Revival of carbon dioxide as a refrigerant

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In the present situation it seems appropriate to avoid as far as possible the use in large quantities of substances that are foreign to nature and will unavoidably be lost into the biosphere. A much safer philosophy must be to revert to 'natural' refrigerants: substances that are already present in our environment and which are known to be harmless. One such possibility is carbon dioxide (CO_2), which comes very close to being the ideal working medium, provided that a process to give competitive energy performance can be designed. The paper presents some examples of how this can be done.
(Keywords: refrigerant; substitute; R12; carbon dioxide; cycle; air conditioning; automobile; heat pump; district heating)

Possibilité d'utiliser le dioxyde de carbone comme frigorigène

Actuellement, il semble raisonnable d'éviter autant que possible l'utilisation en grandes quantités de substances qui sont polluantes et qui seront inévitablement perdues dans la biosphère. Ainsi serait-il sage de revenir à l'utilisation de frigorigènes 'naturels', c'est-à-dire de substances qui sont disponibles et qui sont sans effet nocif. Par exemple, on peut citer le dioxyde de carbone (CO_2) qui semble être le milieu actif idéal, à condition de le rendre rentable du point de vue énergétique. L'article propose des exemples.
(Mots clés: frigorigène; substitut; R12; dioxyde de carbone; cycle; conditionnement d'air; automobile; pompe à chaleur; chauffage urbain)

The first pioneers of refrigeration, Perkins (1934), Harrison and others, used ether as a refrigerant. Later, a large number of substances were tried with varying success (one old handbook lists 30), until in the first 30–40 years of this century the main refrigerants in practical usage were:

1. ammonia (NH_3) for medium and large stationary systems and also sometimes in ships, often with brine as a secondary refrigerant, but increasingly with direct cooling;
2. sulphur dioxide (SO_2) for household equipment and small commercial applications, but occasionally for capacities up to several hundred kW;
3. carbon dioxide (CO_2) with brine distribution for most ships' installations and also many stationary ones.

The systems were built to a high standard of workmanship and normally functioned very satisfactorily, and the engineers of the time were not aware of any particular problems. One can sometimes wonder if there is not relatively more trouble today, owing to sloppy practice and lack of professionalism in the wake of the odour-free non-toxic refrigerants.

In the 1930s and 1940s the freons were introduced, with a massive advertising campaign, and quickly took over a large part of the market. Only ammonia has remained the preferred refrigerant in large industrial machines, while all other conventional fields of application are now completely dominated by the various types of CFC and HCFC. The main arguments put forward in the propaganda were their complete safety and harmlessness to the environment.

Both of these claims have turned out to be wrong. Many people have been killed over the years by suffocation in spaces below threshold level and in ships; others have suffered injury from dissociation products. Damage to the global environment has led to the Montréal Protocol and universal banning of most CFC and HCFC compounds.

This is by no means a unique experience. Similar predicaments have occurred from the release to the environment of many other new chemicals. The extensive use of ever-new compounds is one of the big problems of our time. In this situation it does not seem very sensible to replace the CFC/HCFCs with a new family of related halocarbons, equally foreign to nature, to be used in quantities of hundreds of thousands of tons every year.

In principle it must be a much better solution to use naturally occurring substances as refrigerants: compounds already circulating in quantity in the biosphere and which we know are harmless. Examples are water, air, carbon dioxide, ammonia, hydrocarbons, nitrogen or the noble gases. Some of these are already in the focus for extended use in the future, in particular air, ammonia and propane. It is the purpose of this paper to show how the old refrigerant CO_2, which has been completely abandoned for more than 40 years and nearly forgotten, can actually be used to advantage in properly designed systems.

Comparison of refrigerants

Let it be said again: the absolutely ideal refrigerant in every respect does not exist. All available compounds have their weak points, which must be taken into account in the design and operation of the system. The

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halocarbons, for instance, have in addition to their environmenal effects relatively large flow resistance losses and poor heat transfer, associated with their high molar mass. Other substances can be combustible or poisonous, like ammonia, and require special safety precautions. This is not technically difficult, but more an economic question, and restricts their use in certain applications.

The alternative refrigerants also differ greatly with regard to pressure and volume requirements, the relative influence of water, oil and impurities, their effect on the common machine construction materials, leak detection, availability and price. The art of choosing the best compound for a given application consists of minimizing the effect of all these properties on the overall cost, reliability of operation and efficiency.

The detailed comparison of refrigerants is standard textbook material and will not be elaborated here. Two points of particular importance need some comments, however: the power requirement and the desirable pressure level.

Power requirement

The efficiency, or COP, of any reversible (loss-free) process, working between given temperature limits, is exactly the same and completely independent of the properties of the working medium used. This follows directly from the Second Law of Thermodynamics. The thermal properties of the refrigerants influence the power consumption only through their effect on the thermodynamic losses.

Much misteaching has been heard concerning the 'efficiency or COP of different refrigerants'. A host of publications in recent years compare different substances when they are used in the traditional reference process, the Evans–Perkins (E–P) process from 1834 (*Figure 1a*). This is encumbered by two irreversible elements: the superheat peak, which is lost in heat exchange in the condenser; and the throttling loss, which is supplied as heat to the evaporator. Both these losses are shown hatched in the figure. They are affected differently by the refrigerants' specific heat capacity properties, and this more or less cancels out, so that only a marginal difference shows up in the efficiency comparison. This is often much overestimated in importance, and many people are actually given to believe that a refrigerant really has 'an efficiency'.

In a real machine a number of additional irreversibilities occur, which depend on the refrigerant properties as well, and may be far more important than those included in the E–P process. They are also heavily influenced by plant design. *Figures 1b* and *c* give a schematic survey in the temperature–entropy chart for refrigeration and heat pumps, respectively, with the individual losses indicated by the squares $\Delta S T_0$. The loss energy, which is released as heat at the evaporation temperature, also represents a loss of refrigeration capacity. The part of the loss area between this level and the ambient T_0 can be interpreted as the work that is theoretically required to compensate for this. For the heat pump this effect is small, as the evaporation temperature is normally close to ambient.

The main losses in a common single stage system are as follows:

1. compressor loss $\Delta S_{comp} T_0$, depending on the absolute pressure level, the pressure ratio for a given temperature

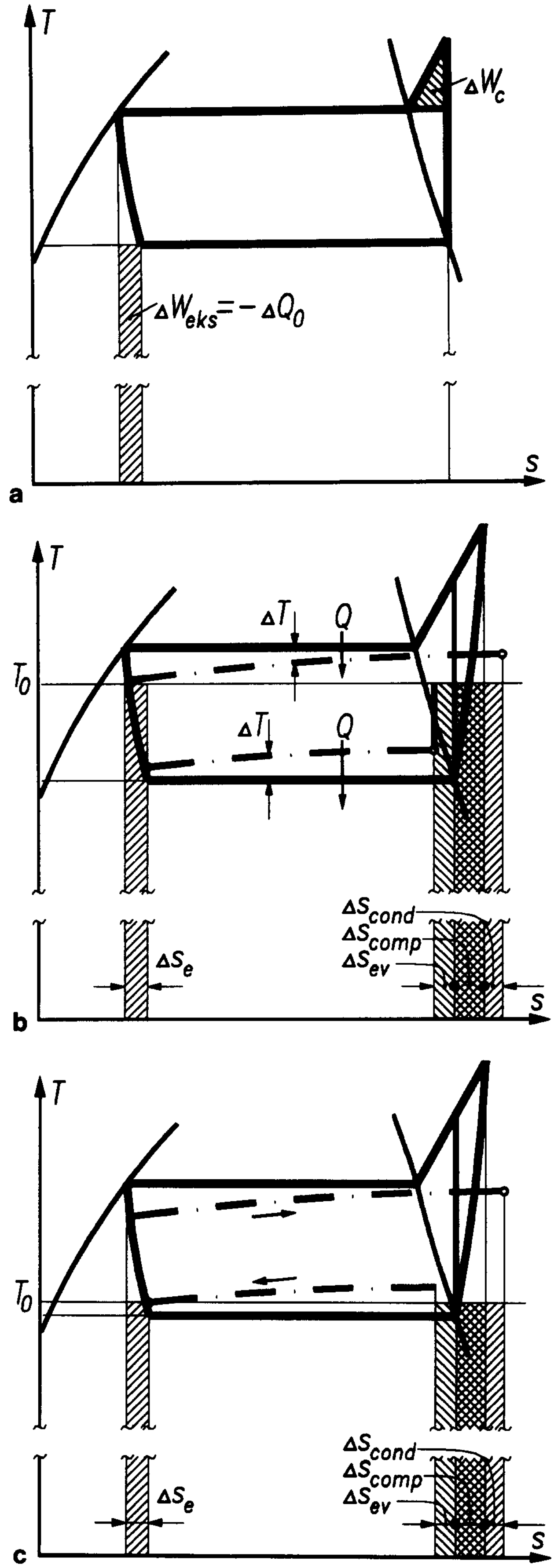

Figure 1 Temperature–entropy diagrams: (a) irreversibility losses in the normal reference (E–P) process; (b) real cycle losses (T_0 ΔS) in a refrigeration machine; and (c) in a heat pump

Figure 1 *Diagrammes température–entropie: (a) pertes d'irréversibilité dans le procédé de référence normal (E–P); (b) pertes réelles du cycle ($T_0 \Delta S$) dans une machine frigorifique, et (c) dans une pompe à chaleur*

lift, the adiabatic exponent and other thermal properties of the working medium, and diverse design parameters;
2. condenser loss $\Delta S_{cond} T_o$, depending on the fit to the actual temperature boundary of the application and the heat transfer properties of the medium;
3. throttling loss $\Delta S_e T_o$, depending on the ratio of the specific heat capacity of the liquid and the evaporation enthalpy, c_p/r;
4. evaporator loss $\Delta S_{ev} T_o$, again a function of temperature fit and heat transfer properties.

In addition the flow resistance losses in the circuit depend on the refrigerant properties and in particular its molar mass, deciding the amount which has to be pumped for a given capacity.

All the individual losses, including those integrated in the theoretical E–P process, can be influenced by the design of the system: type and design of compressor, dimensioning of heat exchangers and piping, adjustment to temperature boundaries, choice of expansion aggregate etc. In principle any thermodynamic loss may be reduced as much as desired by increased expenditure in technical resources; optimization is an economic question. The evaluation of the suitability of a refrigerant is only possible in relation to the system in which it is going to be used. A compound that does not show up well by the conventional comparison on the basis of an E–P process may actually give superior performance in a suitably designed plant. An extreme example may be the use of helium in a Stirling cycle machine. In the E–P comparison it would give zero efficiency and COP; in reality it performs extremely well at low temperatures.

Desirable pressure level

The work exchanged in any thermodynamic machine, be it a heat engine, a water turbine or a refrigeration plant, is a direct function of the product PV: the pressure and volume changes of the working medium, or more exactly the area of the process in the P–V chart, $\int P\,dV = -\int V\,dP$. A high pressure means small dimensions and, within reasonable limits, low first cost. In modern energy systems, where the working pressure can be chosen freely, it has increased steadily over the years in direct consequence of technical development. Such is the case for instance with cold gas machines (such as Philips, ~ 100 bar) and in the oil hydraulics field, where pressures in the range 100–400 bar are common today. Refrigeration and heat pump systems are generally composed of piping units and displacement machinery of a similar nature, and a corresponding pressure level would be desirable for optimal economy.

A parallel development has not been possible in refrigeration and heat pump technology, owing to the desire to operate well below the critical point, in order to avoid excessive overheating and throttling losses in the E–P process. For nearly all currently used refrigerants, including all the halocarbons, the critical pressure is in the range 30–50 bar. The exceptions are water vapour (221 bar), ammonia (113 bar) and carbon dioxide (73.8 bar). Standard compressors on the market therefore have a maximum pressure rating in the order of 25 bar. Recently an ammonia machine for 40 bar pressure was introduced for use in heat pumps with temperatures up to 80 °C[1], but in general refrigeration pressures are used which were common in other fields of power machinery 100 years ago.

In order to operate at rational pressures from an engineering viewpoint it is necessary to use a supercritical or transcritical process, except when water vapour or possibly ammonia refrigerant is used. Supercritical systems are well established for the so-called 'cold gas machines', which are best suited for applications with high temperature lift. Continued efforts to adapt them for the normal range of refrigeration usage have not been successful so far. Transcritical cycles have been avoided except for gas liquefaction, owing to their poor efficiency when they are evaluated on the basis of the E–P process.

Refrigerants of the future

When we accept the philosophy of preferring 'natural' refrigerants, we still have a fair choice of substances with a wide variation of properties to fit the different applications. For really low temperature uses one will continue to prefer gas cycle machines, such as for instance the Philips type, using helium or sometimes air as the working medium. High-temperature heat pumps will go on using water as the near-ideal alternative, and this refrigerant may also find a new application in ice slush production. But for the 'normal' refrigeration field (the temperature range from say -40 to 10 °C or so, covering by far the most important number of systems), the probable preference will be limited to three likely alternatives: ammonia, propane and carbon dioxide. The last candidate will perhaps be surprising to some.

Ammonia and hydrocarbons will be discussed by others, and this paper will be confined to the presentation of some new developments towards the efficient use of CO_2 in some typical applications.

CO_2, a near ideal refrigerant

Carbon dioxide was a commonly used refrigerant from the late 1800s and well into this century. Owing to its complete harmlessness it was the generally preferred choice for usage on board ships, while ammonia was more common in stationary applications. With the advent of the freons, and R12 in the first place, the use of CO_2 was rapidly interrupted. The main reasons for this development were the rapid loss of capacity at high cooling-water temperatures in the tropics, and the failure of the manufacturers to follow modern trends in CO_2 compressor design towards more compact and price-effective high-speed types. The time is now ripe for a reassessment of this refrigerant for application with present-day technology.

Some important properties for the comparison of CO_2 with currently used refrigerants can be found in *Table 1*. CO_2 is naturally present everywhere in our environment. The air contains about 0.35 promille of it, in total nearly 3000 billion tons for the whole atmosphere, and several hundred billion tons per year circulate in the biosphere. No complicated and time-consuming research is needed to ascertain its complete harmlessness.

Someone may possibly argue that CO_2 is also a greenhouse gas, and this is of course correct, although its effect is minute compared with that of the halocarbons. But in reality gas will be used that is already available as a waste product in unlimited quantity from other activities. What we do is just to postpone its release. This is in principle good for the environment, like planting a tree to bind carbon for a period of time.

Table 1 Characteristics and properties of some refrigerants
Tableau 1 *Caractéristiques et propriétés de quelques frigorigènes*

	CFC12	HCFC22	HFC134a	NH_3	CO_2
Natural substance?	N	N	N	Y	Y
ODP	1.0	0.05[a]	0	0	0
GWP[b]					
100 years	7100	1500	1200	–	1(0)[c]
20 years	7100	4100	3100	–	1(0)
TLV_{8h} (ppm)	1000	1000	1000[d]	25	5000
IDLH[e] (ppm)	50.000	–	–	500	50.000
Amount per room vol.[f] (vol%/kg m^{-3})	4.0/0.2	4.2/0.15	–	–	5.5/0.1
Flammable or explosive?	No	No[g]	No[g]	Yes	No
Toxic/irritating decomposition products?	Yes	Yes	Yes	No	No
Approx. relative price	1	1	3–5	0.2	0.1
Molar mass	120.92	86.48	102.03	17.03	44.01
Volumic[h] refr. capacity at 0°C (kJ m^{-3})	2740	4344	2860	4360	22600

(*a*) Somewhat higher values have been suggested by recent studies.
(*b*) Global warming potential in relation to CO_2, with 20 and 100 years integration time (IPCC 1990, 1992).
(*c*) Abundant amounts of CO_2 are recovered from waste gas. Thus the effective GWP of commercial carbon dioxide, for instance used as refrigerant, is 0.
(*d*) Suggested by ICI etc.
(*e*) Maximum level from which one could escape within 30 min without any escape-impairing symptoms or any irreversible health effects.
(*f*) Maximum refrigerant charge in relation to refrigerated room volume, as suggested in ANSI/ASHRAE 15-1989: *Safety Code for Mechanical Refrigeration.*
(*g*) Although considered to be non-flammable, both R22 and R134a are combustible in certain mixtures with air at elevated pressures, but ignition may be difficult.
(*h*) Enthalpy of evaporation divided by saturated vapour volume.

With regard to personal safety, CO_2 is at least as good as the best of the halocarbons. It is non-toxic and incombustible, of course. By release from the liquid form about half will evaporate while the remainder becomes solid in the form of snow and can be removed with broom and dustpan, or just left to sublimate. Most people are already familiar with the handling of 'dry ice'. In the event of accidental loss of a large quantity, a good ventilation system is required in order to eliminate any risk of suffocation, in particular in spaces below ground level. In this respect the situation is the same as for any large halocarbon plant.

It is sometimes maintained that the high pressure of CO_2 could constitute a special danger in the event of accidental rupture. Actually this is not so, as the volume is so small. In the same way as the product *PV* is approximately the same for all systems with the same capacity, the same holds for the explosion energy, regardless of the refrigerant used.

CO_2 has a number of further advantages:

1. pressure close to the economically optimal level;
2. greatly reduced compression ratio compared with conventional refrigerants;
3. complete compatibility with normal lubricants and common machine construction materials;
4. easy availability everywhere, independent of any supply monopoly;
5. simple operation and service, no 'recycling' required, very low price.

The effective application of this medium depends on the development of suitable methods to achieve a competitively low power consumption in operation near and even above the critical point. The following sections will give examples of how this can be done.

A cooling system for motor-car air conditioning

More than 20 million new cars are equipped with an air-cooling system every year, and the number is increasing. Until now these units have been working with CFC12 refrigerant exclusively, and the yearly loss to the atmosphere has been estimated at 120000–150000 tons, much more than for any other field of refrigeration application. At present a rapid conversion to R134a has been initiated, and this will solve part of the problem. But an important greenhouse effect remains, increasing the total contribution of an average future motorcar by 15–20%, and there is a pressing need for a more environmentally benign solution[3].

At the Institutt for Kuldeteknikk, NTH/SINTEF, we have for some years been working on the development of a motor-car air-conditioning unit using CO_2 refrigerant, presenting a complete solution to the environmental problem. The system and its performance characteristics are more completely described in a recent publication[4]. This section will therefore be limited to a brief presentation of the principles.

A schematic diagram of the new system and its working principle in the *P–h* chart are shown in *Figure 2.* This also demonstrates how the refrigeration output and power consumption can be changed by varying the discharge pressure of the compressor, and this should normally be regulated to give a COP near its maximum. When needed, the capacity can be increased above its normal value by a further increase of pressure, at the cost of a somewhat higher power consumption. This is an important advantage, for instance in starting from an overheated condition of the passenger compartment. The cooling time is shortened, the size of the system can be smaller than otherwise required, and the total energy expenditure is actually decreased.

The low-side liquid separator/receiver in combination with the internal heat exchanger is essential to the proper functioning of the system, and has a multiple purpose:

1. to permit a certain excess amount of liquid supply to the evaporator in order to simplify the control system and enhance heat transfer;

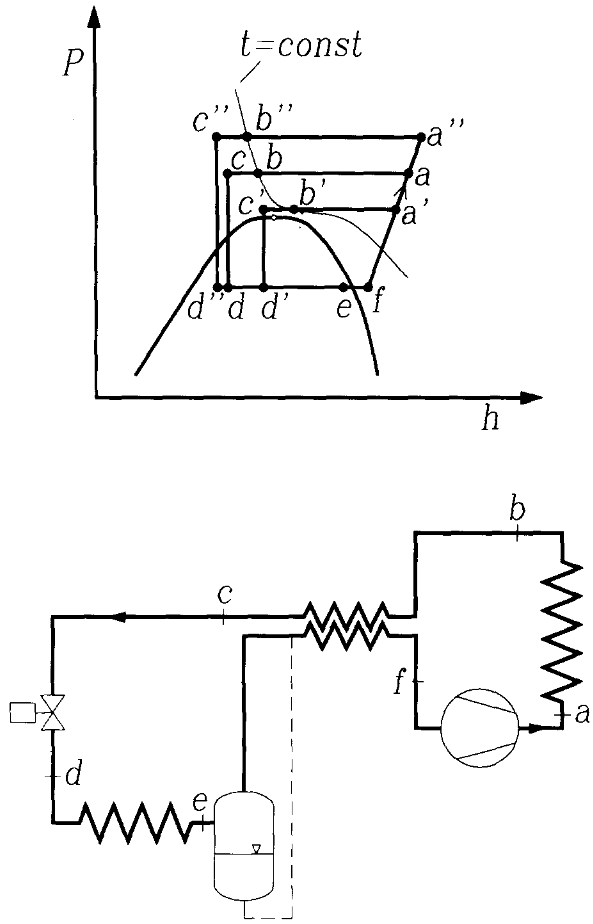

Figure 2 Flow circuit and *P–h* diagram for the transcritical cycle

Figure 2 *Circuit d'écoulement et diagramme P–h pour le cycle transcritique*

2. to withdraw or deliver extra charge of refrigerant for the regulation of the high-side pressure by means of the throttling valve;
3. to hold a sufficient amount of liquid to cover the needs under all possible working conditions and compensate for unavoidable losses by leakage over a reasonable time;
4. to meter a suitable amount of lubricant to the compressor by means of a capillary tube or throttling valve to the suction line;
5. to provide a sufficient gas volume to avoid excessive pressure build-up when the plant is idle at very high ambient temperature.

Extensive testing of prototype equipment in the laboratory has shown that the new system works well under all conditions that can be expected in practical usage. The energy performance is as good as or even better than that of the systems currently used, as demonstrated by the exergy flow charts in *Figure 3*. This good result is largely due to the excellent performance of the compressor, when operating with a very high mean pressure in combination with a very moderate compression ratio. The efficient heat transfer in the evaporator also contributes to the result.

Large heat pumps

Another example relates to a very different application: space heating by means of large heat pumps.

Most heat pumps extract low-temperature heat from the immense reservoir of the environment (air, water, rock and so on), and a process with constant-temperature evaporation is therefore quite suitable for the purpose. Heat pumps give off the thermal energy at a higher temperature to a finite stream of air or water with limited heat capacity, resulting in a more or less gliding temperature. The amount of temperature change can range from a few degrees in a small direct condensation air heater (15–20 K for normal 'split units') up to 30–40 K in large district heating networks and even more in some industrial applications and direct tap-water heating. This causes a very considerable excess power requirement in the normal type of cycle with condensation and heat rejection at essentially constant temperature (*Figure 4A*). Ideally one should have a cycle with gliding temperature output as indicated schematically in *Figure 4B*.

Such a process can be approached by a transcritical cycle, using CO_2 as the working medium (*Figure 5*). In order to get a satisfactory fit to the near-logarithmic temperature curve of the heat-absorbing medium (water), the discharge pressure should be well above critical, in the order of 90–100 bar or higher. This means that when the evaporation temperature is, for instance, 0 °C, the discharge temperature in single-stage compression with dry saturated suction will be about 70–80 °C. This temperature can be adjusted in a certain range, up or down, by varying the discharge pressure and suction gas state, as illustrated schematically in *Figure 5B*, using a conventional suction gas heat exchanger or, possibly, some liquid injection. The flexibility in the direction of a temperature reduction is rather limited, however. Lowering the pressure too much will deteriorate the temperature curve fit in heat exchange, while wet suction rapidly leads to poor compressor performance. The single-stage system shown is therefore most suitable when the required temperature glide is higher than 40–50 K, depending on the heat source temperature. In such cases it may easily reduce the specific power consumption by up to 40% compared with the conventional process, improving the COP correspondingly.

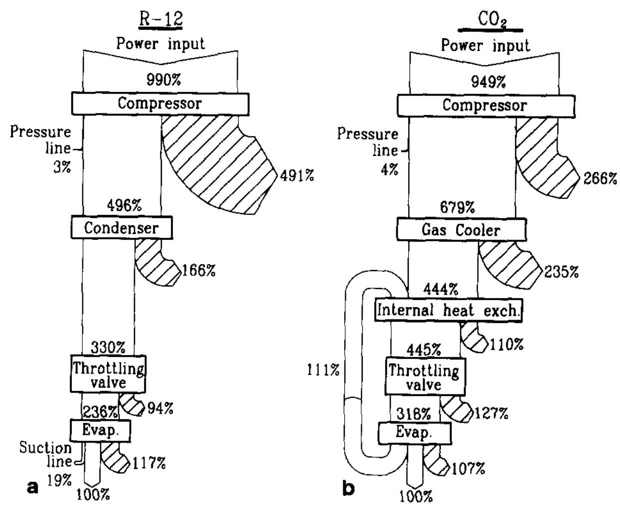

Figure 3 Exergy flow in (a) standard R12 system and (b) CO_2 prototype at ambient temperature 43 °C and 30 °C in the passenger compartment, driving conditions

Figure 3 *Ecoulement exergétique dans (a) l'installation 'standard' au R12 et (b) le prototype du CO_2 à température ambiante 43 deg C et 30 deg C dans le compartiment passager*

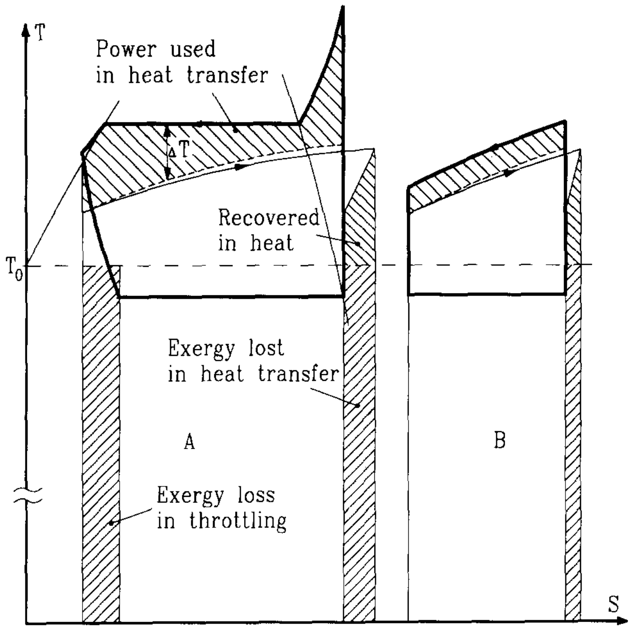

Figure 4 Exergy loss in heat rejection from a heat pump to a stream of fluid with finite capacity: A, conventional E–P process; B, gliding-temperature ('rhombic') process

Figure 4 *Perte d'exergie du rejet de chaleur d'une pompe à chaleur dans un écoulement de liquide de capacité finie: A, procédé classique E–P; B, procédé (rhombique) de température glissante*

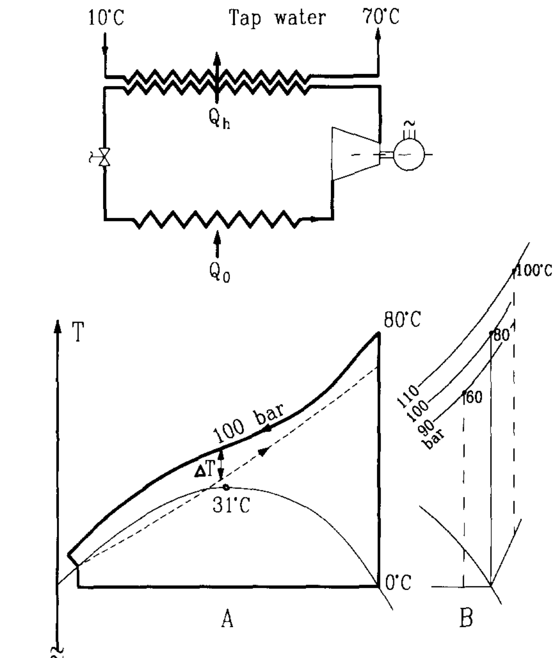

Figure 5 Temperature–entropy chart for a transcritical heat pump producing hot water

Figure 5 *Diagramme température–entropie pour une pompe à chaleur transcritique produisant de l'eau chaude*

For most applications heat pumps with a smaller temperature glide are needed. This problem is readily solved by a system of staged compression, to give a discharge temperature close to the desired level. *Figure 6* gives an example of a two-stage arrangement, suitable for a modern low-temperature district heating application with a heating range of, for instance, 35–60 °C. The temperature–entropy chart compares the transcritical CO_2 process with a conventional R12 system for the same capacity and temperature requirement. The power saving achieved by the better temperature fit in heat transfer is a good 20%, but much of this is lost again in the throttling from a relatively high temperature. However, the compressor efficiency will be much better for the CO_2 alternative as a result of the greatly reduced compression ratio. All told, the CO_2 system will be considerably superior to the conventional in energy performance.

It may be objected that a two-stage system is a complication and means added cost. The difference in this respect is very modest, however. Two-stage centrifugal compressors are normal in district heating plants anyway, and the splitting of the high-side heat exchanger in two parallel circuits is actually a minor adjustment. The modifications will be paid for many times over by reduction in the physical dimensions alone, not to mention the considerable saving in power. Very important also is the elimination of the heavy leakage of CFC or HCFC to the atmosphere, often associated with large district heat pumps.

As pointed out above, the relatively high throttling loss for CO_2 detracts considerably from the advantage of this refrigerant in a simple system as shown. This can be amended by a number of well-known methods[5]. One solution is a normal two-flow throttling and subcooling as shown schematically in *Figure 7*. It is also expedient to recover the expansion energy directly, as the properties of CO_2 make this feasible.

With a conventional refrigerant like R12 most of the theoretical expansion work comes from the flash gas and

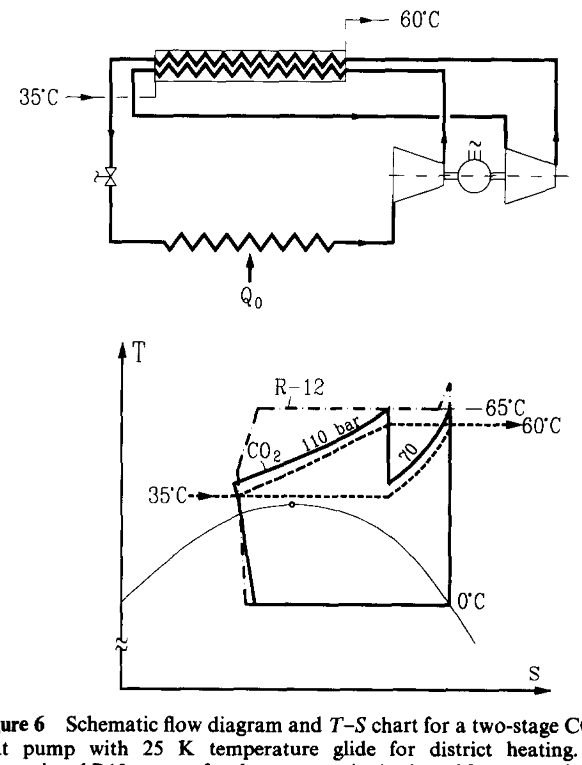

Figure 6 Schematic flow diagram and *T–S* chart for a two-stage CO_2 heat pump with 25 K temperature glide for district heating. A conventional R12 process for the same service is plotted for comparison. Ancillary equipment needed for pressure regulation etc. is omitted

Figure 6 *Schéma et diagramme T–S pour une pompe à chaleur biétagée au CO_2 avec un glissement de température de 25 K pour le chauffage urbain. Un procédé classique au R12, pour le chauffage urbain également, est calculé en comparaison. On ne considère pas les équipements anciens nécessaires pour le contrôle de la pression, etc.*

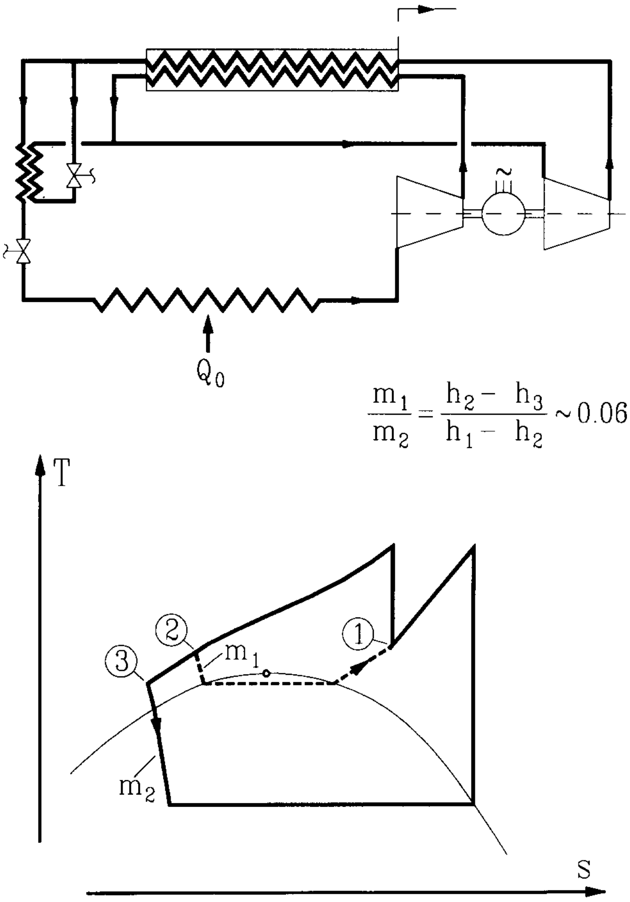

Figure 7 Two-flow throttling and subcooling

Figure 7 *Double détente et sous-refroidissement*

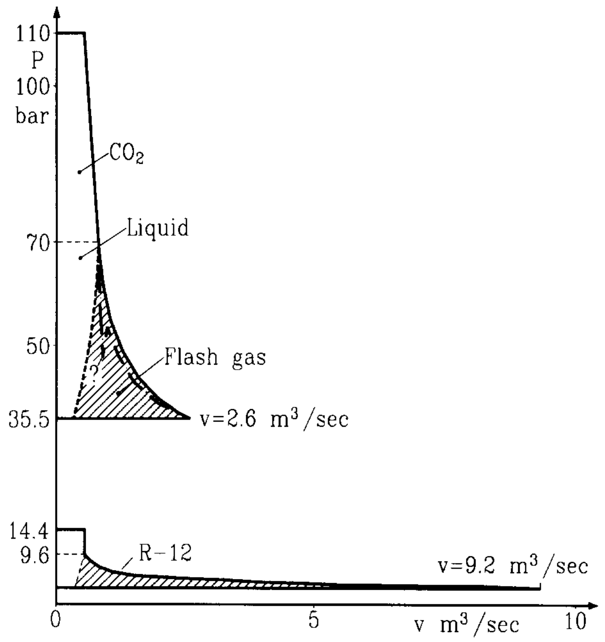

Figure 8 *P–V* diagrams for isentropic expansion of R12 and CO_2 respectively in a 100 MW heat pump with identical boundary conditions

Figure 8 *Diagrammes P–V pour la détente isentropique du R12 et du CO_2 respectivement dans une pompe à chaleur de 100 MW avec des conditions aux limites identiques*

the *P–v* diagram becomes very thin, with a low mean pressure (*Figure 8*). For CO_2 the situation is quite different, with most of the work in the liquid phase, a high mean pressure and small volume requirement. An expansion aggregate becomes a cost-effective element in large installations, which may take the form as shown schematically in *Figure 9*.

Such a system, with a temperature glide in the order of 25–30 K, has the capability of increasing the COP by at least 20–25% in comparison with conventional installations. If the system is operated under conditions deviating from the ideal temperature match the efficiency will suffer, but will still be superior over a fairly wide range.

Commercial refrigeration with CO_2

The term 'commercial refrigeration' includes a great variety of equipment, from small self-contained units of less than 1 kW to extensive and complex systems for large supermarkets and cold stores with different temperature levels and a power consumption of several hundred kW. They almost invariably use halocarbon refrigerants, typically R12 and R502, and the leakage to the atmosphere worldwide is considerable. A change-over to a more harmless working medium is a pressing necessity.

A large supermarket refrigeration system may be quite complex and does not lend itself to the direct use of ammonia, as even a small and harmless leak will give an objectionable smell, and can even cause anxiety and perhaps panic among the customers. There is a trend in some countries to solve the problem by using a central ammonia plant in a closed space and a secondary refrigerant such as glycol or calcium chloride brine for distribution in the shop area. This works well for chilling temperatures, but is less suitable for freeze units, owing to the high viscosity of the normally available brines. Some suppliers therefore use separate small R22 aggregates with a very limited charge for the freezers, cooling their condensers with brine from the central system.

It is also possible to use a boiling liquid as a secondary refrigerant in a pump circulation system. Ammonia was sometimes used in this way in the old days ('ammonia condensing')[6]. In a recent remarkable paper to the IIR Dr S. Forbes Pearson describes the use of liquid CO_2 in a similar way and its experimental realization in freezer storage in Scotland[7]. Similar development work has been done at our institute in Trondheim. The principle is shown in *Figure 10*, which is taken from Dr Pearson's paper. Defrosting can be arranged on the 'hot gas' principle by adding a special pump and boiler as shown, using heat of condensation from the primary refrigerant.

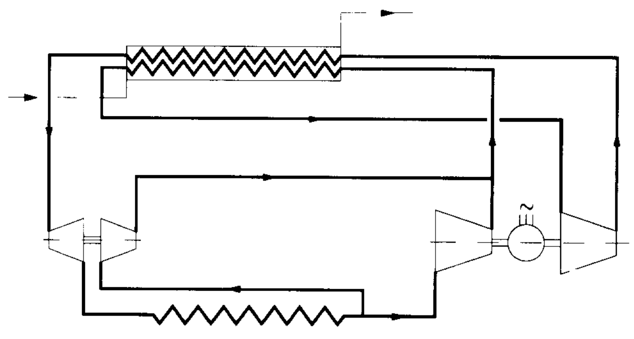

Figure 9 Schematic flow diagram for a two-stage CO_2 pump with expander unit for a typical temperature glide of 20–30 K (district heating)

Figure 9 *Schéma d'une pompe biétagée au CO_2 avec détendeur de récupération pour un glissement type de température de 20–30 K (chauffage urbain)*

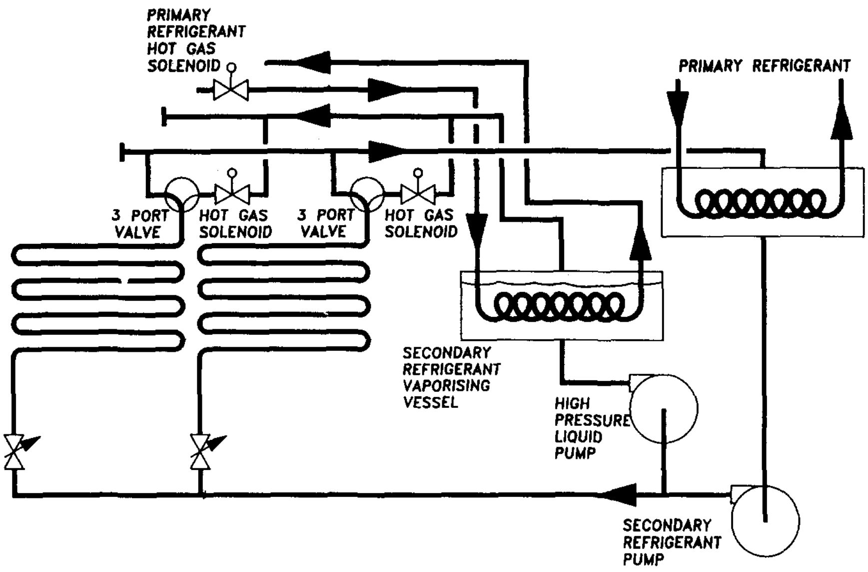

Figure 10 CO_2 condensing system as described by S. F. Pearson[10]
Figure 10 *Système de condensation du CO_2 décrit par S. F. Pearson*

Any supermarket will need at least two temperature levels and this can of course be arranged using two separate CO_2 distribution circuits. It is also possible, as for the brine system described above, to use a main pump circulation loop for the coolers in combination with decentralized small CO_2 compressor units for the freezers, discharging into the cooler return line. Another obvious solution is to use a central CO_2 compressor for a low-temperature circuit in a more-or-less normal cascade arrangement. The most suitable plan will depend on the number of freezer units, capacity and local conditions.

It may, however, be more practical to use a complete CO_2 refrigeration system in a similar way as described above for motor-car air conditioning. The relatively large condenser and throttling losses will, at least in part, be recovered by improved compressor performance due to much reduced pressure ratio and the elimination of an extra heat-exchange temperature loss. In a large installation further improvements may be achieved by staged compression and throttling, and possibly recovery of expansion work as discussed in the section on large heat pumps.

A novel solution

A number of circumstances combine with the characteristics of a transcritical process to make a CO_2 refrigeration system particularly attractive for many applications such as supermarkets, for instance, where a need for hot water also exists.

1. The necessary water consumption for gas cooling in a transcritical CO_2 plant is very small and only in the order of 10% of what would be needed for the condenser of a conventional system.
2. The temperature of the public water supply is nearly constant the year round and much lower than the summer ambient, in Norway typically 4–8 °C.
3. Hot water is more valuable than cold water wherever there is a demand for it.
4. Some hot water is needed nearly everywhere that refrigeration is required.
5. Water is a good heat accumulator.

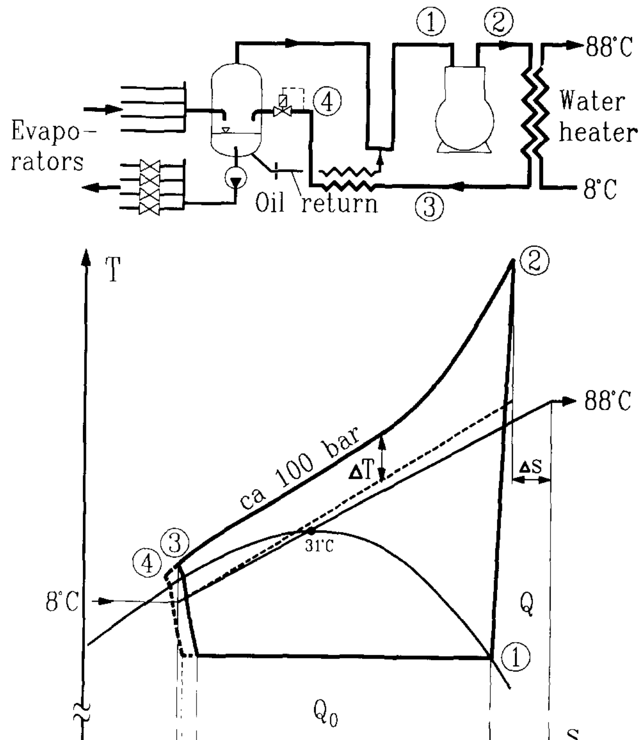

Figure 11 Sample system diagram and temperature–entropy chart for a transcritical CO_2 process, simultaneously producing refrigeration and hot tap water

Figure 11 *Schéma du système et diagramme température–entropie pour un procédé transcritique au CO_2, produisant simultanément du froid et de l'eau chaude au robinet*

The schematic piping diagram and T–S chart of a CO_2 system for the simultaneous supply of refrigeration and hot tap water in a typical supermarket are shown in *Figure 11*. The plant works year round on a transcritical cycle with nearly constant conditions of loading and temperatures. As the compressor discharge temperature will normally be well above 100 °C, the water can be heated to near this point.

As there is no liquid present at supercritical conditions at the high-pressure side, the receiver is combined with the liquid separator in the suction line. The throttling valve serves to control the discharge pressure at the desired level, in the order of 100 bar, and liquid forms only as the refrigerant is expanded under the saturation line. A pump circulation overfeed system guarantees simple distribution and good efficiency of the evaporators, but it is also possible to use direct pressure supply 'dry expansion' coils if so desired. The liquid pump can be placed in the liquid part of the separator in order to avoid high-pressure design and shaft-seal problems.

Normal lubricating oils are only slightly soluble in liquid CO_2, and it is desirable to choose one that is heavy enough to sink to the bottom in the separator. Oil return to the compressor can then be arranged in the usual way via a heat exchanger to the suction line or directly to the separation volume of the machine. The loop of the suction line as shown in the diagram may serve to help the circulation when only a limited liquid head is available, but other arrangements are possible as well.

The system is similar to the one developed for automobile air conditioning and covered by patent. Based on measurements on that design we may assume that the power consumption will be about the same as for a conventional condensing plant or even a little better, taking into account the advantage of low cooling-water temperature. The heat discharge loss will be compensated by a much improved compressor performance and better air cooler efficiency. The cost of cooling water must be taken into account, but this can be more than recovered if hot water is utilized.

In some cases cooling water may be available at no charge, but more often it has to be taken from the community supply and paid for at the regular rates. These vary a great deal: in Norway typically in the range 1–10 NKr m^{-3} (sewer fee inclusive), with a most frequent value about 5–6 NKr m^{-3}. With a normal power price of about 0.5 NKr kWh^{-1}, this means that the ratio of water to power cost varies around 10 kWh m^{-3}, in the range from 1 to perhaps 20 in extreme cases. In most other countries the cost of both water and power is generally a bit higher, but the ratio varies in the same range, except in cases of extreme water shortage[8].

The power consumption cost is given by

$$K_p = Q_o/\varepsilon \cdot \alpha \quad \text{NKr h}^{-1}$$

and the water cost by

$$K_W = Q_o \frac{860(\varepsilon+1)}{1000\varepsilon\Delta t}\beta \quad \text{NKr h}^{-1}$$

where ε is the coefficient of performance, Q_o/P; Δt is the water temperature increase (K); α is the power price (NKr kWh^{-1}); and β is the water price (drain inclusive (NKr m^{-3})).

The percentage additional cost of water over power is

$$A = 100\frac{K_W}{K_p} = 86 \times \frac{\varepsilon+1}{\Delta t} \times \frac{\beta}{\alpha}$$

and is shown in *Figure 12* for a typical heating range Δt of 80 K and three values of ε.

The power cost for an equivalent electric water heating would be

$$K_{pw} = Q_o \times \frac{\varepsilon+1}{\varepsilon}\alpha$$

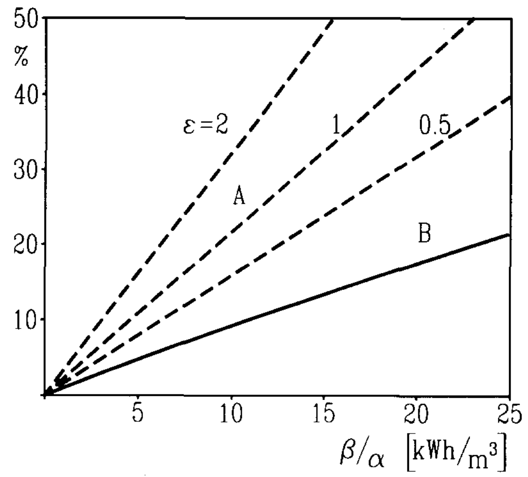

Figure 12 A, water cost as an additional percentage to power cost at varying refrigeration ε; B, percentage of hot water required to recover cost

Figure 12 *A, pourcentage du coût de l'eau ajouté au coût de l'électricité, pour plusieurs COP de refroidissement ε; B, pourcentage d'eau chaude nécessaire pour amortir le coût*

and the total running cost of hot water production would be

$$K_{WW} = Q_o\left[\frac{\varepsilon+1}{\varepsilon}\alpha + \frac{0.86(\varepsilon+1)}{\varepsilon\Delta t}\beta\right]$$

This leads to a required fraction of the hot water from the combined unit, which must be used in order to recover the total water cost:

$$B = 100\frac{K_W}{K_{WW}} = 100 \times \frac{860\beta}{860\beta + 1060\Delta t\alpha}$$

This result is also shown in *Figure 12* as a function of β/α, independent of ε.

As an example, assuming near-average conditions $\varepsilon = 1$, $\beta/\alpha = 10$ and $\Delta t = 80$ K, the water consumption will contribute 21.5% to the running cost when no hot water is being used. A useful consumption of only 9.7% is enough to compensate for the total cooling-water cost. Normally the demand will be larger, and a usage of, for instance, 45% will suffice to pay for the total running cost of the system, including electricity. Above this level not only is the refrigeration free, but there is an extra profit from the hot-water production.

It may be objected that hot tap water may be produced by a separate heat pump or by adding an extra compression stage to a regular refrigeration machine to reach the required level. Disregarding the extra investment, this will also demand an extra power consumption. If we assume a heating COP $\varepsilon_H = 2$ for such a system, the running cost recovery time for the transcritical CO_2 process will be roughly doubled. The new process is still greatly superior in such cases where there is a reasonable demand for hot tap water.

It may safely be assumed that such is the case in most places where commercial refrigeration is installed. The water heaters are more often than not actually installed in the same equipment room. In addition, except in very hot countries, there is always need for space heating in the winter time. Any excess hot water can be applied for this purpose in the season. Although this is slightly less advantageous than tap-water production, as the water value is lost and some heat at the low-temperature end is not recovered, it may still help to further improve the cost-effectiveness of the new system.

Conclusion

CO_2 is as close to the ideal refrigerant as it is possible to come, when only a system to achieve competitive power efficiency can be devised. It is completely benign to the environment, safe, inexpensive, compatible with normal machine construction materials and lubricants. Its relatively high pressure is perfectly adapted to modern machine design and gives a dramatic reduction in the required compressor volume and pipe dimensions. The excellent heat transfer characteristics in the vicinity of the critical point are yet another advantage. All this should lead to a considerable reduction in equipment cost when a reasonable production volume is attained.

A new transcritical system has been designed for use in motor-car air conditioning. Extensive testing has demonstrated that it is very competitive with conventional equipment with regard to power consumption, compactness and cost. Similar systems can be used to advantage in other applications too. Some features of the new concept are covered by patent.

Making use of the special properties of CO_2 in transcritical operation, it is possible to achieve a nearly ideal fit to the gliding temperature, which is required in applications such as, for instance, large district-heating schemes. The high pressure differential and relatively small flash-gas volume open the possibility of efficient recovery of the expansion exergy in a compact expander.

A system using CO_2 refrigerant in a transcritical cycle has been conceived, combining commercial refrigeration and production of hot tap water. It is running year round at near-constant conditions and has the advantage of very simple controls and a robust evaporator liquid supply. Even a small fraction of hot water usage is sufficient to bring the running cost well below that of a conventional plant, and at normal tap-water demand the cost advantage is very considerable.

It is believed that the new system will have a bright future as a practical solution to the difficulties caused by the Montréal Protocol restrictions.

References

1 **Korfitsen, E., Christensen, A.** New ammonia technology for heat pumps *Proc Int Symp Refrigeration* NTH, Trondheim (22–24 June 1992) 165–177

2 **Paul, J.** Binary ice technologies for the production of pumpable ice slurries *Proc IIR 1992–93* preprint

3 **Lorentzen, G., Pettersen, J.** New possibilities for non CFC refrigeration *Int Symp Refrigeration* NTH, Trondheim (22–24 June 1992) 147–163

4 **Lorentzen, G., Pettersen, J.** A new, efficient and environmentally benign system for car air conditioning *Int J Refrig* (1993) **16** (1) 4–12

5 **Lorentzen, G.** Throttling, the internal haemorrhage of the refrigeration process, IIR (1983)

6 **Lorentzen, G.** Fish freezing plant in Bergen, Norway *Refrig Eng* (1954) **62** 41–43, 100–102

7 **Pearson, S. F.** Development of improved secondary refrigerants. IIR 1992–93, preprint (4 March 1993)

8 **Ødegaard, H.** NTH: private communication