DESIGNING A HEAT PUMP FOR MINIMUM CHARGE OF REFRIGERANT

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ABSTRACT

Independently of which refrigerants will be used in the future there are strong incentives to reduce the charge of refrigerant in each system. This paper reports on results from a project aiming at reducing the charge of refrigerant in a 5 kW liquid to liquid heat pump (heating only) to 150g of propane. First, the fluid inventory in each part of the heat pump was determined. As a result, the main effort has been on redesigning the heat exchangers using mini-channel tubes. At the present time, the system operates with about 200 g of propane. Further decrease of the charge is possible by redesigning the condenser and by reducing the amount of propane in the compressor, either by using oils in which propane is not soluble or by using compressors with low charge of oil. The designs suggested may open up for the safe use of flammable refrigerants with high energy efficiency such as R152a, R32, R290 and R600a.

Key Words: ***minimum charge, fluid inventory, charge reduction, heat pump***

1 INTRODUCTION

During the last ten years it has become generally accepted that the average global temperature is increasing. Today, it seems highly probable that this increase is a result of human activities causing an increase in the concentration of carbon dioxide and other greenhouse gases in the atmosphere. Among these other gases, recognized in the Kyoto protocol as being of importance, are found most of the chemical compounds used as refrigerants in heat pumps and refrigeration equipment, i.e. hydrofluorocarbons (HFCs) and hydrochlorofluorocarbons (HCFCs). For this reason, both environmentalists, researchers, government officials and representatives of industry have advocated the use of natural refrigerants with low global warming potential (GWP) as a means of reducing the contribution to global warming caused by gases released from refrigeration equipment and heat pumps. Natural refrigerants are commonly understood to be hydrocarbons, ammonia, carbon dioxide and water, all of which are present in large quantities in the environment due to natural processes. As the amounts released from heat pumps and refrigeration equipment is small compared to the naturally circulating quantities, the influence on the global environment from the use of these fluids as refrigerants must be negligible. Also, as opposed to the anthropogenic HFCs and HCFCs, there is no risk of introducing new, unknown threats to the environment by using natural fluids as refrigerants. However, a change to natural refrigerants is not without its problems, different for each of the fluids.

Hydrocarbons have properties similar to the HCFCs and HFCs used today and could be used with only minor changes to today's systems. An important difference, however, is that the hydrocarbons are highly flammable and thus special precautions have to be taken to reduce the risk of accidents.

Ammonia is also flammable under certain conditions, and more importantly has a strong smell which may cause panic if the gas is released. At higher concentrations it is also poisonous. Ammonia is highly corrosive to copper and cannot be used in present systems designed for HFCs. A special problem is the fact that hermetic compressors cannot be used as the electric motor windings are made of copper.

Carbon dioxide (CO_2) is being introduced as an environmentally friendly, non-flammable and non-toxic (at low concentrations) refrigerant. Compared to present day systems, CO_2 will give much higher system pressures (72 bar at 30°C). Also, as the critical point of CO_2 is low (31°C) the process will be trans-critical if temperatures above this value are required. The low critical point results in special system designs being necessary to reach reasonable COPs in many applications.

Water, finally, is limited to temperatures above zero °C. Its saturation pressure is very much lower than all other refrigerants and for this reason the dimensions necessary for tubing and compressors is excessive.

From an environmental point of view, not only the direct effect caused by release of refrigerant into the atmosphere should be considered. Equally important is the secondary effect, i.e. the release of carbon dioxide due to the generation of the electricity necessary for operation of the system. Whereas the direct effect is measured by the global warming potential (GWP) of the fluid, the total contribution to the global warming is measured by the Total Equivalent Warming Impact (TEWI). For systems with low losses of refrigerant, the indirect effect will be dominating, while for systems with large losses, the direct effect will have the largest contribution to global warming. In any case, the TEWI concept points to the importance of designing systems with high COP, not only for economic reasons but also to limit the environmental impact. Of the natural refrigerants, hydrocarbons and ammonia seem to have the potential of giving (at least) as high COP as the HFC and HCFC refrigerants. Hydrocarbons can be introduced with only minor changes in the present technology.

2 BASELINE HEAT PUMP DESIGN

Independently of if the refrigerant used is a hydrocarbon, ammonia or an HFC, there are incentives to seek to reduce the charge in the system, either for safety reasons or for concern for the environment. As noted, the COP is highly important for the environmental impact and the reduction in charge should preferably be achieved without reduction of the COP.

At the Royal Institute of Technology in Stockholm a government sponsored project has been running for some years aiming to demonstrate how charge minimization may be achieved without reduction of COP. The focus of the project has been on single family heat pumps as they are normally designed for the Swedish market. These heat pumps are designed for heating only, using a secondary loop with for absorbing heat either from a vertical borehole in the rock, from the top soil or from the bottom water of a lake or the sea. These types of systems are becoming very popular in Sweden with more than 200000 units installed, representing more than 10% of the total no of single family houses. Figure 1 shows a schematic view of a typical system, in this case also including the possibility to use the cold water coming from the heat pump for cooling purposes. The heat pumps are usually connected to the hydronic heating system of the house and to a tank for sanitary hot water. Heat is distributed by radiators, convectors or floor heating systems. The distribution temperature of the water is typically 50°C or less at design conditions. This type of system is already from the start quite compact with all refrigerant-containing parts housed within a small volume. The evaporators and condensers used are almost exclusively brazed plate heat exchangers, which have quite

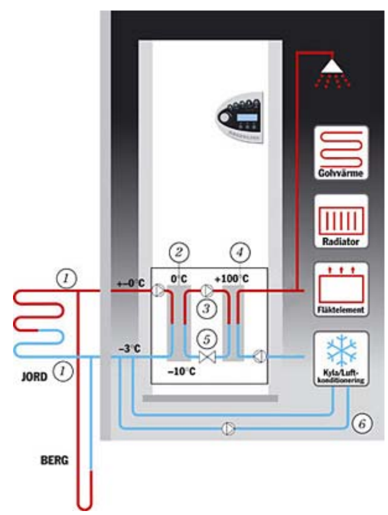

Fig. 1. Typical heat pump system design for the Swedish market. (From IVT)

small internal volume. The total refrigerant inventory for heat pumps in the range 4-10 kW (heating) range from 1 to 2,5 kg HFC (R407C, R134a).

3 AIM OF PROJECT, DESCRIPTION OF TEST RIG

The aim of the project has been to design a heat pump system for a single family house (at least 5 kW heating) using less than 150g of propane. The reason for choosing this level is that this small amount is considered as safe according to some European norms and that the necessary safety requirements therefore are less demanding below this level. As the density of propane is about half of that of the HFCs, the target charge expressed in volume is 1/3 to 1/8th of that of an ordinary compact system. In a first phase of the project a model heat pump was built in the laboratory using standard components; a hermetic scroll compressor, two brazed plate heat exchangers and a thermostatic expansion valve. No receiver was used. All components were mounted to achieve as short interconnecting lines and as small internal volume as possible. Apart from the standard components, the system was supplied with four quick closing valves which could be closed simultaneously while the system was running. The purpose of these valves was to allow the measurement of the mass of refrigerant in the different parts of the system under operation. The valves were placed as shown in figure 2, in between each of the four main components of the system.

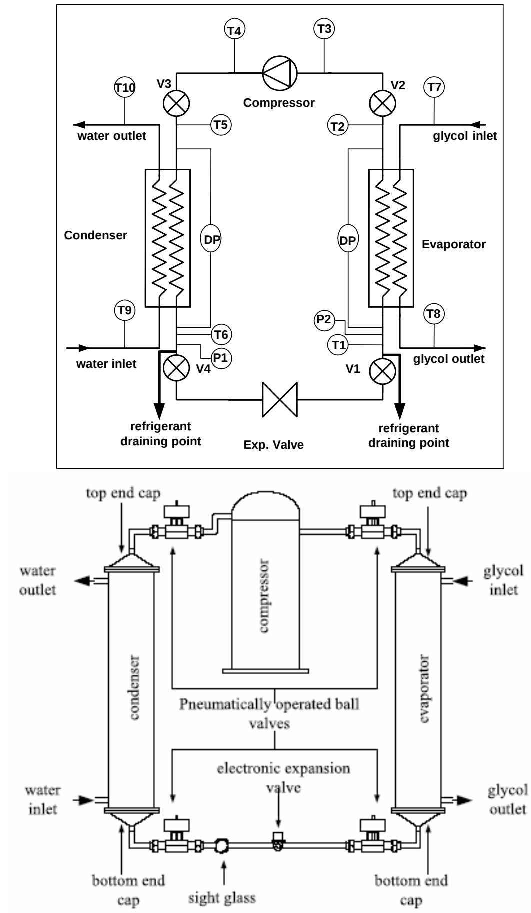

Fig. 2. Schematic drawings of test setup. (From Fernando et al.

4 TEST PROCEDURE

The tests were performed in the following way: The operating conditions of the system (evaporation and condensing temperatures) were fixed by choosing the volume flow and inlet temperatures of the heat source fluid (glycol solution) and the heat sink fluid (water). When the system had been operating stably for some time, the quick-closing valves were closed and simultaneously the compressor was stopped. Each of the four sections was then evacuated of all refrigerant by connecting them one at a time to an empty cylinder immersed into liquid nitrogen. Due to the low temperature, all refrigerant in the section was condensed into the cylinder, which could then be disconnected and weighted. Knowing the initial weight, the mass of refrigerant in the section was thus determined.

5 BASELINE SYSTEM

Tests were run with different total amount of refrigerant in the system and the performance was determined for each charge. Figure 3 shows some of the results. As shown, for the selected set of running conditions, (evaporation temperature -9°C, condensing temperature +40°C) there is a specific charge for which the performance is the highest. For the type of systems studied, the evaporation temperature varies a few degrees over the heating season. In a commercial system, a small receiver should be included to take up the variations in optimum charge. However, as the variation in optimum charge is a function of the total charge, the necessary volume of this receiver is quite small.

Fig. 4 shows the distribution of the charge between the sections of the system. As shown, charging the system above the optimum level increases the refrigerant mass in the condenser while the charges in the other parts remain constant. From figs 3 and 4 it is also obvious that 300 g of propane is perfectly sufficient to operate this baseline system, built from only standard components. Even 250 g would be enough, giving slightly lower COP and lower heating capacity of the system. Looking once again at fig. 4 it is clear that the two heat exchangers (and the short connecting tubing up to the valves) are the parts containing the largest mass of refrigerant.

6 SYSTEM WITH FLAT COPPER TUBE HEAT EXCHANGERS

In a first effort to reduce the charge, a commercial type of heat exchanger based on flat copper tubes was tested. These mini-channeled flat tube heat exchangers were made from units (figure 5) consisting of 60 parallel flat rectangular-shaped copper tubes in two rows and connected at each end to two common headers. The distance between two tubes is 1mm. The length of a tube is 0,4 m and the internal hydraulic diameter is 1,096mm (cross section 0,6x 6,3 mm^2). The total tube area of the refrigerant side is 0,331 m^2 and the total internal tube volume is only 90,7 ml. The tube package is enclosed in a shell with baffles and the brine/water is passed back and forth in between the flat tubes. In this case, two packages were connected in series as condenser and two in parallel as evaporator as shown in fig. 6. Test results with these heat exchangers are shown in fig. 7 in terms of COP1 vs. refrigerant charge at different evaporation

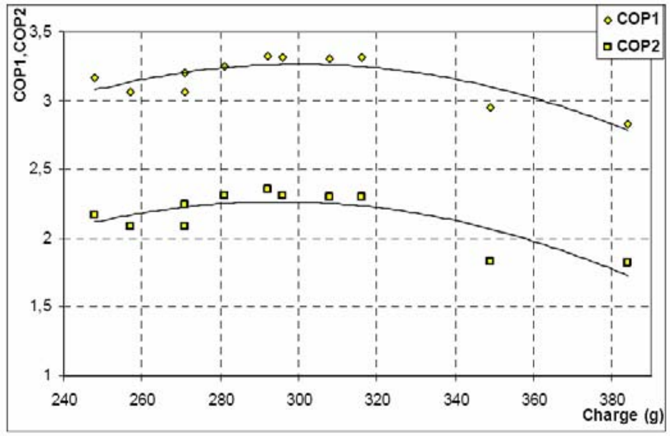

Fig. 3. COP of baseline system vs refrigerant charge. (From Fernando et al, 2001)

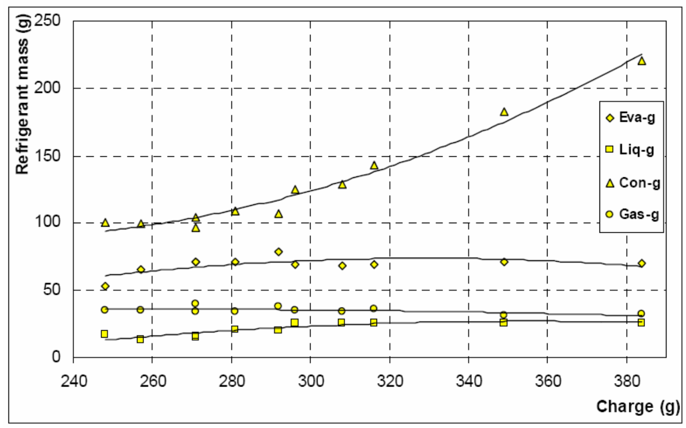

Fig. 4. Refrigerant distribution between the four sections in the baseline system. (From Fernando et al, 2001)

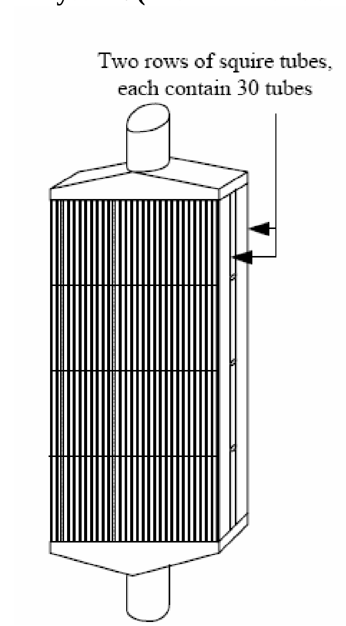

Fig. 5. Unit of flat copper tubes used for evaporator and condenser as shown in fig 7. (From Fernando et al, 2002)

temperatures. The dependence of the optimum charge on the evaporation temperature is quite obvious. At -16°C 145 g of refrigerant is enough to reach very close to the maximum COP1, whereas at the evaporation temperature +5°C about 250g of propane gives the highest COP1.

Comparing the optimum charge to that with the plate heat exchangers (at evaporation temp -8°C) it is clear that the flat copper tube heat exchangers have resulted in a decrease of charge of about 100g. Not shown, however, is the fact that the temperature differences in both the evaporator and the condenser were larger with the flat tube heat exchangers than with the plate heat exchangers. In an actual installation where the heat source and heat sink temperatures are fixed rather than the evaporation and condensing temperatures, this would have resulted in a considerable decrease in the performance.

7 ALUMINUM MULTICHANNEL HEAT EXCHANGER

Having difficulties finding a sufficiently compact heat exchanger on the market it was decided to develop a new prototype similar to the design of the flat copper tube heat exchanger but based on extruded aluminum multi-channel tubes. A cross section of the tubes used in the first design is shown in fig. 8. Two heat exchangers were manufactured from these tubes, one as evaporator, one as condenser. The evaporator consisted of 30 tubes and the condenser of 36 tubes. In both cases the tubes were fixed into a shell with 31 baffle plates directing the brine/water back and forth in between the tubes, forming channels with the height 2 mm (figs 9a and b). The heat pump was tested with these heat exchangers in a similar fashion as described above. Fig. 10 shows the measured COP1s at different charges. In this case the temperatures indicated in the figure are the heat source and heat sink temperatures. As shown in Table 1, the data for -2/+40 corresponds to

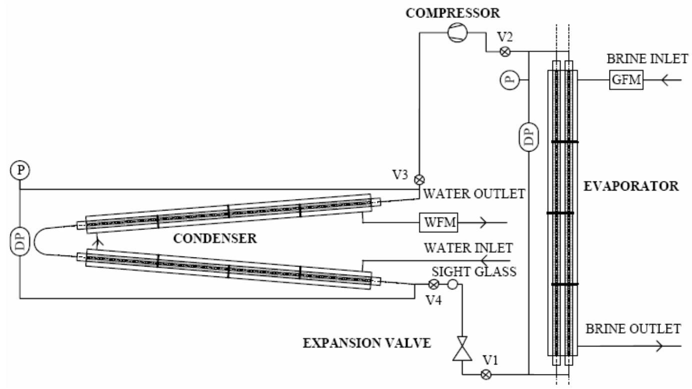

Fig. 6. System with flat copper tube heat exchangers as evaporator and condenser. (From Fernando et al, 2002)

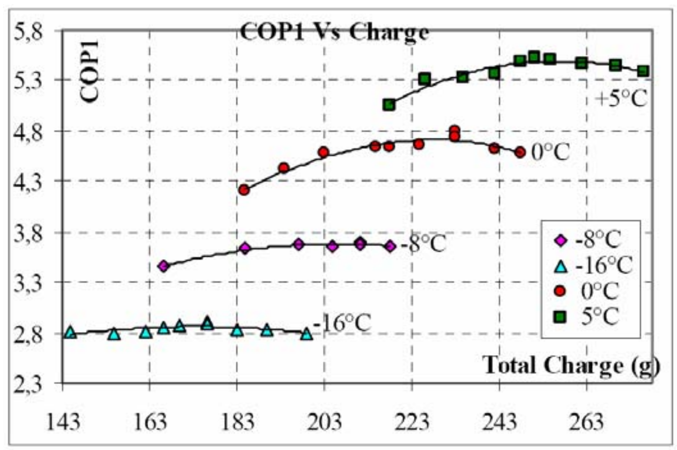

Fig. 7. Variations in coefficient of performance (COP1) with the total amount of refrigerant charge in the test rig with flat copper tube heat exchangers, at evaporation temperatures -16°C, -8°C, 0°C & +5°C with constant condenser cooling water outlet temperature (39-40 °C). (From Fernando et al, 2002).

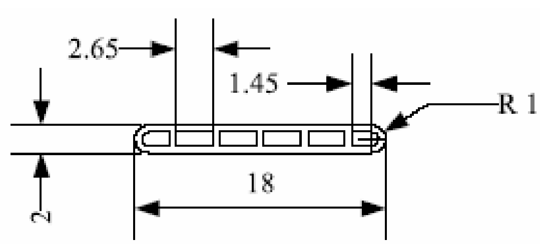

Fig. 8: Profile of aluminum multi-channel tube (From Fernando et al, 2004)

the evaporation/condensing temperatures -9/+40 as discussed above. Under these conditions the optimum charge is about 200g and even at 185g the COP1 is very close to the maximum.

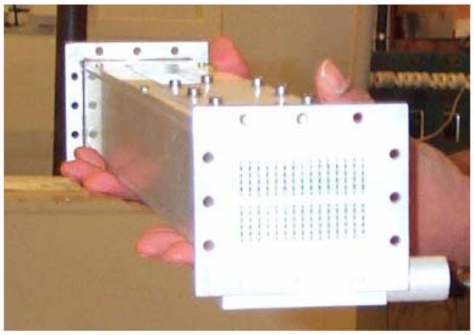

Fig. 9a. End view of aluminum heat exchanger without end caps. (From Fernando et al, 2004)

Table 1 also shows the refrigerant distribution at the charge giving the highest COP1. As shown, the section around the condenser still contains the largest part of the refrigerant at the two lower temperatures. However, at higher temperatures, the compressor alone contains more than 100 g of propane and thereby has the dominating part of the charge. The evaporator, on the other hand, contains only about 25 grams of propane, which must be considered quite satisfactory.

8 REDUCING THE CHARGE IN THE COMPRESSOR

The reason for the large charge in the compressor is mainly the absorption of propane in the compressor oil. This propane constitutes less of a risk than that that in the other parts of the system as it will not readily be desorbed from the oil. However, it would be possible to reduce this amount by using an oil in which propane is less soluble. By measuring the vapor pressure of mixtures of propane and different oils it has been shown that propane is less soluble in PAG oils than in POE or mineral oils. In fact, the solubility in some PAG oils is very low.

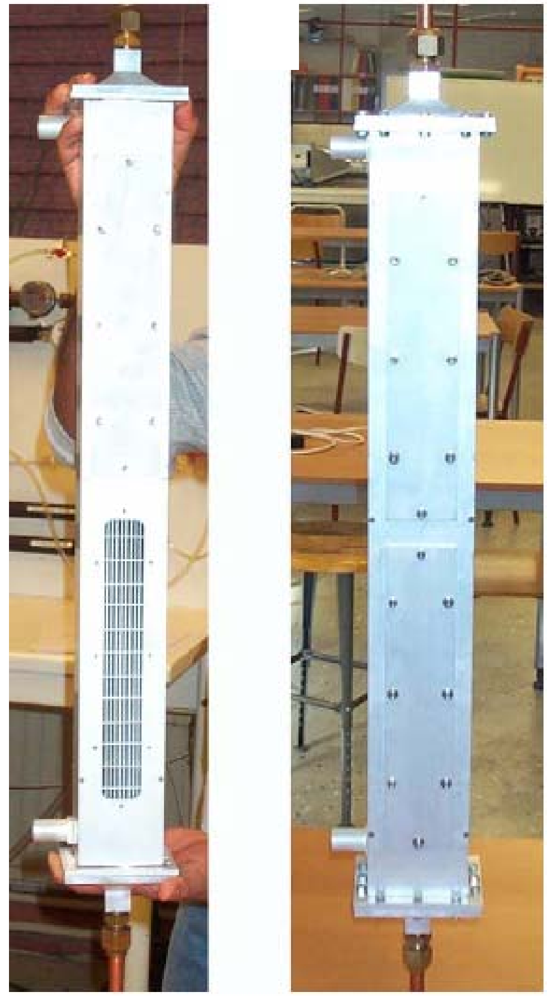

Fig. 9b. Side view of aluminum heat exchanger with and without sidewall open showing flat aluminum tubes and baffle plates. (From Fernando et al, 2004)

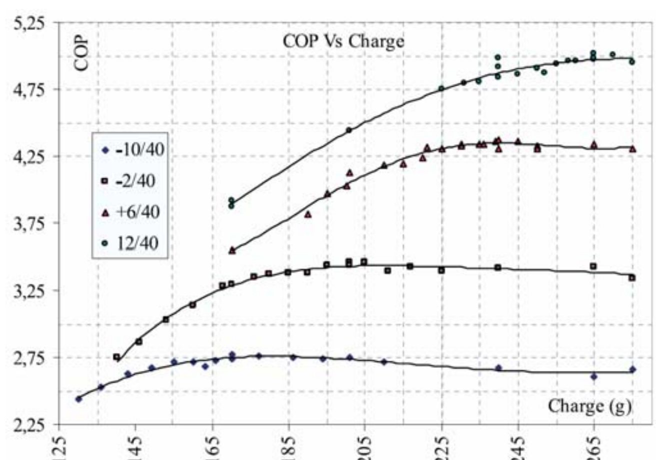

Fig. 10. COP1 vs. charge using prototype aluminum heat exchangers. Temperatures indicated are heat source and heat sink temperatures. (From Fernando et al, 2004)

Table 1. Test conditions and refrigerant charge distribution with aluminum multiport heat exchangers.

Heat source/heat sink temperatures (°C)	−10.22/40.77	−2.09/40.67	6.93/40.77	12.64/40.74
Evaporation/condensing temp. (°C)	−16.49/39.73	−8.79/40.44	−0.51/41.44	4.55/39.95
Evaporator/condenser capacity (kW)	2.67/4.00	3.66/5.01	4.90/6.31	5.89/7.23
Super heat/sub cool (K)	4.66/3.91	5.30/4.76	5.18/4.56	5.77/3.82
Optimum refrigerant charge (g)	170	201	240	265
Measured in evaporator (g)	27	23	25	26
Measured in condenser (g)	69	80	90	93
Measured in liquid line (g)	24	24	23	24
Measured in compressor (g)	50	74	102	122

Another way of reducing the amount of refrigerant in the compressor would be to use compressors with smaller internal volume and lower oil charge. We may compare to the types of compressors (scroll or swashplate) used in the automotive industry. There are already on the market compressors of this type supplied with an electric motor, designed for use in hybrid cars. Table 2 gives a comparison between one such compressor and the compressor used in the tests. At the present time we are preparing for running tests in our laboratory with such a compressor.

Table 2. Comparison of traditional hermetic compressor and compressor designed for hybrid car (approximate values)

Compressor	*Traditional*	*Hybrid car*
External volume [dm^3]	**17**	**5**
Mass of propane [gram]	**66**	**<20**
Weight [kg]	**29**	**10**
Oil content [cm³]	**1000**	**170**

9 CONCLUSIONS

Charge reduction has up to now not been an important goal for the heat pump industry. With new legislation concerning the use of fluids with high GWP and with the introduction of flammable refrigerants this may be an important factor in the future. In this paper it has been demonstrated that there is a large potential for charge reduction by using compact heat exchangers and short interconnecting lines between the components of the system. It has also been shown that a substantial amount of refrigerant may be contained in the compressor, mainly dissolved in the oil. This amount may be reduced by using compressors with small internal volume, small amount of oil and/or by using oils which are not miscible with the refrigerant.

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