of suitable hermetic compressors forced the SFOE to accept HFCs as refrigerants for a first phase of realization.

eration. Furthermore a heat pump of this type is too complex to compete with the much simpler boiler, which a retrofit heat pump should replace.

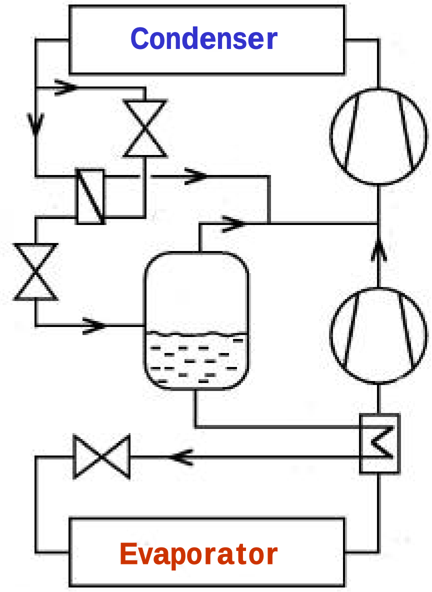

Figure 6: Heat pump cycle with two compressors and an intermediate injection. Reciprocating compressor in the first and scroll compressor in the second stage. R407C as refrigerant [Zehnder et al. 1999].

The cycle with economizer and vapour injection as shown in Figure 7 is a much more simple and cheaper solution. This cycle is well known from larger heat pumps with screw type compressors: a part stream of the condensate is expanded to a middle pressure level. The created liquid-vapour mixture is then brought to saturation by subcooling the rest of the condensate and is injected into the compressor. This cycle has the following advantages:

1. Higher mass flow rate at the compressor outlet → higher heating power.

2. Reduction of the compressor outlet temperature → meeting the temperature limits of the compressor.

3. Subcooling the condensate → increasing the COP, if a suitable compressor is available.

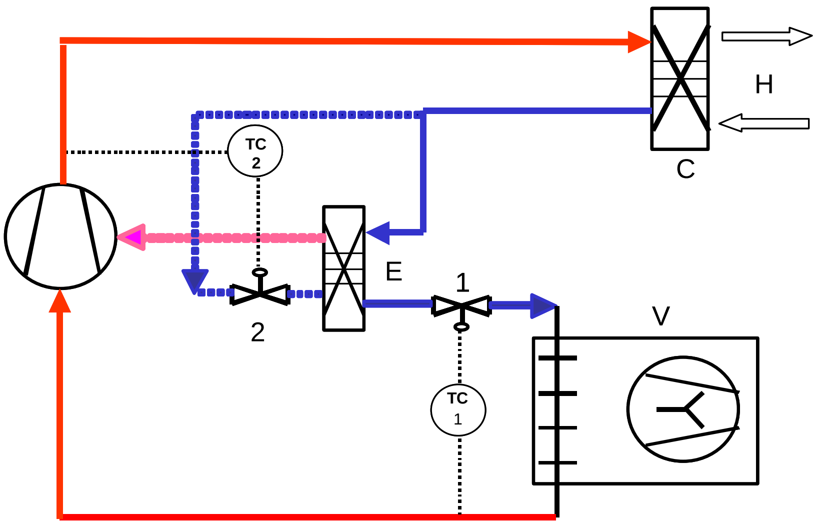

Figure 7: Heat pump cycle with economizer/vapour injection. Commercial scroll compressor with liquid injection port; R407C as refrigerant. C condenser, E economizer, V evaporator, H hydronic heating system, 1 main expansion valve, 2 expansion valve for the injected stream.

While the advantages 1 and 2 are obvious, the advantage 3 results from two opposite effects. With a higher injection rate the isentropic efficiency of the compressor decreases (→ lower COP). But at the same time the subcooling of the condensate by the economizer is intensified (→ higher COP) [Vaisman I.B. 2000]. Depending on the compressor characteristic at an optimal injection rate the subcooling effect can dominate. Thus the COP of the whole process can be increased. Unfortunately for heating powers below 25 kW there exists no compressor suitable for this process on the market. Therefore the best available scroll compressor with a liquid injection port was used. Of course, the pressure drop of this compressor is too high for vapour injection. Thus the improvement of the COP was only minor. The tests have been done with R407C as refrigerant. At the highest temperature lift (–12°C/60°C) an improvement of the heating rate of 15% was measured [Zehnder et al. 2000]. The compressor outlet temperature could easily be kept below 85 °C. But there is no simple answer to the influence on the COP of the heat pump cycle. As demonstrated with the next cycle a proper dimensioning of a vapour injection port would lead to a substantial enhancement of the COP. The retrofit market for smaller heat pumps definitely needs a compressor with optimal vapour injection design!

For this reason in a subsequent research project a prototype scroll compressor with an injection port optimised for the vapour injection has been used. Supplementary a suction gas superheater was inserted (Figure 8). With this prototype heat pump the following improvements compared to a simple single stage cycle were attained at high temperature lifts: increase of the heating power by up to 30%, increase of the COP up to 15% (measured at –7°C/60°C) [Brand et al. 2000]. These results are very promising. Hopefully this new type of scroll compressor will become available on the market soon!

A further research project [Reiner et al. 1998] investigated a cycle with a separate auxiliary heat pump loop. This uses the condensate subcooling as heat source and delivers the heat at the loop of the hydronic system coming from the main heat pump (Figure 9). This prototype heat pump has been tested with R407C and R417A (Isceon 59). For high temperature lifts an increase of the heating power by up to 20% and of the COP by up to 5% compared to a simple one stage cycle was achieved. With R417A there were no problems with the compressor outlet temperature. R407C lead to a too high compressor outlet temperature and is therefore not suitable for this cycle in retrofit applications. For a small heat pump this cycle is rather complex. But it should be taken into account for larger heat pumps.

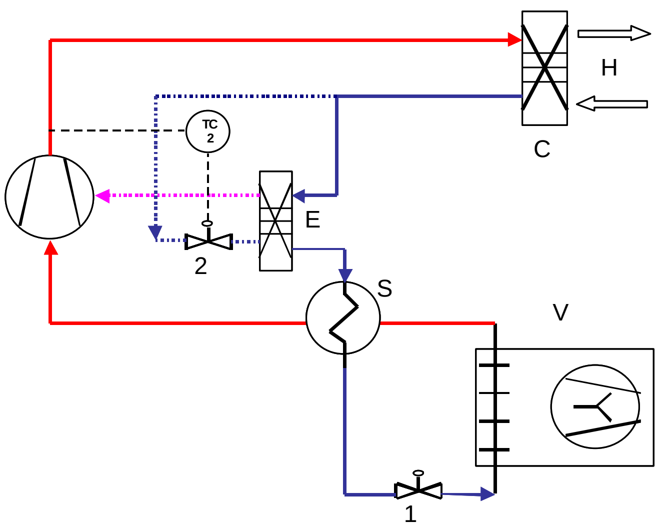

Figure 8: Heat pump cycle with economizer/vapour injection and suction gas superheater. Prototype scroll compressor with vapour injection port; R407C as refrigerant. C condenser, E economizer, V evaporator, H hydronic heating system, 1 main expansion valve, 2 expansion valve for the injected stream.

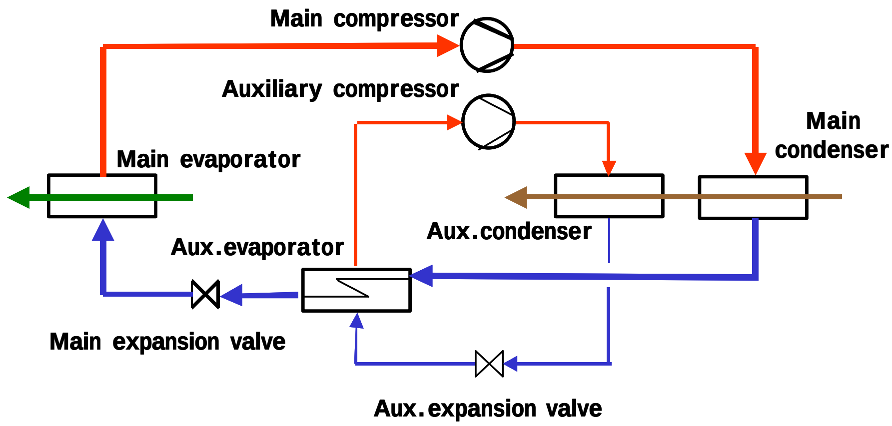

Figure 9: Heat pump cycle with a separate auxiliary heat pump loop for subcooling the condensate of the main heat pump loop [Reiner et al. 1998]. Refrigerants R407C and R417A; main heat pump with a scroll compressor, auxiliary heat pump with a reciprocating compressor.

Finally another project to develop a small retrofit heat pump with ammonia as a natural refrigerant was started [Kopp et al. 2000]. In order to tackle the enormous superheating problems connected with the low specific heat capacity of ammonia a rotary vane compressor was chosen with a very high oil flow rate in order to cool the ammonia during the compression. After surmounting a lot of hurdles the prototype is running successfully even at the highest temperature lift of -15°C/65°C. The first experiments show promising COP-values and no problems with the compressor outlet temperature. To deactivate any eventual ammonia leaks a special adsorption type casing has been developed. This pioneer project needs still a lot of efforts before being commercialised. Table 1 gives an overview of the results of the different approaches for a retrofit heat pump.

Table 1: *Comparison of the processes investigated for retrofit heat pumps. Heating power and COP: Improvements compared to conventional single stage heat pump circuits at high temperature lifts.*

	Two stage with two compressors	**Economizer**	**Economizer and Suction gas superheater**	**Separate HP-loop for condensate subcooling**	**Economizer and oil cooling**
Figure	6	7	8	9	
Refrigerant	R407C	R407C	R407C	R407C, R417A (Isceon 59)	Ammonia
Compressor	1st stage Reciprocating 2nd stage Scroll	Commercial Scroll with liquid injection port	Prototype scroll with vapour injection port	Scroll (main loop) Reciprocating (aux. loop)	Rotary vane compressor
Heating power up to	50%	15%	30 %	20%	Comparable
COP up to	14%	Insignificant	15 %	5%	Comparable
Compressor outlet temperature	Unproblematic	Unproblematic	Unproblematic	To high with R407C	Unproblematic

Conclusions

Retrofit heat pumps with a heating power up to 25 kW can be built according to the cycles of the Figures 7 and 8 with commercially available scroll compressors. If the defrosting and the hot tap water heating are managed efficiently, they already represent good retrofit solutions . In Switzerland, such a retrofit heat pump has been developed and thoroughly tested. It should be commercially available in 2003. The cycles with economizer and vapour injection would become significantly more efficient when scroll compressors with ports optimised for vapour injection would become available. Attracted by the huge retrofit market potential, the compressor manufacturers will hopefully act soon!

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