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P. Neksa
SINTEF Energy Research

R. Aarlien
SINTEF Energy Research

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Petter Nekså and Rune Aarlien

SINTEF Energy Research, Refrigeration and Air Conditioning
N-7034 Trondheim, Norway
E-mail: Petter.Neksa@energy.sintef.no, Rune.Aarlien@energy.sintef.no

ABSTRACT

After the CFCs and the HCFCs were deemed unfit as working fluids in refrigeration, air conditioning, and heat pump applications, there has been a renaissance for carbon dioxide technology. Heat pumps is one of the application areas where theoretical and experimental investigations are now performed by an increasing number of research institutions and manufacturers. This paper gives an overview of some of the current activities in the CO_2 heat pump field. Discussed are the important characteristics of the transcritical CO_2 process applied to heat pumps, and also discussed are theoretical and experimental results from several heat pump applications. Provided that calculations are performed on the premises of the working fluid, and that test plants are constructed and operated to fully exploit the specific characteristics of both the fluid and the transcritical process, the results show that CO_2 is an attractive alternative to the synthetic fluids. Competitive products may be launched in the near future.

INTRODUCTION

When the CFCs and the HCFCs were introduced in the 1930s and 40s, they were by many regarded as the ultimate solution. Advertisements promised that problems, hazards, and discomfort hitherto experienced would soon vaporise from the surface of "planet refrigeration". The only solution was to convert to the last wonders of the chemical laboratories. Today, in spite of the important role the (H)CFCs have had in building up a widespread use of refrigeration, we know that this was a true mistake. The *safety fluids* were not quite as safe as though. It took roughly 40 years for man to discover that the mistake of a century had been made.

In many ways the heat pumping industry today finds itself in the same setting as during the advent of the (H)CFCs; the undesired working fluids, the quest for alternatives, and the introduction of a new generation of synthetic compounds. Advertisements today provide the same message as they did 50-60 years ago - the environmentally safe working fluids are here. The difference is that their name this time is the HFCs. History has repeated itself, once again. An interesting question in this connection is; are we also doing the same *mistake* again? And if so, how long time will it take us to discover it this time?

Many times over the use of laboratory chemicals has proven disastrous. Therefore, in cases where the chemicals will eventually be released to the atmosphere, and when viable alternatives exist, one should abstain from their use. Common sense would suggest that the heat pumping industry now should opt for the natural fluids. For millions of years nature's own refrigerants have proved to be harmless. Carbon dioxide is one of them.

It is understandable that some actors within the industry have not developed any particular liking for CO_2. However, some of the arguments used against CO_2 are some times directly at fault. The classical one being that the efficiency of a CO_2 system is somewhere in the range of 50 to 75% of that of the halocarbons. Forgotten then are the very different properties of CO_2 and the peculiar transcritical process characteristics at high average heat rejection temperatures. A heat pumping process, of course, must be calculated on the working fluid's own premises. Therefore, it has little or no meaning to analyse a CO_2 process under the assumptions used for the halocarbons.

Different kinds of heat pumps applying CO_2 as working fluid are currently being investigated both in theory and in laboratory. The intention with this paper is to give an overview of these research activities, and also to show that both types of activities yield very competitive performance compared to the halocarbons - provided they are done correctly. Additional advantages are also provided.

FUNDAMENTALS

The application of CO_2 as working fluid introduces challenges, but certainly also opportunities and possibilities, compared to off-the-shelf technology. First of all, components for much higher design pressures are needed, typically 150 bar maximum operating pressure. When CO_2 started loosing market shares around 1940, the lack of such components was one of the reasons for the decline. However, with the manufacturing technology and the knowledge base existing today, it is possible to fully utilise the *advantages* introduced by the high pressures, for instance in reducing component size due to high volumetric capacity.

The critical temperature of CO_2 is 31.1°C. This implies that subcritical operation, as known from common refrigeration technology, is only possible when the average heat sink temperature is rather low. In the subcritical operation CO_2 systems competes very well with most other refrigerants.

At higher average heat sink temperatures transcritical operation is necessary. A typical one-stage CO_2 process with internal heat exchange is shown in the *T-s* diagram of Figure 1. A simplified sketch of a heat pump, with its main components and corresponding process points, is included. Counter-flow heat exchange with a heat source and heat sink, is indicated by the broken lines. Characteristic for the process is heat rejection at a supercritical pressure, introducing a gliding temperature instead of condensation at constant temperature.

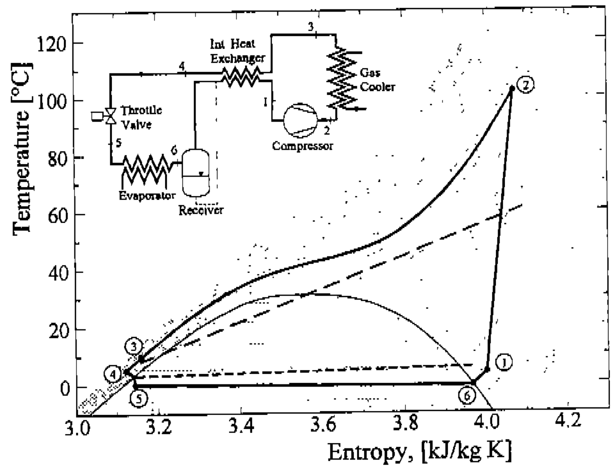

Figure 1 Ts-diagram showing the transcritical CO_2 cycle used for water heating.

Operation of the transcritical process introduces the need to control the high-side pressure in order to obtain optimal efficiency. The efficiency, or coefficient of performance (COP), of a heat pump is defined as the ratio of the energy output, in principal represented by the enthalpy difference at heat rejection, to the power input to the compressor, in principal represented by the enthalpy difference during compression. The highest COP is, of course, obtained with a lowest possible power consumption in combination with a high heat output.

The power consumption increases more or less linearly with the high-side pressure. When the heat sink is characterised by a high temperature glide, as shown in Figure 1, the obtainable enthalpy difference at heat rejection drops fast when the high-side pressure becomes too low, due to a temperature pinch inside the heat exchanger. The result is a rather strong reduction in COP at low heat rejection pressures, as explained in detail in (Nekså, Rekstad et al., 1998). At high pressures the COP is limited by the power consumption of the compressor

In rejecting heat to a constant temperature heat sink, the temperature pinch will, of course, occur in the cold end of the heat exchanger. Consequently, the enthalpy difference is fixed by the heat sink temperature and the pressure at heat rejection, together with the discharge enthalpy from the compressor. Due to the large variation in resulting enthalpy at constant temperature in the supercritical region, a too low high-side pressure will again yield a sharp drop in COP, see for instance (Pettersen and Skaugen, 1994).

In subcritical operation, the COP is in principal limited by the highest heat sink temperature. In transcritical operation, on the contrary, the COP is most often limited by how low refrigerant temperature it is possible to achieve after heat rejection. Consequently, the lowest heat sink temperature gives the limitation. This must be reflected in the design of the total heat pump system, including the heat distribution system. For instance, at a given average heat rejection temperature, a heat distribution system with a high temperature glide would benefit the CO_2 heat pump.

Operation near the critical point offers a good potential for efficient compression. This is due to low pressure ratios and that relatively high valve pressure drops can be accepted for a given loss in temperature (Pettersen, Nekså et al., 1995). The volumetric capacity of CO_2 is in the order of 5-10 times higher than for the common alternatives. This opens the possibility for compact compressors, even though the pressures are high.

The heat transfer characteristics of CO_2 is very good, even in the supercritical region. Combined with the high volumetric capacity, this makes it possible to develop compact and efficient heat exchangers (Bredesen, Aflekt et al., 1997) and (Bredesen, Hafner et al., 1997).

HEAT PUMP WATER HEATERS

(Nekså, Rekstad et al., 1998) describes experimental results for a CO_2 heat pump water heater (HPWH) application. The prototype system results show that CO_2 is very well suited as working fluid for tap water heat pumps. The energy consumption can be reduced by 75% compared to electrical or gas fired systems, when hot tap water is supplied at 60°C, with the ambient air as heat source. Figure 2 shows measured hp-COP as function of the evaporation temperature. The tap water is heated from 8 to 60°C, typical Norwegian conditions. Hp-COP is defined as the heat output divided by the power input to the compressor motor. The shaft power, rather than the input power to the motor, has been measured. The power input to the motor was obtained assuming a motor efficiency of 0.9.

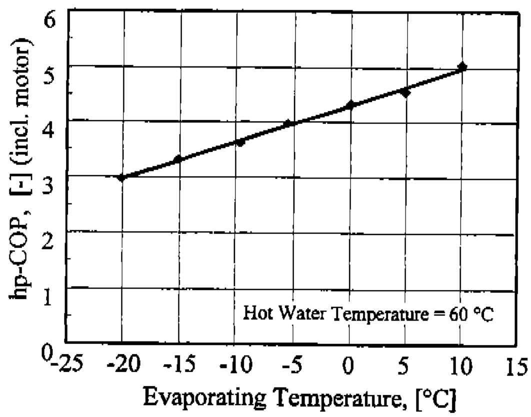

Figure 2 *Coefficient of performance for the heat pump when varying the evaporation temperature.*

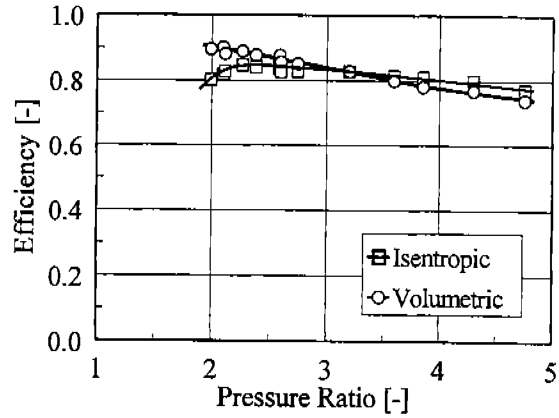

Figure 3 *Isentropic and volumetric efficiency for the CO_2 compressor at varying pressure ratios.*

The high process efficiency is partly due to good adaptation of the process to the application, but also efficient compression and good heat transfer characteristics for CO_2 are contributing to the high COP. Figure 3 depicts isentropic and volumetric efficiency for the prototype compressor at different pressure ratios. The isentropic and volumetric efficiency at design conditions (pressure ratio 2.6, heat rejection 90 bar and heat absorption 35 bar), are 0.84 and 0.86, respectively. Corresponding pressure ratio for a halocarbon reference process is about 4, illustrating the favourable pressure ratio for the CO_2 process. The performance of the compressor is quite satisfactory considered that this is a prototype under development.

A CO_2 heat pump water heater may produce hot water with temperatures up to 90°C without operational problems and with only a small loss in efficiency. Increasing the required hot water temperature from 60°C to 80°C reduces the hp-COP from 4.3 to 3.6 at an evaporation temperature of 0°C. The area of application is thus much larger than for the traditional heat pump systems, often restricted to hot water temperatures lower than 55°C.

Due to the high volumetric efficiency, leading to small flow areas and the good heat transfer characteristics of CO_2, it should be possible to manufacture the systems compact and cost efficient.

The market potential for hot water heat pumps is large. Roughly 20% of the energy use in residential and commercial buildings goes to water heating (IEA-HPC, 1993). In addition, there is a substantial need for water heating in industry, where often the tap water heating may be combined with refrigeration and /or freezing by utilising the cold side of the system.

Several research organisations are now developing CO_2 HPWH, see (Saikawa, Hashimoto et al., 1997), (Hwang and Radermacher, 1997), (Rieberer, Kasper et al., 1997), and (Heyl, Preusser et al., 1997).

RESIDENTIAL HEAT PUMPS

A detailed simulation study on the performance of HCFC-22 and CO_2 in residential heat pumps gave quite promising results (Pettersen, Aarlien et al., 1997). Evaporator temperatures were higher in the CO_2 system, and very small approach temperatures were estimated for the CO_2 gas cooler. The mechanically expanded round-tube heat exchangers were designed within the same core dimensions and air-side pressure drop in both systems. The effects of pressure drop, particularly in the evaporator and suction line of the HCFC-22 system and the superheat characteristics of the expansion valve, gave cooling COP that was similar in both systems, even at high ambient temperatures. Differences in heating capacity characteristics between CO_2 and HCFC-22 are important for the seasonal heating performance, since differences in supplementary heating affects the system energy efficiency. The CO_2 system was able to maintain a higher heating capacity than the HCFC-22 system at low ambient temperatures, thereby saving supplementary heat. Even though the heating COP of the two heat pump circuits are similar, the overall result was a 20% increase in system energy efficiency (HSPF) for the CO_2 system, due to a lower need for supplementary heat.

To verify the theoretical results, an HCFC-22 unit and a prototype CO_2 system was set up in the SINTEF's laboratory. The capacity of the units is a little larger than the systems analysed in the theoretical study, and the heat exchangers applied in the CO_2 lab unit are of the multi-channel (parallel flow) concept. Still, the results should give a fairly good indication of the efficiency level between the two systems. Testing is still ongoing.

The main results from the experiments are shown in Figures 4 and 5. In AC mode, the COP values of the CO_2 unit are slightly *lower* than those of the HCFC-22 unit, whereas in HP mode, the COP of the CO_2 unit is slightly *higher* than those of the HCFC-22 unit. In arriving at the compressor power consumption for the CO_2 system, the shaft power is measured, and a motor efficiency of 0.9 is assumed. The motor efficiency is included in the HCFC-22 power consumption measurements.

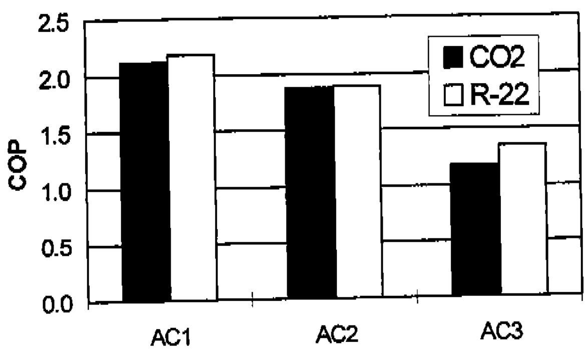

Figure 4 Measured COP, cooling mode. Ambient temperatures are 28 °C, 35 °C, and 46 °C for AC1, AC2, and AC3, respectively.

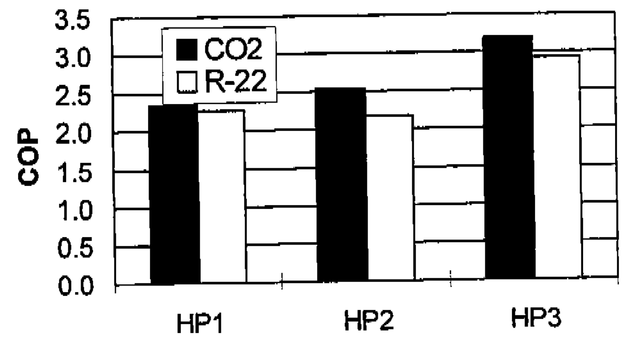

Figure 5 Measured COP, heating mode. Ambient temperatures are 2 °C, 7 °C, and 14 °C for HP1, HP2, and HP3, respectively.

The CO_2 evaporation temperatures in AC mode were 3-4 K lower than for the HCFC-22 unit. One reason for this is water retention. With the flat heat exchanger pipes running horizontally, and the unit only slightly tilted forward, water is effectively prevented from draining. The result is that water builds up until it reaches a *critical mass*, when it is suddenly released in a batch. The water condensation rates were more or less the same for both systems. The CO_2 evaporation temperatures in HP mode were 3.7 K higher to 0.4 K lower than those of HCFC-22. The measured *temperature approach* was between 1.5 and 2.0 K in AC mode and between 7.2 and 15.0 K in HP mode. The relatively large temperature approaches in HP mode are too high for efficient operation. The reason for this inefficiency is that the number of passes through the indoor heat exchanger is too low. There was only one pass through both the front and the rear section. More passes are easily introduced by installing baffles in the headers.

In conclusion, it can be said that the overall system efficiency of the CO_2 system in terms of COP is already competitive to that of the HCFC-22 system in both modes of operation. The heat exchanger design concept applied is neither optimal from a gas cooler point of view, nor from an evaporator point of view. Water retention has been a problem, and this has also caused fluctuating system pressures. The CO_2 compressor has not performed optimally in some of the experiments, owing to worn out bearings, and leakage through the stuffing box. Based on the above, there should be room for further improvements in the performance of the CO_2 system.

AIR HEATING SYSTEMS

(Rieberer and Halozan, 1998) and (Rieberer and Halozan, 1997) made detailed theoretical studies of controlled ventilation air heating system with an integrated CO_2 heat pump. The results look very promising. Figure 6 shows overall and hp-COP as a function of ambient temperature. Ambient air, preheated in a ground-to-air heat exchange system, is used as heat source. At an ambient air temperature of –20°C, the air is preheated to –5°C before entering the heat pump evaporator. The overall seasonal performance factor for a Graz, Austria climate is calculated to be in the range 6.15 to 6.5. This corresponds to a seasonal performance factor of the heat pump of above 4 (author's remark).

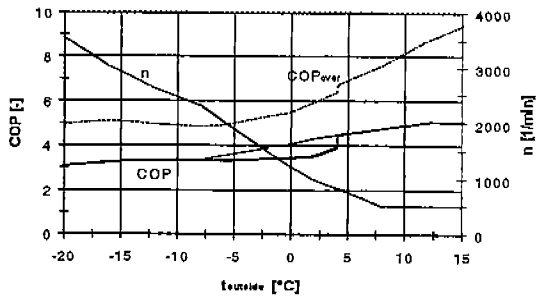

Figure 6: System efficiency and compressor speed.

($q_{tran,-12°C}$ = 15 W/m^2, u_a = 0.8 1/h, p_{cond} = 70 bar)

HIGH TEMPERATURE HYDRONIC HEATING SYSTEMS

(Enkemann, Kruse et al.1997) has made a theoretical study on CO_2 heat pumps for retrofit in typical hydronic heating systems in Western Europe. A system originally designed for temperatures 70/50°C was proposed modified by reducing the mass flow rate of water to obtain a 93/40°C system. The seasonal performance was then increased from 2.8 to 3.2. In addition, this system will be able to supply hot tap water without any loss in energetic efficiency. A prototype system is now being tested at FKW in Germany, as part of an EU project.

HEAT PUMP DRYERS

Another interesting application is heat pump dryers. (Steimle, 1997) reports, based on theoretical considerations, that energy saving is possible due to better temperature adaptation in the heat exchangers, compared to subcritical processes. It is also possible to achieve higher air temperatures without loss in efficiency, thus increasing the moisture extraction rate. A theoretical comparison between a R-134a and a CO_2 system will be published in the June 1998 issue of the International Journal of Refrigeration, and the University of Essen will soon start an experimental investigation on a CO_2 heat pump dryer as a part of an EU project.

OTHER HEAT PUMP APPLICATIONS

Late professor Gustav Lorentzen has published several papers describing the possibilities of using CO_2 as working fluid in heat pumps and refrigeration systems. In (Lorentzen, 1994) he describes system design of large heat pumps for district heating. This is a high-capacity application where turbo expanders may be possible to realise in a cost efficient manner. Described is also the possibility to combine refrigeration/freezing and tap water heating, which will give a very high overall system efficiency.

CONCLUSIONS

- Experimental results show that CO_2 may be successfully used as working fluid in heat pumps with very competitive performance if the system is properly designed.
- There is a good fit between theory and experimental results when calculations are done properly, i.e. when the properties of CO_2 and process characteristics of the transcritical CO_2 process are fully exploited.
- Very promising simulation results exist for application areas where experimental results are not yet available. There are good reasons to continue the experimental work in these areas.
- With all the benefits of a natural working fluid and the very competitive performance demonstrated, CO_2 is and will continue to be a strong contender to the synthetic working fluids.

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