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Refrigeration systems with minimum charge of refrigerant

Björn Palm *

Royal Institute of Technology, Department of Energy Technology, S 100 44 Stockholm, Sweden

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Abstract

Concern for the environmental effects of HFC-refrigerants as well as the use of flammable refrigerants has resulted in a need of decreasing the refrigerant charge in refrigeration and heat pump systems. This paper discusses the possibility of such reductions, both at the systems- and the component level. It is shown that a move towards indirect systems, using secondary refrigerants, on both the cold and the hot side of the system may result in considerable reduction of charge. However, this reduction may come at the cost of slightly reduced system performance, which in itself is detrimental from an environmental point of view. At the component level, it may be shown that the main contents of refrigerant is usually contained in the heat exchangers. By selecting compact designs the charge may be reduced to extremely low levels. Specifically, mini-channel heat exchangers can be used for reaching low charge. With proper selection of heat exchangers, the system performance should not be influenced by the reduction of charge. For indirect systems, the amount of refrigerant solved in the compressor oil may be comparable to the amount in the (compact) heat exchangers. A possible solution to reduce this amount is to use compressors with less oil. With components selected for minimum charge, the system design may be different than what is usual. Instead of a high pressure receiver and a thermostatic expansion valve, a capillary tube may be used in combination with a minimal low pressure receiver, similar to the system design used in household refrigerators.
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1. Introduction

Since the realization of the CFC-refrigerants' negative effect on the ozone layer in the end of the 1980s the refrigeration industry has been facing the challenge of adapting the system solutions to new refrigerants. The global phase-out of CFCs within a ten year period, through the Montreal protocol, is probably the first successful example of an international agreement to overcome a global threat to the environment. During the last five to ten years the focus has shifted towards the threat of global warming and this has raised the demand for strict control of the emissions of HFCs, which do not contain chlorine and thus have no influence on the ozone layer, but which in most cases are strong greenhouse gases.

Parts of the scientific community as well as environmentalists have suggested that natural refrigerants, i.e. compounds which are naturally occurring in the environment, should be used as substitutes for the man-made HFC compounds now being used. In some countries, e.g. Denmark and Austria, legislative measures have already been taken towards a phase out of HFCs. Within the EC, the use of fluids with a global warming potential above 150 (including all commonly used HFCs) is being banned in mobile air conditioning according to a gradual phase out program. On the global scale, regulations on HFCs and other greenhouse gases are discussed within the Kyoto protocol and later amendments, however, without any agreements yet.

The natural refrigerants discussed as substitutes for HFCs are hydrocarbons, ammonia, carbon dioxide and water. In this group, the hydrocarbons are most closely related to the HFCs and can be used in today's H(C)FC systems without any major changes in the design. For example, a system designed for R22 may be run with

* Tel.: +46 8 7907453; fax: +46 8 204161.
E-mail address: bpalm@energy.kth.se

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propane as refrigerant without even changing the expansion valve. However, as the hydrocarbons are flammable, additional safety precautions are necessary, and the system should be designed for the lowest possible charge of refrigerant.

Of the other natural refrigerants, ammonia is already being used in larger industrial systems. Due to the strong smell, which may cause panic, it is not used in air conditioning or refrigeration in public areas. In high concentrations it is poisonous and, at least together with traces of oil, it is flammable. As it is not compatible with copper it cannot be used in traditional HFC-systems or with traditional hermetic or semihermetic compressors.

Carbon dioxide is being suggested as an environmentally friendly refrigerant but requires a different system design due to the high vapor pressure and the low critical temperature (32 °C). For some applications the energy efficiency of CO_2 is inferior to other fluids presently used.

Water, finally, can only be used above 0 °C and requires very large volume flows due to the low vapor pressure at the temperature levels of all common applications.

Independently of if future refrigeration systems will be using HFC refrigerants or natural refrigerants in the form of hydrocarbons or ammonia, reduction of the refrigerant charge in the systems will be of utmost importance, for the HFCs for environmental reasons and for the natural refrigerants for safety reasons.

When evaluating systems from the environmental point of view, both the direct effects due to refrigerant leakage and the indirect effects caused by the production of the electricity used for running the system must be considered. This means that a low-charge system must be almost as energy efficient as a traditionally designed system to be competitive from the environmental point of view.

2. Are leakages unavoidable?

It may be argued that systems should not have to leak and that with proper design the refrigerant in the system should not be a threat to the environment. However, experience is different. The two types of systems responsible for the largest amounts of leakage are mobile air conditioning and commercial refrigeration systems. In the case of mobile air conditioning, the leakages are mainly caused by the fact that these systems have open compressors, i.e. compressors driven by the engine through a belt drive. In this type of compressor, a shaft seal is necessary and this seal is a weak point in the system. Also, these systems typically have mechanical connections in between the components as well as flexible hoses which may start leaking in the harsh and vibrating environment under the hood of the car. Estimates of the leakage rates of AC systems in newer cars have been done in a study for the EC [1]. They concluded that the annual leakage rate per vehicle at normal operation (not including accidents or sudden major failures) was about 53 g or 7% of the charge (or about 70 g including other losses). A similar investigation made for the Californian government, including all types of losses ended up at 80 g per vehicle per year. These figures (Fig. 1) indicate that the leakage rates have decreased since the mid- 1990s, but that the total leakage rate is still substantial considering the large total number of vehicles with AC system.

For commercial refrigeration systems (supermarkets), typical leakage rates are 15–20% of the charge per year [2]. In spite of a large effort in reducing this number, it has been difficult to reach desired levels.

Fig. 2 shows the annual leakage rates for different refrigerants reported for one large supermarket chain in Sweden [7]. During the time shown in the diagram, R12, R502 and R22 have been phased out and the majority of the systems are now using R404A. As shown, there is no clear indication that leakage levels below 10% should be reached in the near future. This in spite of the government and company efforts to reduce leakages by legislation on the design

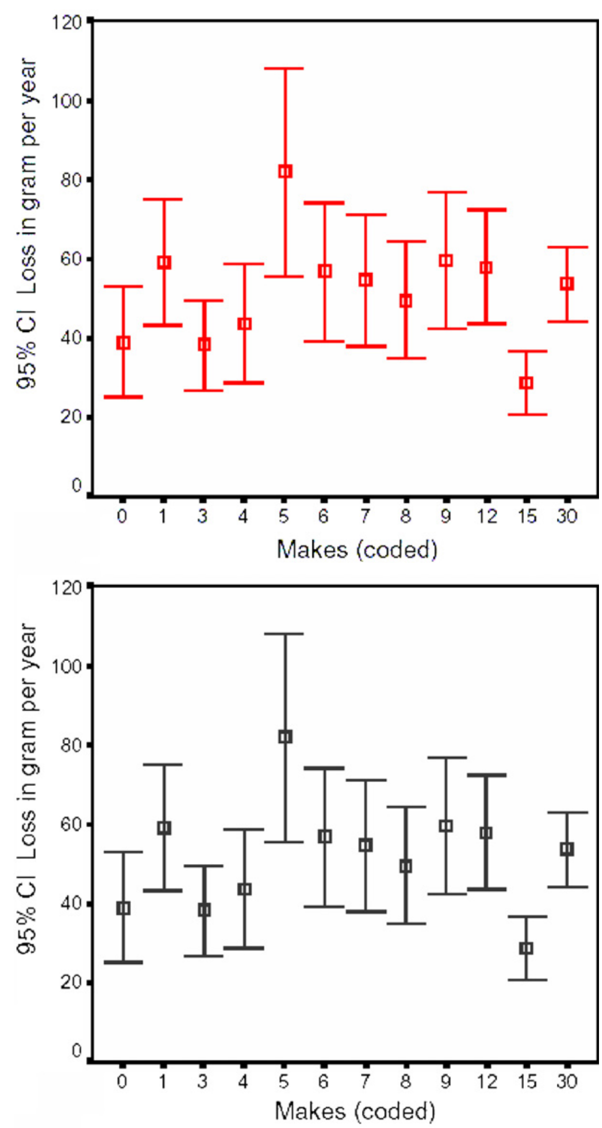

Fig. 1. Annual leakage rates from AC systems of cars of different makes (from [1]).

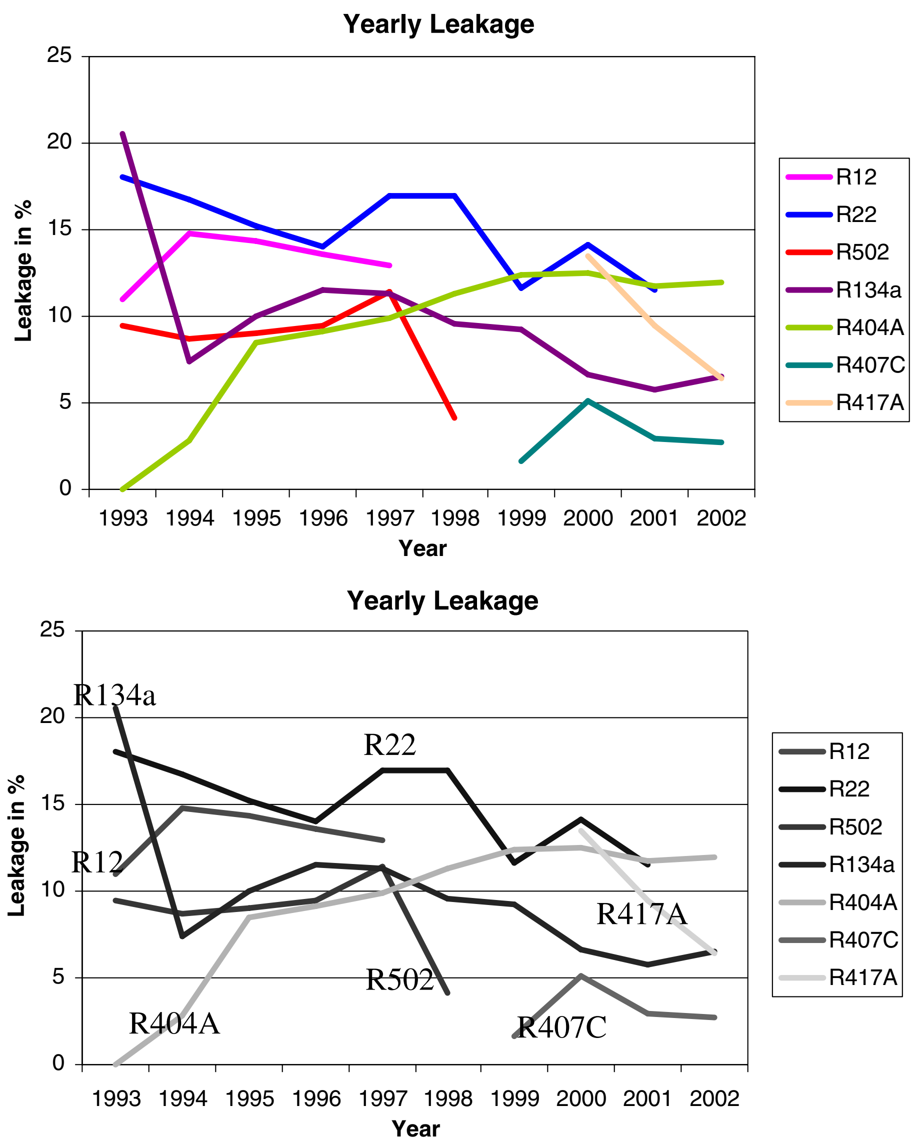

Fig. 2. Annual leakage rates (in %) reported for one large supermarket chain in Sweden (from [6]).

of systems, high tax on refrigerants, yearly commissioning of the system, better training of the personnel etc.

The conclusion seems to be that we cannot rely on being able to keep systems tight and that instead systems should be designed with as small internal volume as possible.

3. Direct or indirect systems

One way of drastically decreasing the charge of refrigerant in many types of systems is to change to indirect systems, i.e. systems where a secondary loop is used for distributing the heat to/from the condenser and the evaporator (Fig. 3). These secondary loops would normally contain water with glycol or other antifreeze added for operation at low temperatures.

With indirect systems, there are no long distribution lines filled with refrigerant and the refrigerant containing parts could be made extremely compact. The magnitude of the reduction in going from direct to indirect systems depends heavily on the application. Fig. 4 shows an example of a coldstore where the amount of refrigerant was reduced from 583 kg of CFC to 22 kg of HFC when the

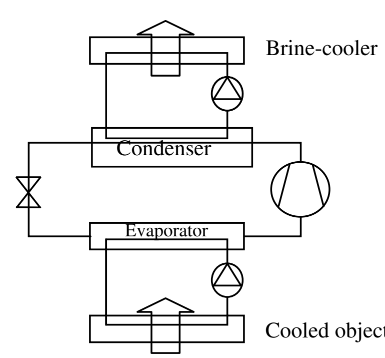

Fig. 3. Principle of indirect system.

Fig. 4. Illustration of refrigerant charge of coldstore before and after rebuilding as indirect system (from Scanref).

system was rebuilt as an indirect system. According to an industry-sponsored report [2], the charge of indirect supermarket refrigeration systems is expected to be 11% of the charge of a conventional centralized system. Another example is the reversible, indirect heat pump system developed and installed at the University of Valencia in the frame of the European 5th framework project GeoCool, for the development of an integrated ground source system for mediterranean type climates in Europe. The heart of this system consists of a scroll compressor and two brazed plate heat exchangers. The heating capacity is 17 kW and the system is charged with only 490 g of propane [6].

The disadvantage of indirect systems is the extra temperature difference introduced due to the additional heat exchanger needed between the refrigerant and the secondary fluid. Also, the pumping power necessary for circulating the secondary fluid has a negative influence on the energy efficiency and thereby on the total environmental impact of the system. However, these effects are at least partly balanced by the fact that in the direct system, the pressure drop in the distribution lines may correspond to a drop of several degrees in the saturation temperature. A measure of the environmental impact over the life time of the system is the Life Cycle Climate Performance (LCCP), measured as an emission of carbon dioxide causing a corresponding climatic impact. The LCCP has been compared between direct and indirect systems for a typical US supermarket in an industry report [2]. The result is shown in Table 1. As shown, the direct (from leakage)

Table 1
LCCP of direct and indirect system in a supermarket

Configuration	Refrigerant	LCCP million kg CO_2		
		Indirect effect	Direct effect	Total
Direct system	R404A/R507	11.7	12.1	23.8
Indirect system	R404A/R507	13.6	0.18	13.8
	Ammonia	13.6	0.0001	13.6

and the indirect (from electricity production) effects are comparable for the direct system. Also, the indirect effect is slightly larger for the indirect system due to the extra heat exchanger and the additional pumping power. This difference is about 16%. The direct effect of the indirect system is very small, resulting in the direct system having a total LCCP almost 75% higher than the indirect system.

The table also shows that under the assumptions made in the calculation, the direct effect of the indirect system is so low that an additional reduction of the charge would not influence the total LCCP in a substantial way. If this conclusion is assumed to be generally valid, further efforts of reducing the charge should be aimed at systems with flammable or poisonous refrigerants i.e. hydrocarbons and ammonia.

4. Design of a hydrocarbon heat pump

At the Royal Institute of Technology in Stockholm we have been working with the design of a heating only heat pump system for single family houses with a minimum of propane as refrigerant. The aim has been to reach a charge of 150 g at a heating capacity of 5 kW. A typical Swedish heat pump uses a secondary loop to collect heat from the ground or from a borehole in the rock. Heat is then delivered from the condenser to a hydronic heating system in the house. Usually, brazed plate heat exchangers are used as evaporator and condenser. In the laboratory, a heat pump of this type was built and tested as a baseline for further efforts in reducing the charge. To allow tracing the amount of refrigerant in the different parts of the system four quick closing valves were installed as shown in Fig. 5 (V1–V4). The most important components are specified in Table 2.

Note that the system had no receiver. This means that any excess charge will tend to fill up the condenser. The charge of refrigerant is therefore directly influencing the COP of the system. Tests were run with different amounts of refrigerant in the system, with constant evaporation and condensing temperatures (−2/+40 °C) and the COP was determined. At stable conditions, the compressor was stopped and at the same instant the four valves were closed. The four sections were then evacuated and the amount of refrigerant in each section was weighted. The results are shown in Figs. 6 and 7. As shown, the COP increased slightly with increasing charge up to the charge 300 g but decreased substantially at higher charges. At this optimum charge, the liquid line was filled with liquid but there was no liquid pool in the bottom of the condenser except that which was trapped under the level of the outlet. Looking at the distribution of the charge, it is clear that the

Table 2
Components in baseline heat pump system (from [4])

	Type
Compressor	Copeland ZR28K3E-TFD-522
Condenser	SWEP B15x30
Evaporator	SWEP B25x16

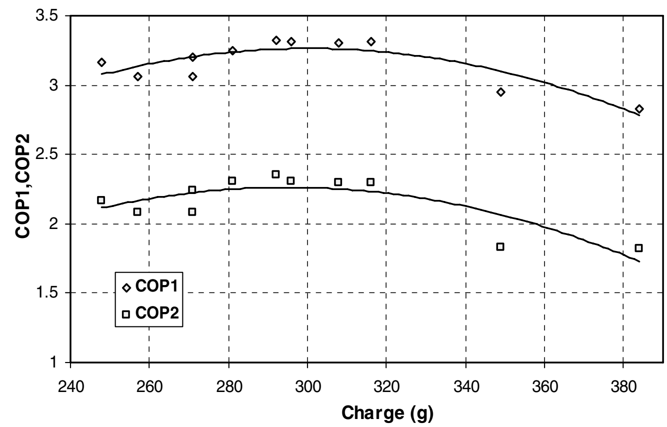

Fig. 6. COP of baseline heat pump system as a function of the charge of propane (from [4]).

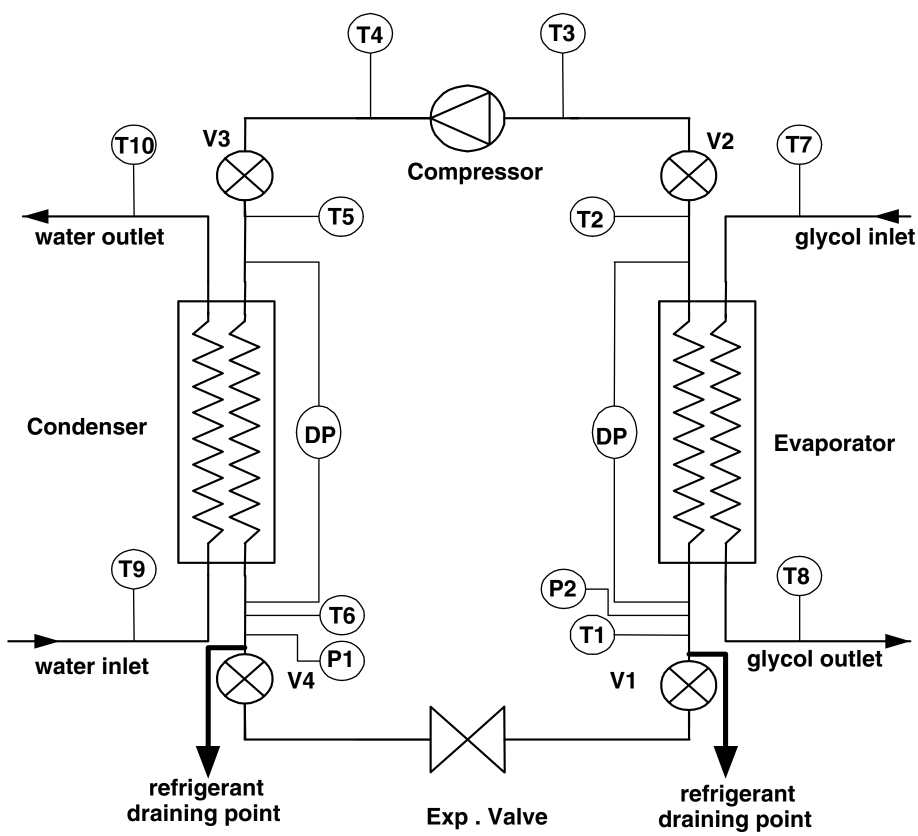

Fig. 5. Schematic drawing of heat pump system with quick-closing valves (from [4]).

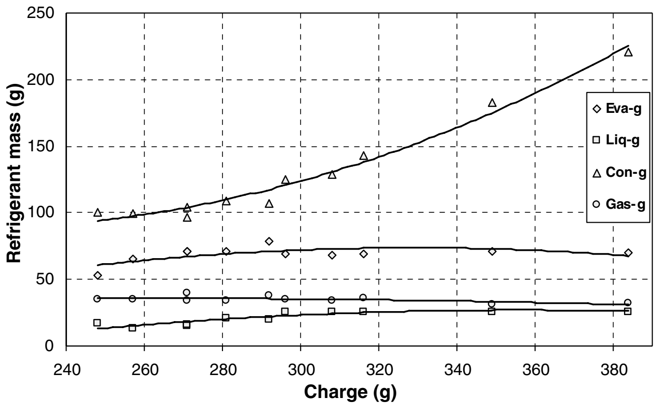

Fig. 7. Charge distribution in baseline heat pump system as a function of total charge (from [4]).

largest amount of refrigerant was in the heat exchangers, and primarily in the condenser (including the short part of the liquid line up to the quick-closing valve).

It should be noticed that the sum of the amounts shown in Fig. 7 do not add up to the total amount charged into the system. This is because approximately 40 g of propane was solved into the oil of the compressor and could not easily be evacuated from the system.

The lessons learned from these tests were the following:

- Just using standard components, but designing for minimum charge, quite low total amounts of refrigerant may be reached.
- The components holding the largest amounts of refrigerant were the heat exchangers.
- Quite a large amount of refrigerant was dissolved in the compressor oil. A different compressor design, or the use of a non-soluble oil would decrease this amount.

To further decrease the charge, the plate heat exchangers were substituted for a set of heat exchangers made from small, flattened copper tubes inside a shell (Fig. 8). 60 tubes were connected in parallel in two rows with a distance of 1 mm in between the tubes to form the unit shown in Fig. 8. Two such units were used in parallel as evaporator and two units in series as condensers. The hydraulic diameter of the inside was about 1.1 mm.

Tests were run with this system at different heat source temperatures from −16 to +5 °C with different charges of propane, keeping the condensing cooling water outlet temperature at +40 °C. The results showed that the COP of the system was dependent on the charge, and that the highest COP was reached at different charges at different evaporation temperatures. The optimum charge at around 0 °C evaporation temperature is about 230 g and thus considerably lower than with the plate heat exchangers.

Within the project at the Royal Institute of Technology we have also designed a new liquid to refrigerant heat exchanger to be used as condenser and evaporator. Prototypes of this new design has been manufactured and tested

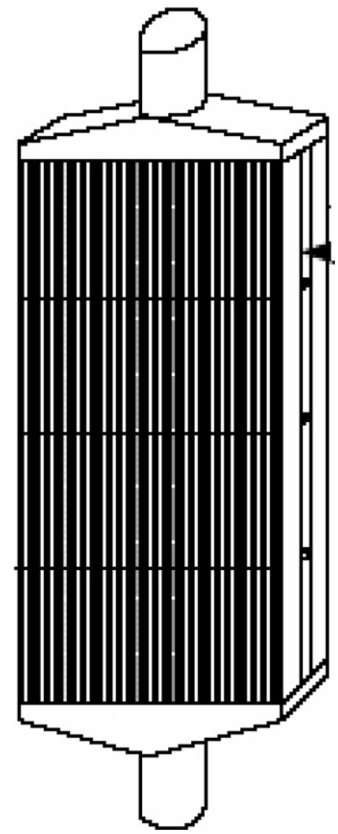

Fig. 8. Flat copper tube heat exchanger.

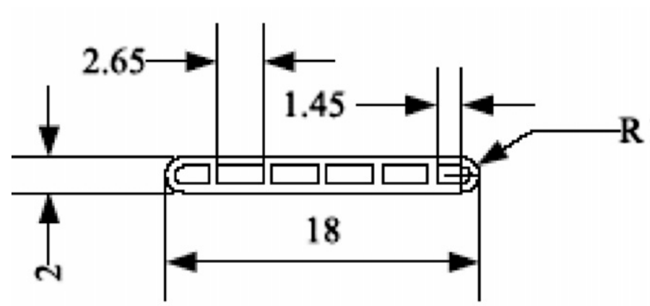

Fig. 9. Profile of multiport aluminium tube used for evaporator and condenser shown in Fig. 10 (from [5]).

in a similar way as the brazed plate heat exchangers and the flat copper tube heat exchangers. This new type is made from flat multiport aluminium tubes with channels having hydraulic diameters of about 1.4 mm (Fig. 9).

The flat tubes were arranged in two rows and placed within a shell with baffle plates directing the secondary refrigerant back and forth in between the tubes. Pictures of one of the heat exchangers are shown in Fig. 10. The length of the tubes was 651 mm, the total refrigerant surface area was chosen to be similar to the baseline plate heat exchangers, i.e. 0.82 m^2 for the evaporator and 0.98 m^2 for the condenser.

The test results with these heat exchangers are shown in Fig. 11. As for the other heat exchanger types, the optimum charge depends on the evaporation temperature. For evaporation temperatures close to zero (heat source temperature +6 °C) the optimum charge seems to be about 230 g. This is about the same as for the flat copper tube heat exchanger, but the performance in terms of the UA-value was considerably higher for the new design. At 200 g charge and the evaporation temperature −2 °C, the charge was distributed as follows: Evaporator, 23 g, condenser, 80 g, liquid line, 24 g and compressor, 74 g. It is expected that new header design may decrease the charge in the evaporator and the condenser somewhat more.

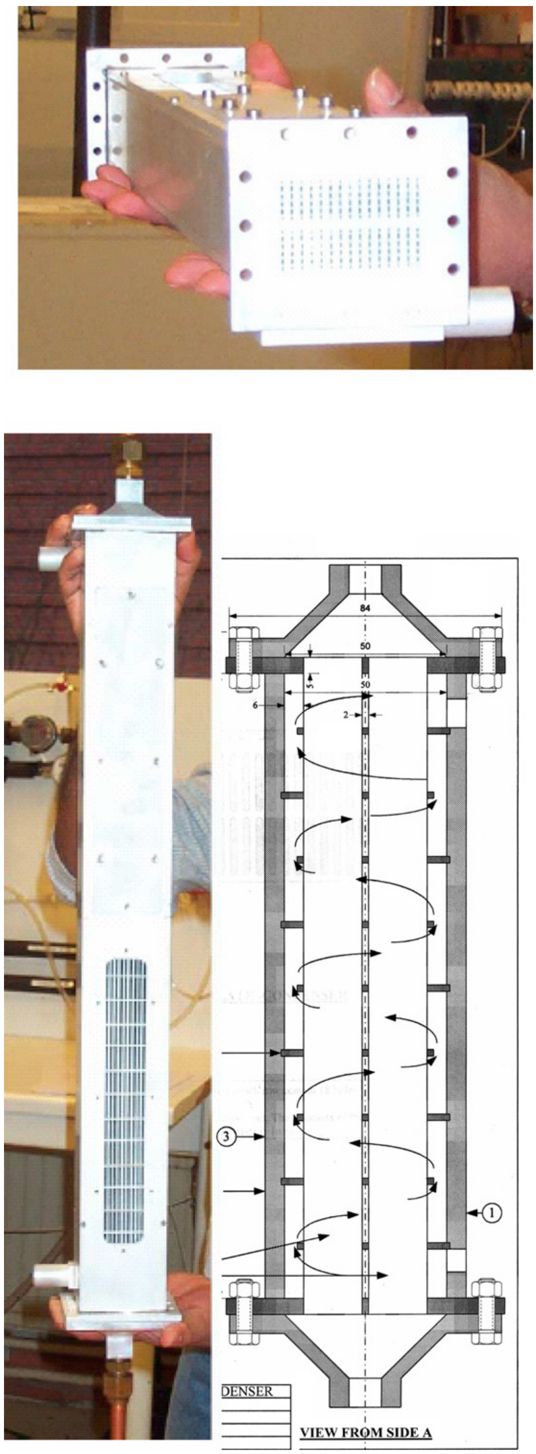

Fig. 10. End view and front view of multichannel aluminium heat exchanger (from [5]).

With this type of heat exchangers, the amount of refrigerant in the compressor is no longer insignificant. On the contrary, it is comparable to that in the condenser and three times that in the evaporator. There are different ways of reducing this amount: First, it may be possible to use oil which is not soluble in the refrigerant. For most applications, mineral oil can be used with propane. However, as the propane is highly miscible with this type of oil, it may result in a substantial decrease of the viscosity, which may be harmful to the compressor. Propane is also highly soluble in ester oils (POE) so a change to this type will not solve the problem. However, some alkylene (PAG) oils are not miscible with propane and should therefore be possible to use to reduce the amount of propane in the compressor [3].

It may be argued that if the propane is absorbed in the oil it cannot easily escape into the atmosphere, and therefore is less of a risk. Even though this may be true under most circumstances, it is possible to imagine conditions under which this propane will be released, and therefore it is not likely that safety regulations and standards will take this into consideration.

Using a non-miscible oil may influence heat transfer in the evaporator and condenser as the heat exchange surfaces may be covered by a thin layer of oil. From this point of view, a miscible refrigerant is preferable.

The second option to decrease the amount of propane in the compressor is to use a different type of compressor with less oil and smaller internal volume. Compressors used in the automotive industry have these features. Fig. 12 shows a figure of such a compressor. This specific type has also been developed for applications in hybrid cars where the compressor needs to be electrically driven. Fig. 12 shows the complete unit, compressor and motor and in the bottom part a cut view of the compressor.

The possible gains by changing to this type of compressor can be seen from Table 3. It is obvious that the reduction in the volume and the decreased amount of oil will result in a substantial reduction in the amount of propane within the compressor.

Designing for low charge may open the possibility for new system solutions. With indirect systems, the evaporator and the condenser may be located within centimeters from each other. It may also be possible to run such a system with a low-pressure receiver in the suction line rather than the common high pressure receiver in the liquid line. The advantages with this solution are the following:

- A capillary tube may be used as expansion device
- With the capillary tube mounted directly onto the condenser outlet no charge is trapped in a liquid line, which should reduce the charge

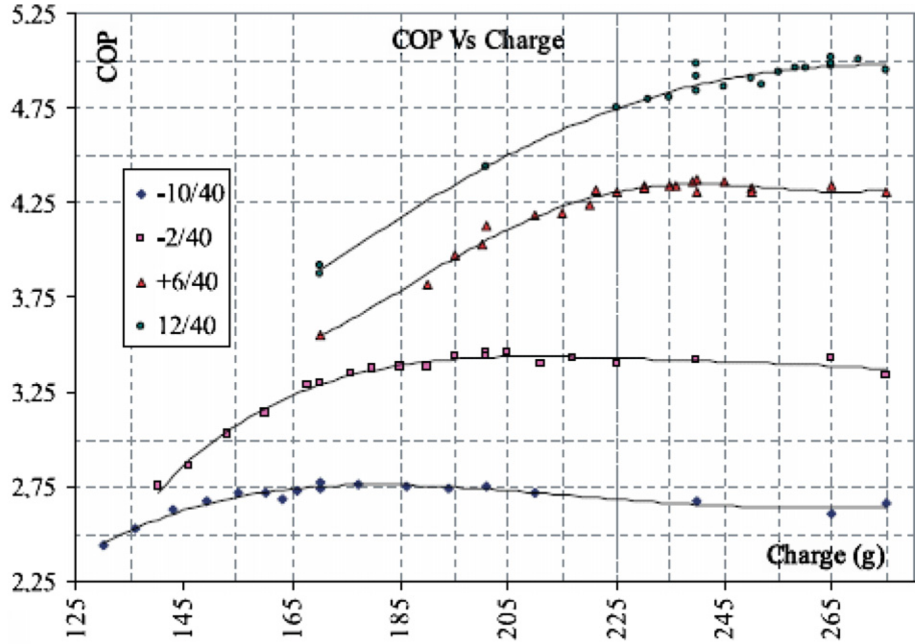

Fig. 11. COP1 vs charge for multichannel aluminium heat exchangers at different evaporation temperatures (from [5]).

Fig. 12. Compact automotive compressor, with and without electric motor attached (Sanden Corp.).

Table 3
Comparison between traditional hermetic compressor and semi-hermetic compressor developed for hybrid car

Type of compressor	Traditional	For hybrid car
External volume, dm^3	17	5
Mass of propane, g	70–80	<20
Weight, kg	29	10
Oil charge, cm^3	1000	170

Mass of propane is estimated.

- With a low pressure receiver and the correct charge of refrigerant all evaporator surfaces will be used for evaporation (no superheat at the evaporator outlet) in all but the most extreme conditions, which will enhance heat transfer and COP.

The system described is frequently used in household refrigerators but is not common in larger systems.

When designing for low charge it is important not to forget the system performance: Reducing the charge at the cost of lower system performance is not beneficial to the environment. However, it is our experience and belief that it is possible to design for minimum charge without reducing system performance.

5. Conclusions

This paper has discussed the reasons for, and the possibilities to, decreasing the refrigerant charge in refrigeration and heat pump systems. Due to the large contribution to the greenhouse effect caused by all commonly used HFC refrigerants it is necessary to reduce the atmospheric emissions. A reduction of the charge in the systems is important to reach this goal. Reduced charge is achieved by redesigning centralized systems to operate as indirect systems with secondary refrigerants on both the hot and the cold side. Heat exchangers with small internal volume on the refrigerant side should be used.

The second option to reduce HFC leakage is to use other types of refrigerants, primarily hydrocarbons and ammonia. As these refrigerants are flammable it is important to reduce the charge for safety reasons.

From measurements on a domestic water to water heat pump it is known that most of the refrigerant is located in the heat exchangers during operation. With heat exchangers based on mini-channel technology it is possible to reduce the charge substantially, up to the point where the refrigerant in the compressor is comparable to that in the heat exchangers.

The amount of refrigerant in the compressor may be reduced by using non-miscible oils or by using compressors with small internal volume and small oil charge.

Systems with low charge may be built with capillary tube expansion devices and low pressure receivers, i.e. with system solutions similar to most refrigerators.

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