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Laboratory Evaluation and Field Tests of Replacements for R22 and R502

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Abstract

There are several important issues when dealing with new refrigerants in a retrofit situation. This paper is however restricted to system performance and a discussion of methods suitable to determine the performance.

The cooling capacity and COP are strongly dependent on the system design (type of heat exchangers, compressors, receivers etc.). The difference between two refrigerants in the same system is usually much smaller than two different systems operating with the same refrigerant. The studied systems are therefore both plate-type heat exchanger systems with small charge and larger chillers with shell and tube type heat exchangers. The paper deals with some of the problems that may occur in the latter type of equipment.

Experiences of methods and limitations for field evaluation are given. The differences with pure and a zeotropic refrigerants are discussed with respect to this problem. The importance of collecting refrigerant samples for gas chromatography is stressed.

Introduction

The retrofit activities in Sweden have, so far, been focused on refrigeration and heat pump plants operating with R12, R500 and R502. The schedule for phase-out is described in figure 1. The timetable means a real challenge for the Swedish refrigeration industry and, above all, a hard pressure on the users. The retrofit of R12 plants is now believed to have reached 50% of the refrigerant charge! The alternative here is R134a. The introduction of R404A for R502 plants has met minor competition from the zeotropic alternatives R407A and B.

ASHRAE Number	Primary replacement	Type of refrigerant	Stop for import or new installations	Stop for refill	Stop for use	Share of the total refrigerant charge[1] in Sweden /1/
R12	R134a	CFC	1/1 1995	1/1 1998	1/1 2000	32%
R500	R134a	CFC	1/1 1995	1/1 1998	1/1 2000	6%
R502	R404A[2]	CFC	1/1 1995	1/1 1998	1/1 2000	12%
R22	R407C	HCFC	1/1 1998	1/1 2002	-	50%

Table 1. *Schedule for the phase-out of ozone depleting refrigerants in Sweden*

One aim of this paper is to describe, and compare, the methods for evaluation of alternative refrigerants that have been used at our department in cooperation with the Swedish refrigeration industry. We believe that the work described here has been an important part of the "knowledge formation" required for the retrofit activities on the field. Parallel with the work described here is a work dealing with the practical retrofit methods coupled to the chemical environment in the refrigeration circuit after, for example, a conversion from R12 with mineral oil to R134a with ester oil (this work is presented in another paper at the conference).

The work is mainly sponsored by NUTEK as a joint research program called "Alternative Refrigerants". Parts of the work reported here are also sponsored in-house by several individual companies involved in the field. Some of the work have been performed by the students at Applied thermodynamics and Refrigeration, KTH, as master of science projects. The program "Alternative Refrigerants" involves several other projects such as natural refrigerants, heat transfer studies etc.

[1] Other refrigerants such as R11, R13, R13b1 etc have been omitted
[2] Some competition from R407A and B

Methods for evaluation of the performance

No standard procedures have been adapted for the tests. The overall idea has been to systematically vary as many parameters as the particular piece of equipment, time and cost allows. A typical laboratory test of a R22 replacements may typically involve three condenser temperatures (30, 40 , 50°C) and a number of different evaporator temperatures (see fig. 6). A field measurement may give **one** operating point!

The test may be categorized into the following groups:

- Simple field tests
- Rigorous field tests
- Laboratory drop-in tests
- Full laboratory evaluation

We hope that the field oriented tests, in cooperation with local refrigeration service companies, can be used *as a bridge* between the laboratory/theoretical work in our laboratory and the practical work on the field. Simple and reliable measurement methods for field use are an important part of the introduction of all new refrigerants.

Equal conditions

Several retrofits reported in the literature are claimed to have been evaluated under *equal conditions*. One example of difficulties involved with this is how to define subcooling and superheat for zeotropic refrigerants . A definition using the mean condensing and evaporating temperatures are much more realistic, but unfortunately, too complicated for field use /3/. Equal conditions means, in this paper:

for a field drop-in situation:	Equal external conditions plus that nothing is changed in the refrigeration circuit except for an adjustment of the expansion valve and the refrigerant charge (soft-optimization).
for a laboratory evaluation:	Equal evaporation and condensation temperatures defined according to the AREP definitions for zeotropic refrigerants /4,5/. Subcooling has been the resulting subcooling when an equal condeser heat exchanger area have been occupied and the superheat has been the lowest possible with stable operation.

Simple field tests

These tests are categorized by the lack of stable conditions, difficulties to mount pressure transducers (if used at all) and thermocouples. The method chosen here is to monitor the electric power to the compressor and then, in some way, estimate the cooling capacity. The example presented here is a milk cooler retrofitted from R22 to R407C.

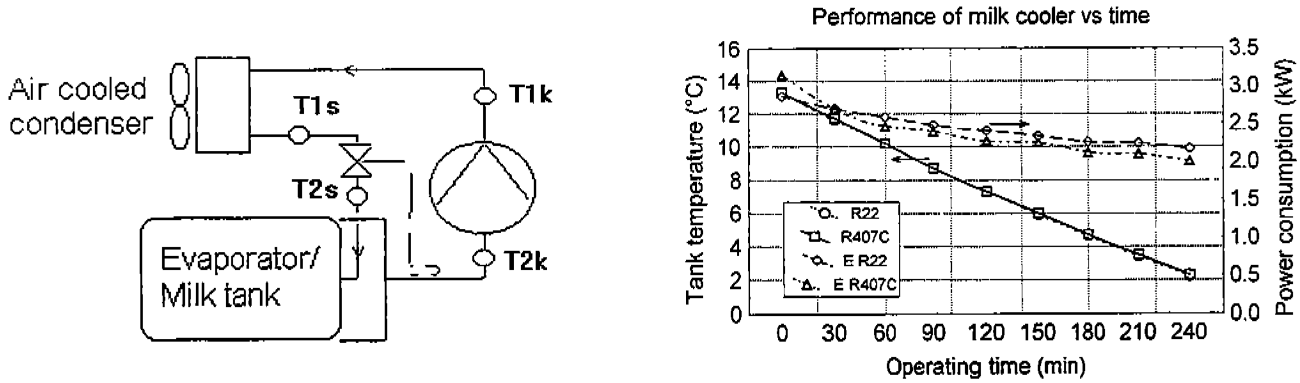

Fig 1-2. *One of three different milk coolers tested for the Swedish Dairy Association*

The measurements are done under transient conditions during the cooling of a batch of milk in the tank. This was done with constant air temperature for the condenser cooling. The electricity to the compressor was measured along with the temperature in the milk tank. Figure 2 shows the result from a comparison of R22 and R407C. The amount of milk (in this case water) is the same in the two tests.

Both R22 and R407C gives the same milk temperature versus time plot. This means that the cooling capacity is the same over the entire operating range. R407C gives surprisingly slightly lower power consumption i.e. better cooling COP. It is well worth noting that these tests not requires any knowledge of refrigeration engineering, "glide-refrigerants" etc. The refrigerant giving the lowest power consumption is the best in this unit, at least for the given air cooling temperature.

This type of tests, supplemented with analysis of the composition of the refrigerant, is used for evaluation throughout the entire milk-cooler test program. It is efficient, reliable and simple if the amount of milk in the tank in known. The equipment used is a simple data logger with six thermocouples and power measurement transducers.

Rigorous field tests

These tests are usually carried out under "laboratory-like" conditions. The difference is that it is difficult to **choose** an operating point. The test conditions are usually dictated by the cooling load and available cooling fluid temperatures.. The equipment used for these tests is called "The Refrigeration Analyzer" and it is developed by the Swedish company ETM. The idea with this device is to measure condenser and evaporator pressure along with a limited number of temperatures in the cycle. The electric power to the compressor is measured and gives the basis for a heat balance over the compressor. The heat loss from the compressor is small and well known. The refrigerant flow may thus be estimated (thermophysical properties for the common refrigerants are handled by the software). The method is convenient for field use but a full analysis, as descibed in next section, requires a heat balance over the evaporator or condenser. Many large chillers are equipped with flow meters in the secondary circuits (not always calibrated though). This means that a heat balance over the evaporator or condenser is possible.

One efficient way to evaluate refrigeration plants is to use the *Total Carnot Efficiency Concept* /2/. The efficiency of refrigerators or heat pumps is often compared to the ideal efficiency according to Carnot. The basis for this analysis have traditionally been the condensation and evaporation temperature. The drawback of this concept is that inefficiencies due to temperature differences in the heat exchangers are excluded from the analysis. This might be avoided if the mean thermodynamic temperature for the heat source and sink is used as the basis for the Carnot efficiency. If the two methods are combined an understanding of the **distribution** of the losses may be found.

There are often several reasons for different efficiencies in a heat exchanger in a refrigeration system. The heat transfer coefficients on both the refrigerant and the secondary side are important but practical problems such as maldistribution of the refrigerant often leads to large deviations. The method presented here suggests a general approach to this problem.

The idea is to evaluate the inefficiencies of the heat exchangers by a comparison of the classical definition of the Carnot efficiency with a definition based on the mean thermodynamic heat source and sink temperatures. Evaporation and condensation temperatures "for glide" refrigerants are defined according to AREP /4,5/.

$$COP_{2C} = \frac{T_{2evap}}{T_{1cond} - T_{2evap}} \quad \text{(eq 1)} \quad \text{or} \quad COP_{2Ctot} = \frac{T_{2mtdt}}{T_{1mtdt} - T_{2mtdt}} \quad \text{(eq 2)}$$

where:

COP_{2C} = Efficiency according to Carnot for the refrigerant cycle

COP_{2Ctot} = Efficiency according to Carnot for the whole system

T_{2evap} = Evaporation temp., $(T_{in} + T'')/2$, T_{1cond} = Condensing temp., $(T' + T'')/2$

T_{2mtdt} = Mean thermodynamic temperature for the heat source (brine)

T_{1mtdt} = Mean thermodynamic temperature for heat sink (water)

The Carnot efficiency is obtained by comparing the real measured COP_2 with the above two definitions:

$$\eta_C = \frac{COP_2}{COP_{2C}} \quad \text{(eq 3)} \quad \text{and} \quad \eta_{Ctot} = \frac{COP_2}{COP_{2Ctot}} \quad \text{(eq 4)}$$

The losses in one heat exchanger may be studied if the two definitions (eq. 1 and 2) *are combined.* This means that an analysis of the influence from the condenser is studied with a definition based on the evaporation temperature, T_{2evap}, and the mean thermodynamic temperature for the condenser coolant, T_{1mtdt}. This is represented in figures 1 and 2 for a small heat pump with plate type heat exchangers.

"Tot" means total Carnot efficiency, "Cond" means analysis of the condenser, "Evap" analysis of the evaporator and "Cyc", finally, means a comparison with the Carnot efficiency based on evaporation and condensation temperatures calculated according to AREP /4,5/.

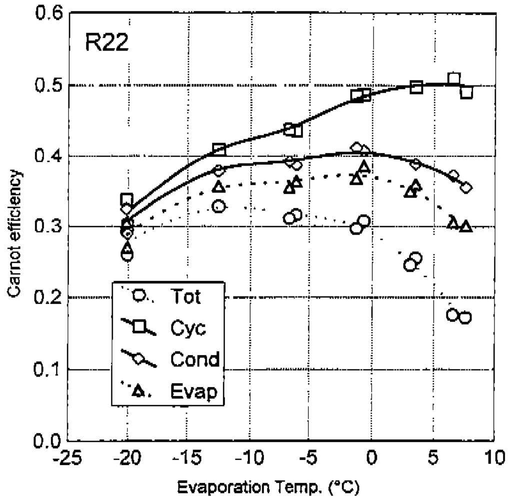

Fig 3. *Comparison of different Carnot efficiencies for R22 in a 10 kW water to water heat pump (Condensation temperature = 50 °C). Measured values (Hammarberg, Nyman, 1994 /1/)*

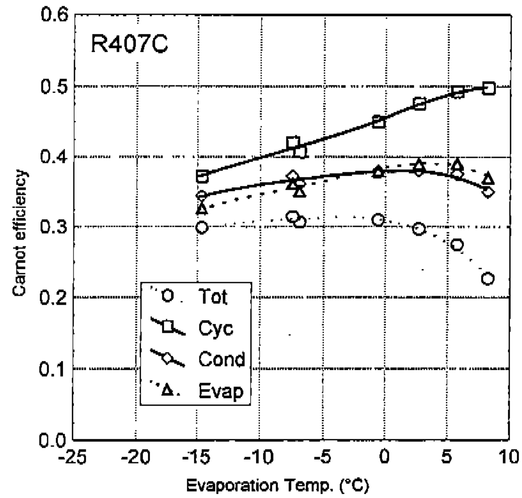

Fig 4. *Comparison of different Carnot efficiencies for R407C in the same heat pump (Condensation temperature = 50 °C). Measured values (Hammarberg, Nyman, 1994 /1/)*

The lowest values in figures 3 and 4 (Dotted /Rings) gives the curve for the total Carnot efficiency where inefficiencies in **both** evaporator and condenser have been taken into account. The highest value (Solid/ squares) is the Carnot efficiency based on the evaporation and condensation temperature. The intermediate curves represent the losses in the evaporator **or** the condenser.

The evaporator seems, in this case, to be more efficient for R407C, especially at high evaporation temperatures. The loss caused by the temperature differences in the evaporator is always larger than the condenser loss for R22. Experience from several systems converted to R407C (from R22) shows that the heat transfer problems, if any, occurs in the condenser. This is only slightly indicated in the above tests. This shows that a well designed system may operate with R407C as well as R22 with no significant loss in efficiency.

The method have been used to evaluate a typical shell and tube chiller converted from R22 to R407C. This type of heat exchangers are not ideally suited for a zeotropic refrigerant such as R407C. The evaporation takes place inside tubes but the condensation is on the outside. This leads to a well-known phenomena for zeotropic refrigerants: an extra mass transfer resistance due to the fractionation of the refrigerant during the condensation. The result is an increased condenser pressure and thus lower efficiency.

Refrigerant	"Cyc"	"Evap"	"Cond"	"Tot"
R22 (before)	0.53	0.42	0.38	0.25
R407C (after)	0.54	0.46	0.32	0.26

Table 2. *Result from total Carnot efficiency analysis of chiller with typical shell and tube heat exchangers*

It is apparent that the *distribution* of the losses is different for R407C. The heat transfer in the condenser is much poorer. A more detailed study showed an increase in the logarithmic temperature difference over the condenser with ca. 7K (approximately the glide). It is interesting to note the increase in efficiency of the evaporator that compensates for this loss.

It is in this case partly caused by an adjustment of the expansion valve at the time for the retrofit. This means that the efficacy for R22 **could** have been better.

A part of this study was to investigate the magnitude of expected changes in the composition of the circulating refrigerant. These tests showed that more advanced software (such as Refprop from NIST) was needed to carefully evaluate the tests. Gas samples was taken from the top of the condenser and from the liquid line before the expansion valve. The composition for a typical operating point was:

	R32	R125	R134a
Initial composition	0.23	0.25	0.52
Top of condenser	0.45	0.32	0.23
Liquid line	0.20	0.21	0.59

Table 3. *Composition shifts in the evaluated chiller (mass fraction)*

The composition of the circulated refrigerant is, as expected, changed towards more R134a and less R32. This means, in this case, a small loss in capacity.

Laboratory drop-in tests

The laboratory drop-in are characterized by the freedom to choose operating points. Laboratory drop-in tests are evaluated with mass flow measurements on the refrigerant side and/or heat balances over the evaporator and condenser. This lead to better accuracy and the possibility to evaluate compressor efficiencies and the heat exchangers thoroughly.

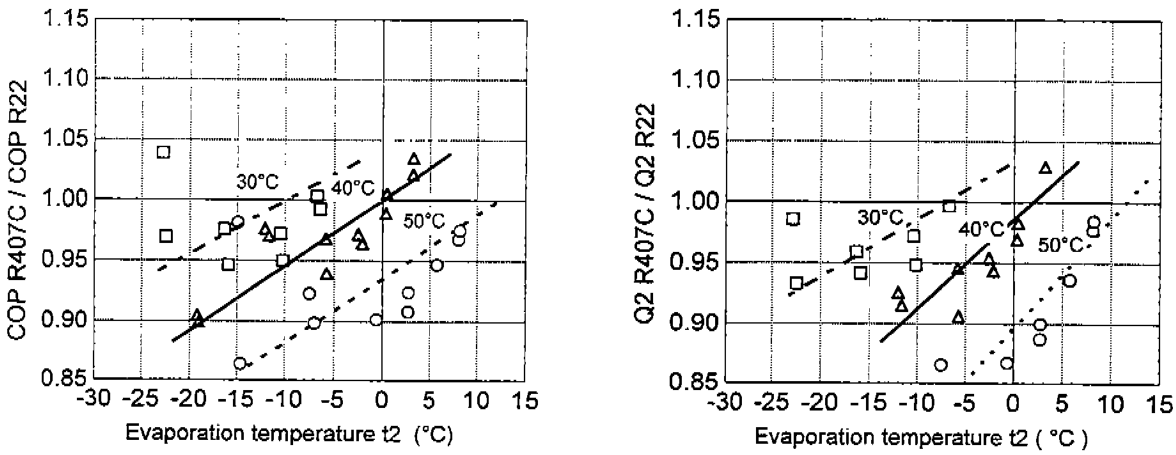

Fig 5-6. *Results from laboratory evaluation of R22 and R407C. Relative COP and cooling capacity versus R22. Measurements by Hammarberg&Nyman /2/.*

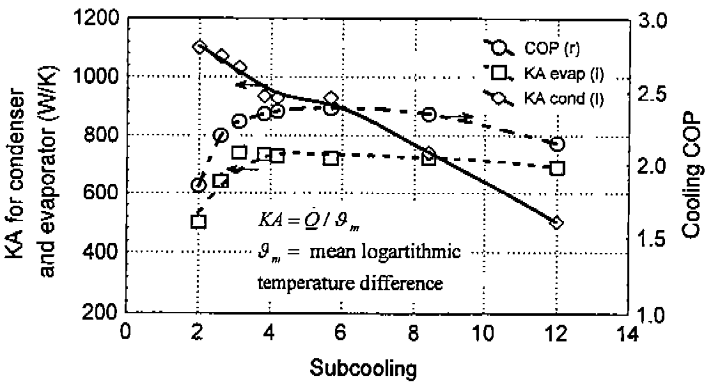

Fig 7. *Optimization of the refrigerant charge with R407C for the unit in figure 5-6 /1/. The charge is here given as the resulting subcooling (usually used as a measure when filling)*

Figure 7 stresses the importance of a correctly charged plant. The figure shows that lack of refrigerant is obvious for low charges. Maximum COP is reached at a subcooling of 6K. More refrigerant means that precious heat transfer surface in the condenser is used as subcooler instead of condenser! This is clearly seen for the "KA-curve" for the condenser.

Full laboratory evaluation

The most important difference from the laboratory drop-in tests is the time spent for measurements. The tests reported in figure 8-9 is for a scroll compressor equipped with a liquid injection port. This port has, in one of three operating modes, been used as an economizer port. The other two modes are one stage with and without suction gas heat exchanger. The refrigerants tested here are the two proposed alternatives for R502: R404A and R407A. The tests have in this case involved a more detailed analysis of the heat transfer on the refrigerant side. The so called *Wilson-plot method* have for example been used to evaluate the heat transfer on the refrigerant side (this work is still in progress).

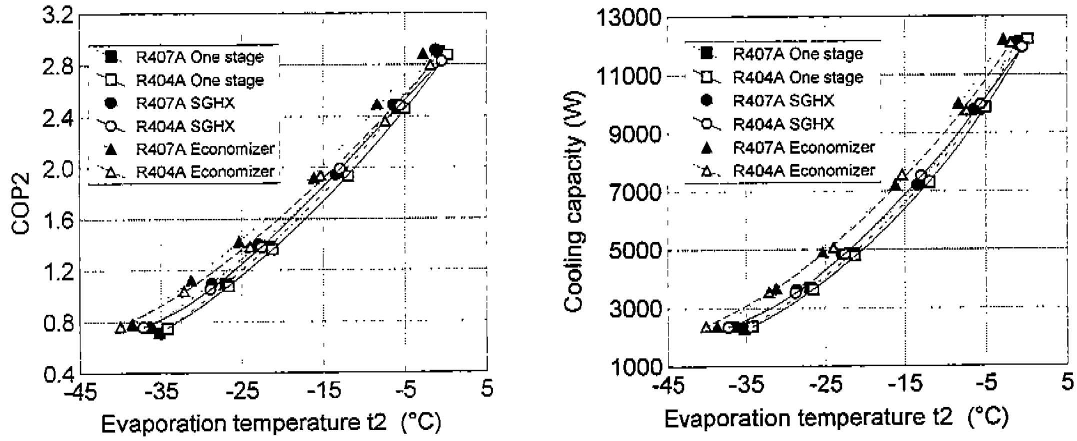

Fig 8-9. *Results from laboratory evaluation of R404A and R407A. Measurements by Mårtensson & Skoglund, 1996 /6/. Filled symbols- R407A, Open symbols - R404A.*

It is difficult to Separate the curves in figure 8-9, at least in the scale used in this paper. This means that the difference in COP and cooling capacity is fairly small. The performance for the economizer cycle is the best followed by the suction gas heat exchanger cycle (as expected).

These tests show, again, that a well designed system, in this case for R404A, may operate efficiently with a zeotropic refrigerant (R407A). Counterflow heat exchanger and a minimum of refrigerant charge seems to be the key. Tests of the circulating refrigerant composition showed that the deviation from the initial filling was undetectable for both refrigerants. The flow in the economizer port was restricted in these tests. Full benefit of the economizer cycle was therefore not possible. The results so far are promising and further work will show the full potential of this cycle.

Literature

1. **Hammarberg, G., Nyman, H.,** 1994, "R407C a zeotrope as an alternative to R22", Master of Science thesis, KTH, Energy Technology, dept of Applied Thermodynamics and Refrigeration, Stockholm
2. **Lundqvist, P.,** 1995, "Analysis of plate type heat exchangers with zeotropic refrigerant blends", Proceeding from ASME Winter Meeting, San Francisco, AES-Vol. 34, pp 37-58
3. **Lundqvist, P.,** 1994, "Some reflections on R502 substitutes.....", presentation at " Nordic Seminar on Alternative Refrigerants", OSLO, 27 September, 1994
4. **Godwin, D., Menzer, M.,** 1993, "Results of Compressor Calorimeter Tests in ARI's R-22 Alternative Refrigerants Evaluation Program", ASHRAE/NIST Refrigerant conference, Gaitersburg
5. **Morikawa, Y.,** 1993, "R-22, R-502 Alternative Refrigerant Evaluation Program in Japanese Industry", ASHRAE/NIST Refrigerant conference, Gaitersburg
6. **Mårtensson, M., Skoglund, E.,** 1996, "Cycle performance test of R404A and R407A using a scroll compressor", Master of science thesis, KTH, Energy Technology, dept of Applied Thermodynamics and Refrigeration, Stockholm