REVIEW OF AIR-SOURCE HEAT PUMPS FOR LOW TEMPERATURE CLIMATES

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ABSTRACT

Air source heat pumps are widely used for heating because of the relatively low installation costs. A major disadvantage is that the heat output and COP decrease for low outdoor temperatures, when most heat is needed. This paper presents the results of an extended literature and patent review of air source heat pump concepts for low temperature climates as experienced in the northern parts of the United States (excluding Alaska) and the southern parts of Canada. The heat pump concepts can be characterized by cycle improvements to lower the discharge temperature of the compressor and increase the COP and heat output at low outdoor temperatures. The advantages and disadvantages of several concepts shown in different papers are presented and the thermodynamic performances are compared by simulation. In addition, results of an investigation of refrigerants suitable for low temperature climates are shown.

Key Words: ***heat pump, low temperature, Nordic climate, high efficiency, comparison***

1 INTRODUCTION

Heating requirements in northern climates, such as climates in Minnesota, South and North Dakota, and Manitoba, can represent a challenge, especially for air source heat pumps, since the ambient air temperature in this region can reach values as low as -30°C during the heating season. Nevertheless, most heat pumps for space heating in this area are still air source heat pumps. Their main advantages are an easy installation and low installation costs. The disadvantages are that the performance of the heat pump decreases towards lower outdoor temperatures, when most heat is needed. The four main problems at very low ambient temperatures for heat pumps are:

a) Insufficient heat output as the required heat is the largest whereas the heat pump capacity is reduced due to lower mass flow rates of the refrigerant
b) High compressor discharge temperature caused by the low suction pressure and high pressure ratio across the compressor. This leads to bivalent heat pump systems which use the heat pump at medium ambient temperatures. At very low ambient temperatures the compressor is turned off to protect it from overheating and electric resistance heating is used instead. The combined efficiency of such systems is quite low.
c) The coefficient of performance (COP) decreases rapidly for high pressure ratios which can be found at low ambient temperature conditions.
d) If the heat pump is designed for low temperature it will have a capacity that is far too large at medium ambient temperature conditions. Therefore, the heat pump needs to cycle on and off at

higher temperature ambients in order to reduce its capacity, which leads to a lower efficiency of the system compared to steady-state performance.

Several concepts have been proposed to address these problems and design heat pumps for low temperature climates. However, no papers were found which summarize and compare the advantages and disadvantages of the concepts proposed thus far. The concepts proposed include for example, two-stage compression systems with either intercooling or economizing, systems which inject two-phase refrigerant into the compressor to decrease the compressor discharge temperature (*Zogg 2002)* and systems which use high oil flow rates to cool the compressor. In addition, cycles which use the subcooling of the refrigerant after the condenser outlet (*Zogg et al. 1999b),* cascade systems, and heat pumps with two parallel compressors *(Hwang et al. 2003)* have been proposed to increase capacity. As will be shown later, each of these systems has its advantages and disadvantages.

Not only is the selection of the appropriate cycle important, but also the selection of the appropriate refrigerant. Therefore, several refrigerants were investigated for their application in heat pumps for low ambient temperatures.

2 PRESENTATION OF DIFFERENT REFRIGERATION SYSTEMS

This paragraph lists and compares different concepts of heat pump cycles which are well suited for northern climates. In each case, only the main idea of the concept is presented and not the complete system. This means that there are possibilities to enhance the cycles by combining some of the ideas or by adding several components in one concept, such as internal heat exchangers, oil separators, reversing valves, etc.

1) Single-stage heat pump

It is possible to manufacture single-stage heat pumps even for climates with ambient temperatures as low as -30°C and condensing temperatures as high as 60°C. There are some compressors available (especially for the refrigerants R-507 and R-404A), which are designed for low temperature refrigeration applications and do not overheat even at large pressure ratios. The estimated COP is low compared to the other concepts which will be presented in this paper. In addition, it is difficult to include measures, such as a suction-to-liquid line heat exchanger, to improve the cycle performance since the discharge temperature would get too high. Furthermore, a system designed for this application shows decreased performance at higher ambient temperatures since the compressor is not designed for higher evaporating pressures. Another problem is that the heating capacity of the heat pump increases at higher temperatures while the heating demand decreases. This leads to cycling of the heat pump in short intervals, which reduces the efficiency of the system. Other possibilities include the use of variable speed drives, parallel compressors or variable displacement compressors, or a thermal storage system. These measures raise the price of the single-stage system and abolish the biggest advantage of this heat pump, the low installation costs.

In addition to the subcritical cycles with conventional refrigerants, the transcritical single-stage cycle using CO_2 as the refrigerant has received increased attention. However, it could be shown that this cycle is difficult to realize for low temperature applications unless special measures are taken, since the discharge temperature increases rapidly as the evaporating temperature decreases. In addition, there are currently no compressors commercially available which serve the whole operating range.

2) Two-stage heat pump with intercooler

A schematic of a two-stage cycle with intercooling is shown in Fig. 1. Such cycles are used for large temperature differences between evaporating and condensing temperatures. The cycle can use one two-stage compressor or two single-stage compressors. The advantage of the latter concept is that the compressors can also be run independently to reduce the capacity of the heat pump. By cooling the refrigerant between the two compression stages, the discharge temperature of the second stage can be lowered. This extends the operation of the heat pump over a wider temperature range. One of the main problems of using an intercooler which is cooled with return heating water is that the intercooler temperature is quite high and therefore, the refrigerant vapor exiting the low pressure-stage compressor can not be cooled down to saturated vapor conditions, which results in a less than optimal COP. One possibility to increase the efficiency of the system is to add a suction line heat exchanger before the inlet of the low pressure-stage compressor. This measure increases the heat emission at the intercooler and also raises the COP since the impact of the additional superheating on the power input of the compressor is small as can be see in Fig. 2. Figure 2 indicates the cycle state points of a two-stage heat pump with intercooling in a pressure enthalpy diagram. The gray line indicates a cycle with suction line heat exchanger and the black line one without. Studies by several authors *(Zehnder and Favrat 2000*, *Zogg 1999)* showed that the oil migration in such systems is a critical issue, especially if only one compressor is running. Therefore, the usage of an oil separator or other measures to ensure even oil distribution between both compressors is recommended.

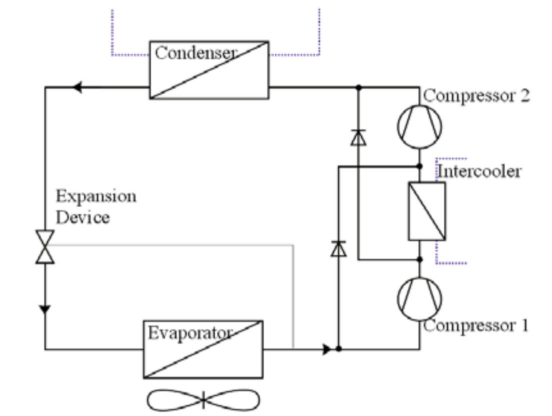

Fig. 1. Schematic of a 2-stage cycle with intercooler and w/o suction line hx

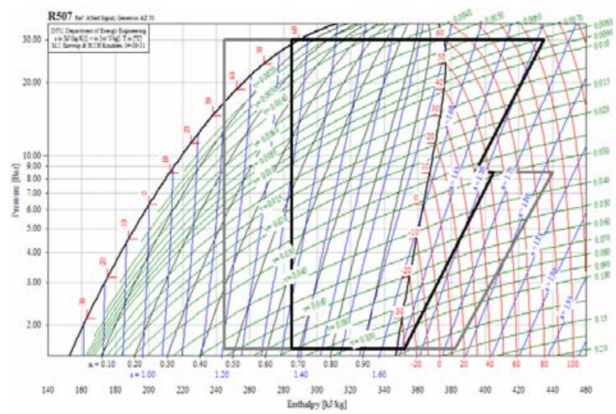

Fig. 2. p-h-diagram for the 2-stage cycle with intercooler (with and without suction to liquid line hx)

3) Two-stage heat pump with economizer

Instead of using an intercooler, a two-stage cycle may be operated with a closed economizer as shown in a simplified schematic in Fig. 3. Figure 4 shows the cycle state points of a two-stage heat pump with economizer in a pressure enthalpy diagram.

The economizer delivers a certain amount of two-phase refrigerant to a mixing chamber in the suction line of the high pressure-stage compressor where it is mixed with the hot discharge gas of the low pressure-stage compressor. This measure allows very accurate control of the inlet state of the high pressure–stage compressor. Therefore, no problems with the compressor discharge temperature should be expected. In addition, the economized refrigerant subcools the remaining condensed refrigerant before it enters the expansion valve, which improves the system COP. Instead of using a closed economizer, an open economizer could be used *(Zogg 1999)*. Furthermore the cycle could be equipped with a suction line heat exchanger and/or an oil separator and bypass valves so that it is possible to operate each compressor separately in order to reduce heating capacity and improve system performance at higher ambient temperatures. A variable speed drive for one of the compressors on this system could achieve even better performance since the intermediate pressure could be adjusted more easily.

This two-stage cycle with economizer shows a very wide variability and results in one of the highest possible efficiency compared to the other refrigeration cycles presented in this paper. The disadvantages are oil management, higher installation costs and more challenges to reverse the cycle for defrosting or air conditioning use. There are some patents available which address some of the problems discussed *(Shaw 2001, 1999, 1986)*.

Transcritical two-stage cycles with CO_2 as the working fluid are also very interesting possibilities which have received increased attention in the last few years. The main drawback of the transcritical CO_2 cycle is the lower efficiency caused by high return water temperatures, which are needed in most northern US heating systems. High water return temperatures into the gas cooler of the transcritical CO_2 cycle leads to a very unfortunate shape of the cycle in the p-h diagram. Studies presented by (*Cecchinato et al. 2003)* and (*Arlien et al. 2002)* list the advantages and disadvantages for heat pumps using CO_2 as the refrigerant.

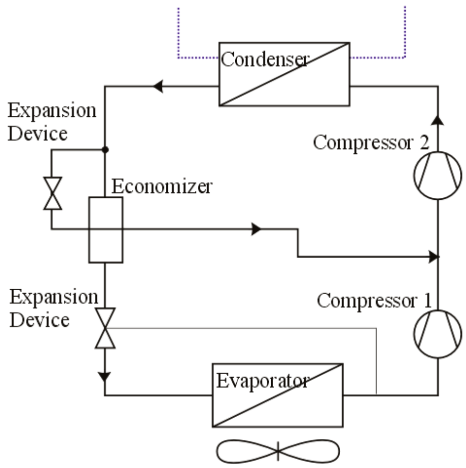

Fig. 3. Schematic of a two stage cycle with economizer

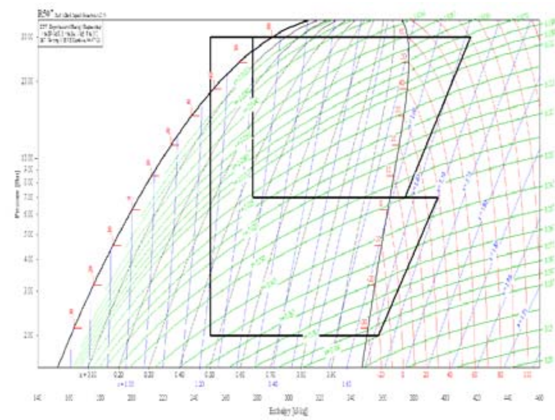

Fig. 4. p-h-diagram for the 2-stage cycle with economizer

4) Cascade cycle

Cascade cycles have been used in many different refrigeration applications for a very long time. The main advantage of the cascade cycle over other two-stage cycles is that two different refrigerants can be used whereby every refrigerant works at its optimum design range. Most low temperature refrigerants have disadvantageous properties at high temperatures and vice versa. This disadvantage can be eliminated by using two different refrigerants. Whereas the COP at high pressure ratios is very good it decreases for higher ambient temperatures due to the temperature gap in the intermediate heat exchanger and the lower pressure ratio over the compressors. The efficiency of a compressor decreases rapidly when the actual pressure ratio falls below the compressor design pressure ratio. Another disadvantage is that it is more difficult to reverse the cycle or run in part load since the system is not able to run one compressor alone. A possible solution to this issue would be to add an additional outdoor-evaporator for the high temperature stage which could be used in single-stage mode. The cascade system shows altogether very good performance and leaves a lot of room for further improvement. Good efficiency and control of the heat output are able to compensate in some cases for higher installation and maintenance costs.

5) Refrigerant injection at intermediate pressure

Instead of using a two-stage compressor where two-phase refrigerant can be injected between the two stages in order to cool the suction gas before it enters the high pressure stage, it is also possible to inject refrigerant at an intermediate pressure during a single-stage compression process. This measure is only possible for a few special compressor types. Examples for such compressors are screw or scroll compressors. However, screw compressors are not used in residential heat pumps, where scroll compressors are most common. There are some scroll compressors on the market which exhibit an intermediate inlet port. However, such compressors are normally designed for injection of single-phase gas and not designed for injection of two-phase refrigerant. (*Trüssel et al. 2000)* and *(Shuangquan et al. 2002)* worked successful on a heat pump design with two-phase refrigerant injection into the compressor. Trüssel et al. also redesigned a compressor for intermediate two-phase injection. The advantage of this cycle is that it extends the operating range of the heat pump to lower ambient temperatures and in addition, raises the heating capacity, whereas the efficiency of the cycle stays unaffected. The cycle can also be combined with other measures to improve the efficiency and raise the heat output, such as a suction line heat exchanger or an additional refrigeration cycle as presented in section 7 *(Zehnder et al. 2002b)*. Figure 5 shows a schematic of a refrigeration cycle using a compressor with intermediate injection and Fig. 6 presents the cycle in a pressure-enthalpy diagram.

Extensive field measurement of heat pumps with refrigerant injection (*Gabathuler et al. 2002)* showed that such systems did not only work perfectly in test chambers but also in the field.

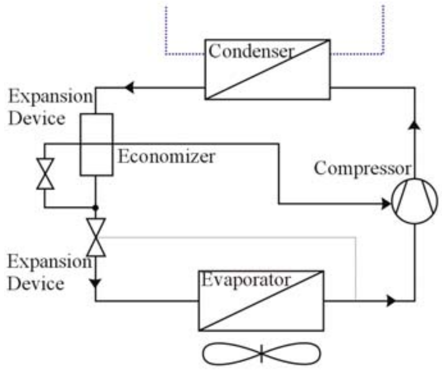

Fig. 5. Schematic of a single-stage cycle with refrigerant injection

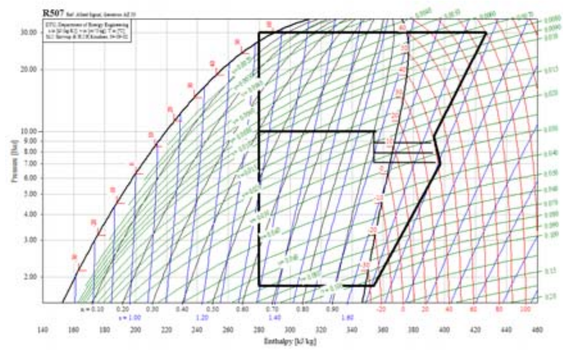

Fig. 6. p-h-diagram for a single-stage cycle with refrigerant injection

6) Oil cooling

Another method to cool the refrigerant during the compression process is to use a large amount of oil during the compression process, which is separated in the discharge line, cooled down in the condenser or in a separate oil cooler and then, returned to the oil inlet port of the compressor *(Michiyuki 2001)*. A lower compressor discharge temperature reduces the compressor input power and in addition, allows the cycle to operate at lower suction pressures. On the other hand, if the oil temperature is higher than the evaporation temperature, this measure raises the superheat, which can increase the compressor power consumption and decrease the COP. In addition, the compression volume of the compressor and therefore, the capacity will be reduced by the amount of oil flowing through the compression chambers, and by the higher specific volume of the refrigerant at the compressor inlet caused by the larger superheat. Altogether, if the cycle is designed properly, the efficiency increases and the operating range can be extended to far lower ambient temperatures without increasing the installation costs very much. Figure 7 shows a schematic of the single-stage cycle with oil cooling and Fig. 8 presents the cycle in a pressure-enthalpy diagram.

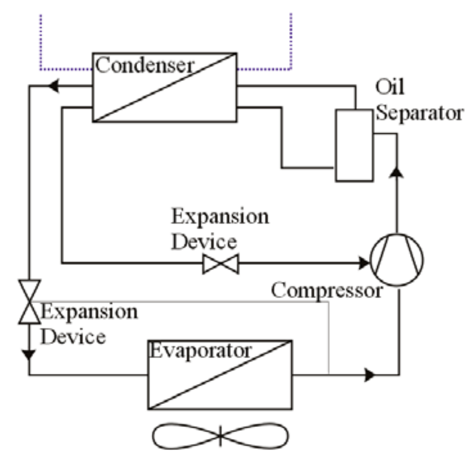

Fig. 7. Schematic of a one-stage cycle with oil cooling

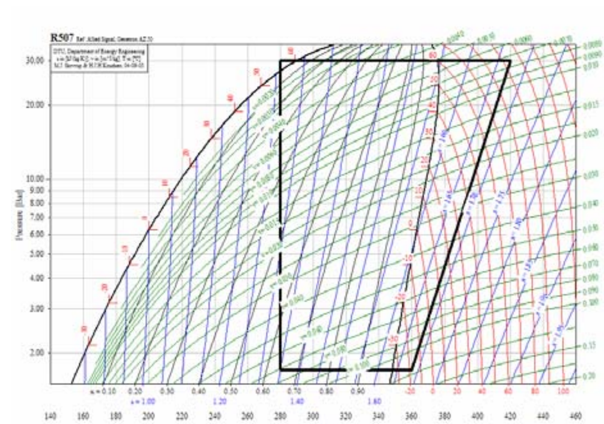

Fig. 8. p-h-diagram for the one-stage cycle with oil cooling

7) Mechanical subcooling

A heat pump system with mechanical subcooling can be realized as shown in Fig. 9 *(Zehnder et al. 2002a)*. By using mechanical subcooling, a second, smaller heat pump cycle subcools the liquid refrigerant of the first stage and uses the heat of the warm refrigerant at the first-stage condenser outlet as the high temperature heat source. Ideally the first-stage cycle is not affected by this measure but the second-stage cycle profits from a high evaporating temperature and therefore, operates at very high efficiencies. Figure 10 shows the pressure enthalpy diagram of a heat pump with mechanical subcooling where both cycles use the same refrigerant. The first-stage cycle is the one with lower evaporating temperature and also lower condensing temperature. The second-stage cycle is symbolized by the cycle which can reach higher condensing temperatures due to the much higher evaporating temperature.

A heat pump with mechanical subcooling can work in one-stage mode or with two-stage mode in order to increase efficiency and heating capacity. Another advantage of this cycle is the possibility to use two different refrigerants which can be optimized for the desired operating range. A typical application for this cycle is a combined system for space heating (low temperature stage) and tap water heating (high temperature stage)

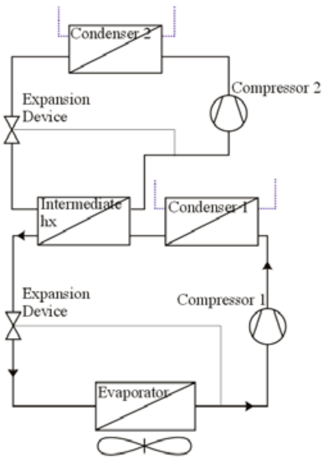

Fig. 9. Schematic of a one-stage cycle with mechanical subcooling

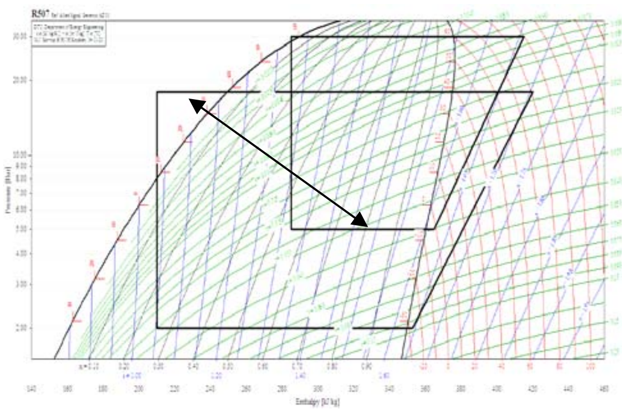

Fig. 10. p-h-diagram for the one-stage cycle with mechanical subcooling

(Zubair et al. 1996) studied an alternate possibility of how to configure a cycle with mechanical subcooling, which will not be explained further in this paper.

8) Other approaches

(Mitsuhiro 1990) described a refrigeration cycle which uses a zeotropic mixture of refrigerants. At high ambient temperatures the heat pump works with the high boiling point refrigerant. As the ambient temperature decreases the refrigerant with the low boiling point, whose pressure is decreasing slower will become the refrigerant with the higher evaporating pressure and therefore, the pressure difference across the compressor will rise slower. To manufacture a heat pump of this kind some additional components such as a rectification column, a heater, a refrigerant ejector, among others are needed.

(Sivakumar 2003) presented a method to use the heat output of the intercooler in a two-stage compressor system to efficiently heat the evaporator in order to not allow the evaporator to cool down to very low temperatures. The author claims that in addition to a better control of the superheat of the second-stage compressor, the efficiency of the system increased.

Of course, there are several approaches of how to configure a heat pump system with two compressors. *(Vaynberg 1995)* presented one possibility of the refrigeration circuit of a two compressor heat pump. The proposed refrigeration circuit provides a high flexibility for the heat output of the system without the need of cycling the system over a very broad operating range. The compressors can run in cascade, parallel, or single-stage configurations.

3 COMPARISON OF DIFFERENT REFRIGERANTS

In order to compare different refrigerants, several aspects such as refrigerant pressure, critical temperature and critical pressure have to be taken into account. The minimal ambient temperature which occurs in the northern states of the US is approximately -30°C. Therefore the evaporating temperature in such a system can be as low as -40°C. As the heat requirement during such cold days is highest, the required condensing temperature reaches its maximum of up to 60°C. Generally refrigerants with a high critical temperature, a low pressure ratio and a high evaporating pressure are preferable. In addition, they should be non flammable, non toxic and environmental friendly.

Table 1 shows the saturation pressures of various refrigerants for temperatures of -40°C and +60°C, in addition to the pressure ratios for this operating condition, and the critical temperatures and critical pressures.

Table 1. Refrigerant data (database: Coolpack 1.46)

	R-744	R-134a	R-290	R-507	R-410A	R-407C	R-404A
P_{sat} [bar] (@ -40°C)	10.1	0.5	1.1	1.4	1.8 - 1.8	0.9 - 1.2	1.3 - 1.4
P_{sat} [bar] (@ +60°C)	$>T_{crit}$	16.8	21.2	29.5	38.2 - 38.3	25.2 - 27.5	28.7 - 28.9
Pressure Ratio	≈10.0	32.8	19.1	21	21.7	29.3	21.6
T_critical [°C]	31	101	97	71	75	87	72
P_critical [bar]	73.8	40.6	42.4	38.0	51.7	46.2	37.3

P_{sat}…Saturation Pressure, T_critical…Critical Temperature, P_critical…Critical Pressure

It can be seen from the data presented in Table 1 that a system with the natural refrigerant R-744 (CO_2) can only be used in a transcritical cycle since the critical temperature is lower than the desired hot water supply temperature. The refrigerants R-134a and R-407C can not be used in heat pump applications over this temperature range since it is not desired to work under vacuum pressure as water vapor and/or air from the ambient could leak into the system. R-290 (Propane) on the other hand would be a good refrigerant, which would still operate at above ambient pressures. However, R-290 is highly flammable, which prevents its usage in the United States. The remaining refrigerants are R-410A, R-507 and R-404A. Simulations show that the discharge temperatures for heat pump cycles with R-410A as the refrigerant are higher than the discharge temperatures with R-404A and R-507 cycles. Therefore, the usage of R-410A is more or less limited to cascade cycles or a heat pump with economizer. Disadvantages for all three refrigerants are their low critical temperatures which reduce the efficiency at high condensing pressures. (*Hewitt et al. 2003)* recommends the use of an expander system which could recover the losses of the expansion device and thus, would be beneficial especially at the high pressure ratios under the given operating conditions. Based on these observations, the most suitable refrigerants in a single-stage heat pump application over a temperature range from -40°C evaporating temperature to +60°C condensing temperature are R-410A, R-404A and R-507 in a subcritical cycle, or R-744 (CO_2) in a transcritical cycle.

4 COMPARISON OF DIFFERENT CYCLES

In this section, the predicted performances of seven different cycles that are presented in this paper are compared. The performance predictions were obtained from the literature (if available) or by conducting simulations accomplished by the authors. The simulations were carried out using the Engineering Equations Solver EES (*Klein, 2004*) Software using the refrigerants R-410A, R-507 and R-404A as well as R-134a for the high temperature stage of the cascade cycle. The heat exchangers were modeled using a lumped capacitance model with specified heat exchanger effectiveness. The compressor data were fitted according to the approach of (*Mackensen et al. 2002)*.

Table 2 presents the results of the performance predictions. The single-stage cycle was used as reference for the other heat pump cycles. Therefore, its efficiency and heat output is set to 100%. This cycle can not be used for very low ambient temperatures since the discharge temperature of the hot gas would be too high.

A two stage cycle with intercooler can be realized with either a two-stage compressor or with two single-stage compressors. Both cycles show the same performance at low temperatures, but the cycle with two single-stage compressors has the advantage that the heat output can be adjusted in three stages by using either one of the two compressors or both compressors together. In order to achieve the best performance a low intercooler temperature is needed.

The two-stage cycle with economizer can also be configured with one two-stage compressor or two single-stage compressors. An advantage in comparison to the intercooler cycle is that the intermediate pressure is easier to regulate and therefore, the control of the cycle is better.

Cascade cycles show big advantages especially when the temperature difference between the sink and source temperatures are large. When the temperature difference decreases, the performance decreases rapidly as the compressors unload. An additional disadvantage compared to other two-stage cycles is that the cascade cycle is very difficult to reverse, which makes an efficient air conditioning mode and also a reverse cycle defrosting mode difficult to achieve.

Refrigerant injection mainly reduces the temperature of the discharge gas and raises the heat output by approximately 15%. The efficiency is almost unaffected compared to the single-stage cycle.

Oil cooling also decreases the temperature of the discharge gas. Both efficiency and heat output stay almost the same compared to the single-stage cycle.

Finally, mechanical subcooling shows a better efficiency and also an improved heat output compared to the single-stage cycle without any oil management concerns. The main problem is that the high discharge temperature for low ambient temperatures is not addressed in this approach. However, this approach could be combined for example with refrigerant injection, which would lower the discharge gas temperature, but increase the cycle complexity.

Several of the cycle concepts presented here and thus, their advantages could be combined. However, any of such combinations will also significantly increase the cycle complexity.

Table 2. Comparison of different heat pump cycles

	# in text	Preferred Compressor*)	Number of heat output steps	Efficiency	Heat output	Discharge temperature
1-stage cycle	1	LT	1	100%	100%	High
2-stage w. intercooler	2	2-stage	1	130%	100%	Acceptable
		Sc, Recip, Rot	3	130%	140%	Acceptable
2-stage w. economizer	3	2-stage	1	130%	100%	Low
		Sc, Recip, Rot	3	130%	150%	Low
Cascade cycle	4	Sc, Recip, Rot	1	140%	140%	Low
Refrigerant injection	5	Sc, Screw	2	Comparable	115%	High
Oil cooling	6	Recip, Rot	1	Comparable	Comparable	Acceptable
Mechanical subcooling	7	LT + Sc	2	110%	120%	High

*) Sc…Scroll, Recip…Reciprocating, Rot…Rotary, LT…Low temperature

5 CONCLUSIONS

There are several possibilities to design residential air source heat pumps for low temperature climates. Seven of these possibilities are described and compared in this paper. All of the cycles presented in this paper have their advantages and disadvantages and their usage may depend on the exact specifications of the given installation. In order to reach a high efficiency, the most promising cycles are systems with two compressors such as two-stage and cascade cycle heat pumps.

It is very difficult to identify refrigerants which show advantageous properties over the whole application range. Most refrigerants show very low pressures at evaporating temperatures of -40ºC. Others have a low critical temperature, which leads to a low system efficiency. Most promising for this type of application are CO_2, R-507, R-410A and R-404A.

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