EXPERIMENTAL EVALUATION OF SOME PROPOSED R22 ALTERNATIVES IN CHILLERS AND UNITARY A/C EQUIPMENT

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ABSTRACT

Tests were conducted on two instrumented air-cooled water chillers (15 kW and 30 kW cooling capacity). Refrigerants evaluated included R22 as the baseline fluid, R407C and R410B. When testing with R407C, the retrofit procedure did not involve any major hardware change. The results with the first chiller equipped with a counter-current plate evaporator showed that both capacity and EER are maintained within ± 5%. The second chiller, which was not equipped with a counter-current evaporator showed poorer results. Capacity and EER were reduced by about 13% and 10%, respectively, as overall heat transfer coefficients were degraded by as much as 30% in the evaporator and about 10% in the condenser.

With R410B, two different compressors with reduced displacement were used in the second chiller in order to normalise the cooling capacity to R22 levels and the thermostatic expansion valve was modified. No modification was brought to the heat exchangers. With a properly sized compressor, capacity was maintained at R22 levels, whilst EER was reduced by about 10%. Despite increased compressor isentropic efficiencies, a significant performance degradation was noticed for R410B at higher ambient temperatures, which limits its application when using air as the condensing medium.

Cooling and heating tests conducted on a reversible room air conditioner (6 kW cooling capacity) with R22 and R407C and two compressors of equal size and of the same type have shown that capacity and EER with R407C are maintained, with only a marginal degradation in capacity (-4%) and EER (-2%); in the heating mode without frost formation, capacity and efficiency were maintained with R407C (± 1 %, and a maximum 5% degradation in efficiency); under frosting conditions heating duties were improved for R407C (+18%).

INTRODUCTION

European and world-wide legislation regulating the use of hydrochlorofluorocarbons (HCFCs) have lead major equipment manufacturers to evaluate alternatives for these refrigerants, and particularly for R22.

HFC long-term substitutes for R22 include R134a, R410A and R407C. Although much work has been done with regard to these alternative refrigerants (Ref. 1 to 8), considerable research efforts are further required to test and tune the new equipment generation within the time schedule imposed by the regulations. These efforts are essentially based on testing of equipment and refrigerants.

In order to compare the performance of alternative refrigerants, an important test program was launched at CETIAT within the "Pôle Fluides Frigorigènes" French research organisation framework. This research is currently being conducted with the support of a governmental environmental institution (ADEME), equipment manufacturers, refrigerant producers and a major utility company (Electricité de France). The objectives of the ongoing test program at CETIAT include the study of the influence of operating conditions on the change of performance relative to the base fluid (R22), of refrigerant charge and leakage on the unit performance.

The final objective of the research programme is to formulate recommendations for the use of alternative refrigerants, and the sizing and optimisation of new equipment. Eventually, these recommendations will be implemented in a real system which will be tested. This two-year programme covers the period 1995-96. This paper describes the results obtained from the "drop-in" testing of R407C in a 15 kW water chiller and tests conducted with R407C and R410B[1] in a 30 kW reversible water chiller, with modified compressor and thermostatic expansion valve. Tests conducted on a 6 kW room air conditioner (reversible split-system) using R407C are also described. For both chillers and the AC unit, tests were first run using R22 to provide reference values.

EQUIPMENT DESCRIPTION

Water Chiller N°1:

The 15 kW chiller operates in the cooling mode only, and is composed of a single circuit comprising a counter-current plate evaporator, a scroll compressor, a thermostatic expansion valve and a 2-row plate-fin-and-tube air-cooled condenser.

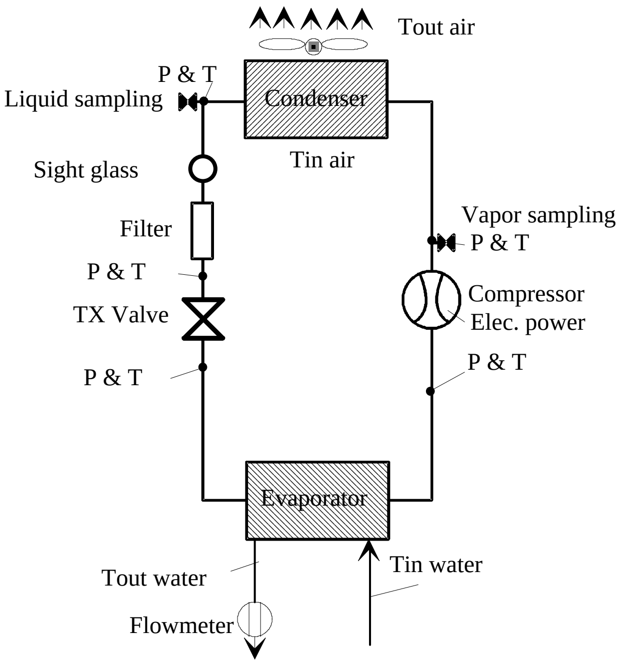

Figure 1: Chiller N°1 process flow diagram and instrumentation

Water Chiller N°2:

The reversible 30 kW chiller is composed of a reciprocating compressor and two parallel refrigerating circuits, each one comprising a spiral plate evaporator (with one water pass and

[1] When the tests were launched, R410B was still an option for some refrigerant manufacturers

two refrigerant passes), a thermostatic expansion valve and a four-circuit air-cooled spine-fin condenser.

The two chillers were fully instrumented with Pt-temperature sensors, pressure transducers and watt-meter on the refrigerant side, and a flowmeter and temperature sensors on the water side. Valves were also available for sampling purposes or to generate refrigerant leaks from the system. Figure 1 shows a schematic of Chiller N°1 flow diagram and instrumentation. The units were then installed in a psychometric test chamber at CETIAT. Inlet air conditions to the condenser were maintained by external air treatment loops controlling temperature and humidity. Data was gathered via a PC and cooling capacity was derived from the water side measurements. The uncertainty on the cooling duty is estimated to be less than 3%.

Room Air Conditioner:

The reversible 6 kW A/C split system is composed of:

- a reciprocating compressor;
- capillary tubes for expansion and a four-way reversing valve;
- an indoor unit finned-coil comprising 4 rows of grooved tubes; tube arrangement is cross-flow, but circuiting approximates it to counter-flow; in addition, intermediate tube branching attempts to keep vapour and liquid phases together by increasing the allowable cross sectional flow area. In reverse mode (condensation), tube joining plays the same role by reducing flow area;
- an outdoor unit finned-coil comprising 3 rows of smooth tubes; tube arrangement is cross-flow, but circuiting approximates it to counter-flow; intermediate tube is provided but branching is not as elaborate as in the indoor unit.

The AC unit was tested in a double-calorimeter room. Instrumentation allowed measurement of the refrigerant flow rate after the compressor discharge and of pressures and temperatures at all cycle important points. All data was gathered via a computerised system. Continuous monitoring of the main variables (temperatures, humidity, duties, etc.) was made possible by a graphical interface.

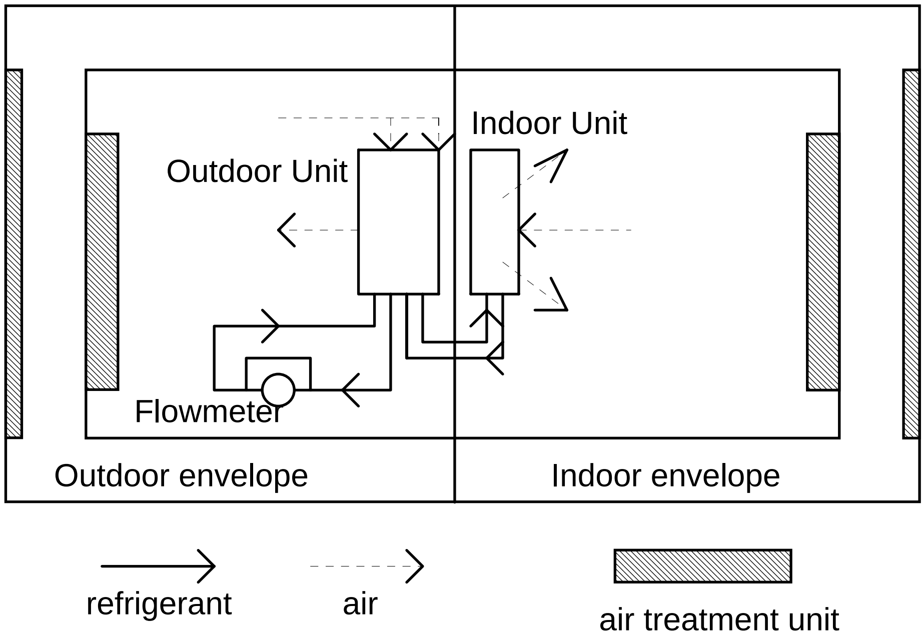

Figure 2: Schematic view of the double-calorimeter room

TEST PROCEDURE

Tests were first performed on Chiller N°1 with R22 to obtain baseline results for comparison. For a given set of operating conditions, an optimum refrigerant charge was determined. Tests were then run for a number of standard operating conditions.

Following the R22 tests, the chiller was retrofitted (change to ester oil and change of filter-dryer) and tests with R407C were performed at similar operating conditions. Simulations of liquid and vapour leaks were made by withdrawing refrigerant from the sampling valves (see Figure 1).

A different procedure was followed for Chiller N°2. Following refrigerant charge determination, tests for R22 were run in both cooling and heating modes using ester oil. The unit was then modified for the R410B tests. A smaller-size compressor, with a 37% reduction in swept volume, was used. A change of expansion valve was also required, but no other device was available. It was therefore decided to keep the same valve, but to change the fluid contained in the sensing bulb. The existing fluid was evacuated from the bulb, and R410B was used instead. The unit behaved perfectly well thereafter. After a few runs, a new compressor with an increased swept volume was installed and tests were run for R410B. Using the original R22 compressor, tests were also performed using R407C.

The reversible room air conditioner was tested in a double-calorimeter room (see Figure 2). Each room is equipped with its own air treatment unit, where the psychometric conditions could be monitored and automatically controlled. Additional air treatment units situated in the annular spaces (or "envelopes") between the test rooms and the surroundings maintain minimal temperature gradients. Heat losses to/from the test rooms were calculated.

The AC unit was first tested using R22, with the amount of refrigerant charge specified by the manufacturer. The system was emptied and cleaned. The compressor was then substituted for by an equal size ester-oil precharged compressor for the R407C tests. Three test runs were carried out for refrigerant charge determination. The optimum charge was found to be 10% higher than for R22.

Two methods could be used to measure the indoor coil duty:

- direct method: by performing an energy balance around the indoor coil, using the measured refrigerant flow rate and the values of refrigerant enthalpy at inlet and outlet of the coil;
- indirect method (or 'calorimetric): by performing an overall energy balance around the indoor testing room; this balance includes all input/output terms, i.e. the coil duty, the energy required to maintain environmental conditions (air treatment unit), heat losses/gains from the test room, plus any auxiliary equipment such as lighting, etc.).

For most of the tests, the direct method was used. The two methods were compared for runs at nominal conditions, with discrepancies smaller than 5%.

Testing conditions

Performance tests were run in representative conditions of the equipment operating range. All these operating conditions are shown in Table 1 for the chillers and Tables 2 and 3 for the reversible A/C unit.

The nominal water flow rate for the chillers was determined during Test n°1, by maintaining the inlet and outlet water temperatures at the required levels.

Comparison of performance

The Energetic Efficiency Ratio (EER) was calculated as the ratio of the useful duty (cooling or heating) over the total electric power consumption.

In order to compare the results, deviations from the reference R22 (baseline) results were calculated as follows:

$$\Delta X_{fluid}(\%) = 100\frac{X_{Fluid} - X_{R22}}{X_{R22}}$$

where X is either cooling/heating duty, EER, flow rate or overall heat transfer coefficient.

Table 1 : Cooling mode test conditions (Chillers)

Test N°	**T air inlet (°C)**	**T water (°C)**	**Water flow rate**
1	35	12 (inlet) to 7 (exit)	test finds nominal
2	25	12 (inlet)	nominal
3	25	20 (inlet)	nominal
4	42	20 (inlet)	nominal
5	40	10 (inlet)	nominal
6	35	20 (inlet) - 15 (exit)	
7	35	15 (inlet) - 7 (exit)	

Table 2 : Cooling mode test conditions (Room air conditioner)

Test	Indoor Unit TDB (°C) ++	Indoor Unit TWB (°C) *	Fan Speed	Outdoor Unit T (°C)
1	27	19	normal	35
2	27	19	reduced	35
3	21	15	normal	27
4	21	15	reduced	27
5	32	23	normal	43
6	32	23	reduced	43

++ air dry bulb temperature * air wet bulb temperature

Table 3 : Heating mode test conditions (Room air conditioner)

Test	Indoor Unit TDB (°C)	Indoor Unit TWB (°C)	Fan Speed	Outdoor Unit T(°C)
7	7	6	normal	20
8	24	18	normal	27
9	2	1	normal	20

TESTS RESULTS FOR CHILLER N°1 (R22 and R407C)

A refrigerant charge determination was first conducted. This test was performed by initially undercharging the system and then progressively adding charge in increments weighing 400 to 600 grams. Refrigerant was charged in the liquid phase. In the case of R407C, no intermediate venting was allowed to prevent change of overall refrigerant composition due to component separation. Rather flat EER vs. Charge curves were obtained, particularly for R22, with an optimum charge of around 6 kg. Similar results were obtained for R407C. Details can be found in Ref. 9.

In terms of performance (EER and cooling capacity), only small differences between R22 and R407C were observed. Figure 3 shows the relative performance (in terms of cooling capacity and EER for the two fluids) as function of the air inlet temperature to the condenser. The maximum deviation occurs when the air temperature is lowered to 25°C (tests 2 and 3). Compressor exit temperatures for R407C were found to be 10°C lower, while condensing pressures were 1 to 1.8 bars higher. Suction pressures remained the same. Overall, however, the results show that capacity and EER are maintained within ± 5% when using R407C.

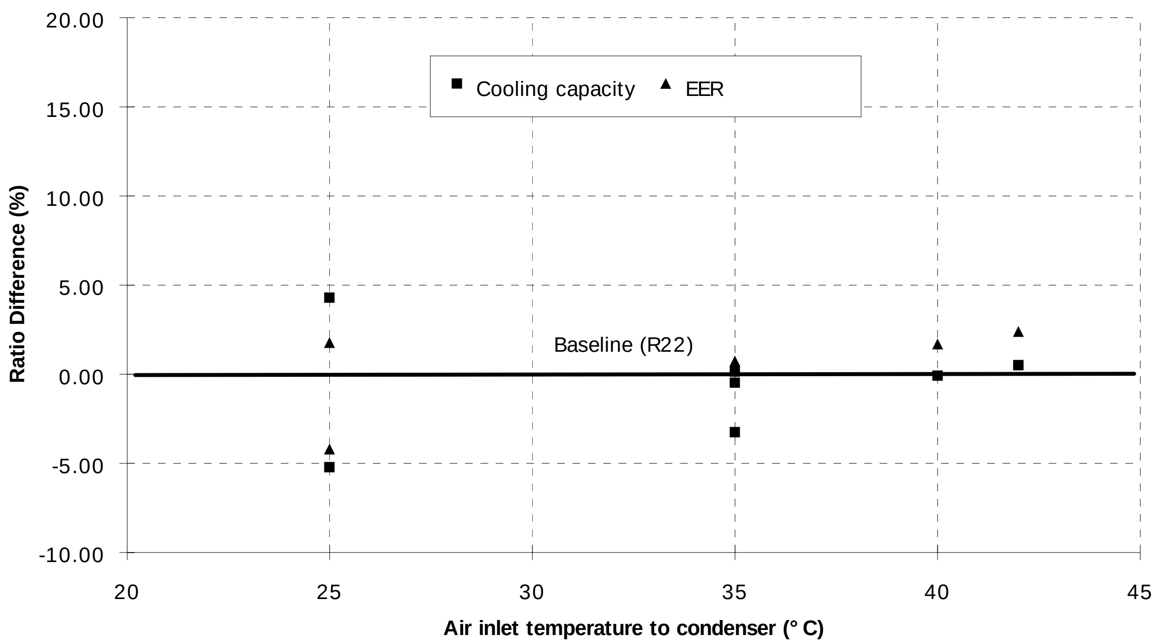

Figure 3: Performance comparison when using R22 and R407C (Chiller n°1)

TESTS RESULTS FOR CHILLER N°2

Preliminaries

Following runs with R22, a compressor with a 37% decrease in swept volume was installed for the R410B tests. A few runs were performed with this compressor, yielding a cooling duty decreased by 2 to 21% compared to that obtained with R22, with a difference in efficiency changing from +12 to -3% (see Ref. 9). A new compressor was then installed, with a 20.6% reduction in swept volume compared to the original R22 compressor and experiments were conducted with R410B.

Following these runs, the original R22 compressor was reinstalled and tests run for R407C.

Experiments with R410B

In order to compare the two fluids, the operating conditions specified in Table 1 were again selected, except for runs at air ambient temperatures of 42°C, which were not performed. A temperature of 40°C was chosen instead for R410B.

Operating pressures for R410B were higher than those obtained with R22, with suction pressures between 7.5 and 9 bars, and discharge pressures between 24 and 35.3 bars. Compared to R22, working pressures are therefore 50% higher for R410B. It is noteworthy that the compressor exit temperatures are decreased by 15 to 20°C compared to R22.

Figure 4 shows the ratio of cooling capacities obtained with the two fluids, for the conditions listed in Table 1. Under standard operating conditions (Test n°1), the cooling duty obtained using R410B is roughly equivalent to that obtained with R22 but the EER is decreased by about 10%. Throughout the chiller operating range, the differences in cooling duties between R410B and R22 range from +8 to -5% based on R22, whilst EER changes from -7 to -12%.

At an ambient temperature of 40°C and a water outlet temperature of 15°C, the relative performance of R410B seems to be increasing and lies above that obtained at 7°C. This is erroneous and is due to the fact that this comparison was based on an ambient temperature of 42°C for R22 and 40°C for R410B. This test was therefore biased and in favour of R410B. Figure 4 shows (in dotted line) the extrapolated relative performance which would have been found, should this test have been conducted at 42°C.

Figure 4 also clearly shows that in terms of thermodynamic efficiency, the use of R410B is more favourable at lower ambient temperatures. This is due to its thermodynamic properties, and mainly to its lower critical temperature.

Experiments with R407C

Following refrigerant charge determination, tests with R407C and the original R22 compressor were performed. Depending on water temperature at evaporator outlet, evaporator duty was decreased from 12 to 16% compared to R22 and EER losses ranged from 8 to 11%. These tests showed surprisingly degraded figures for the second chiller compared to the first one. Figure 5 shows the differences obtained. In contrast with R410B, the use of R407C does not lead to aggravated performance degradation at higher ambient temperatures.

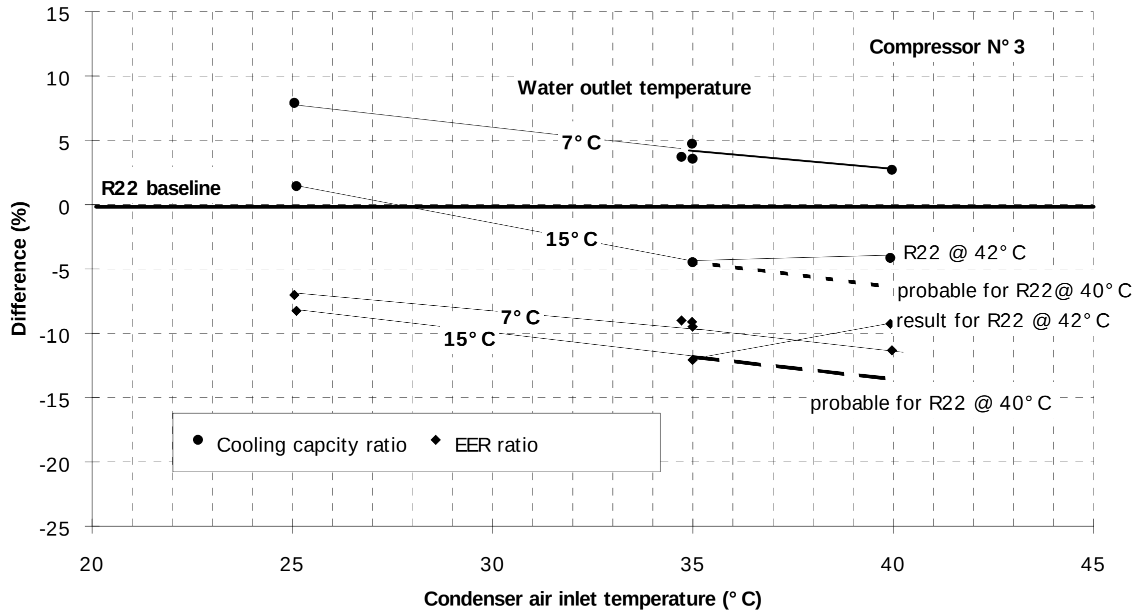

Figure 4: Performance comparison between R22 and R410B in Chiller n°2

***Note**: the EER ratio à 15°C seems to improve, and the EER line crosses the 7°C line for air temperatures greater than 35C. This is erroneous because the ratio is based on a 40°C ambient air for R410B and a 42°C ambient air for R22, thus favouring R410B. If the test were run at 42°C for R410B, the 7°C line would thus fall below the 15°C for high ambient temperatures. Probable lines are shown broken.*

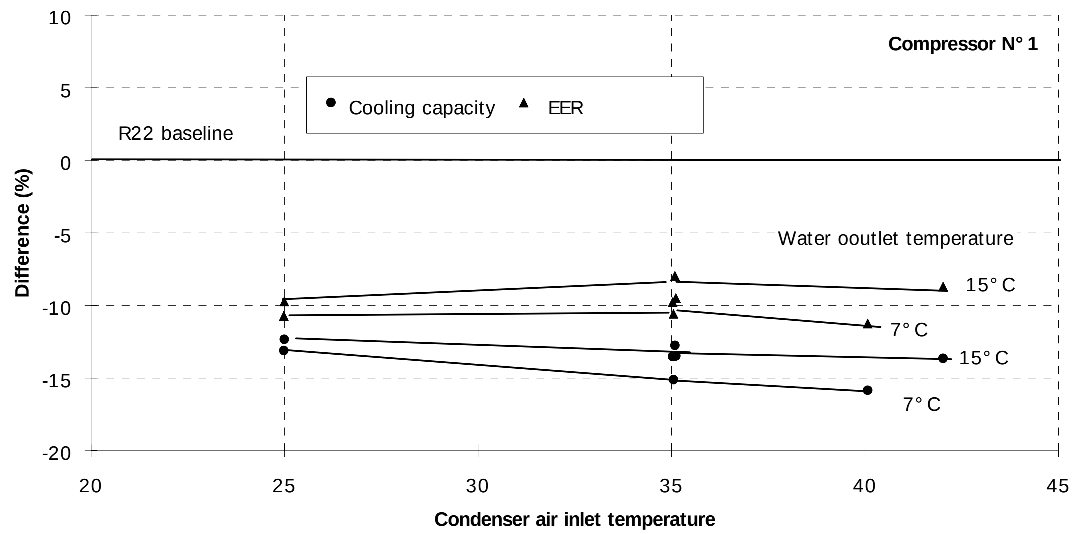

Figure 5: Performance comparison between R22 and R407C in Chiller n°2

TESTS RESULTS FOR THE ROOM AIR CONDITIONER

Cooling mode

Figure 6 shows the ratio of the capacity and EER obtained with R22 and R407C in the cooling mode, with R22 as the base fluid and as function of the indoor air inlet temperature. Since differences are very small, Figure 6 does not discriminate between different air temperatures to the outdoor coil. Although R407C performs consistently worse than R22, there are very small differences between the two fluids (2 to 4%). At the nominal conditions (Test n°1 in Table 2), the two fluids perform much the same way. The worst performance degradation is obtained when the temperature lift is greatest (tests 5 and 6, with 2% loss in EER and 4% loss in capacity). Compressor discharge pressures are higher for R407C (+2 to 2.5 bar) for all tests. Suction pressures are approximately the same. In contrast with tests conducted on chillers, the discharge temperature for R407C and R22 are similar, within a few degrees.

Heating mode

Under conditions of Tests n°7 and 8 (see Table 3), the room air conditioner operates at steady state. For test n°9, the operation is cyclic, with frosting and defrosting periods. For this unsteady-state regime, the indoor coil duty is measured using the indirect (or calorimetric) method. This method involves the calculations of all energy terms required to maintain the room at the conditions shown in Table 3. In order to obtain accurate results, the calculations are carried out over three completed stable cycles (frosting+defrosting).

As shown in Table 4, R407C performance is essentially identical to that of R22 (± 1 % for heating duty and a maximum 5 % EER degradation). For cyclic operation, R407C performs even better (+18% for capacity and +9% for EER). This item will be discussed later.

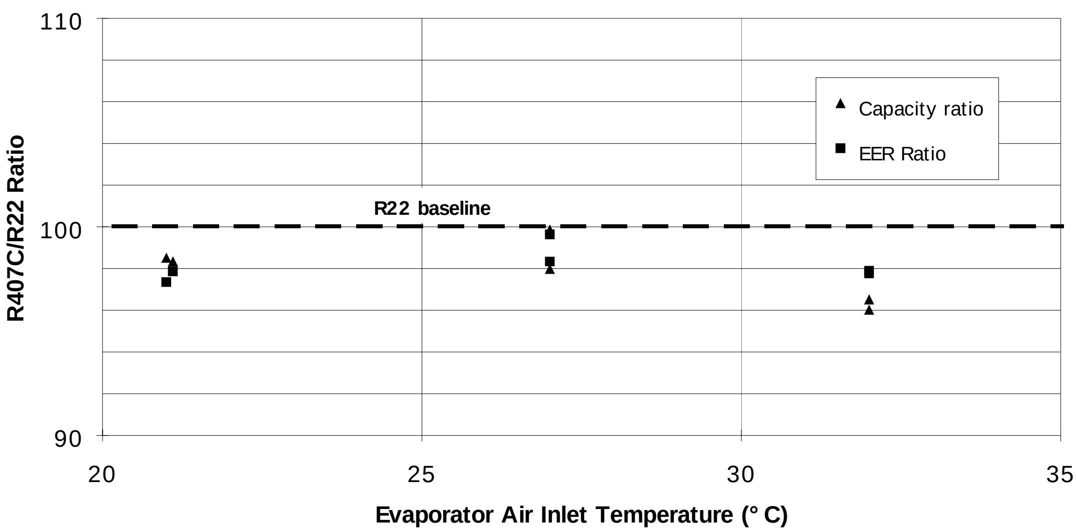

Figure 6: Performance comparison between R22 and R407C
in the room air conditioner (cooling mode)

Table 4: Performance comparison between R22 and R407C
in the room air conditioner (heating mode)

Test		**7**	**8**	**9**
Capacity R407C/R22	%	101	99	118
EER R407C/ EER R22	%	101	95	109

ANALYSIS OF RESULTS

Water Chillers

The performance degradation resulting from the use of R407C in chiller n°2 contrasted with results obtained with the first chiller and justified the need for a further analysis. For chiller n°2, ratios of refrigerant flow rates and overall heat transfer coefficients (HTCs) in the two-phase regions were calculated based on R22.

Heat transfer coefficients and refrigerant flow rates

Refrigerant flow rates were calculated by performing an energy balance around the evaporator. They were then compared on a relative basis (reference fluid = R22)

Comparison of the heat transfer coefficients required more complex computations. "UA" is the product of the overall heat transfer coefficient (which we will refer to as the "HTC") and the heat transfer area. The UA product is defined as:

$$U.A = \frac{Duty}{LMTD}$$

where LMTD is the logarithmic mean temperature difference for the heat exchanger (or part of it) under consideration.

In order to calculate logarithmic mean temperature differences, counterflow was assumed in the heat exchangers. Although this is an erroneous assumption, it is thought that it will not yield serious errors because computed quantities are calculated and compared on a relative basis.

For a given heat exchanger, the area is constant, and the ratio of the "UA" products will yield the ratio of the "U" for the two fluids under comparison.

For the evaporator, the "UA" products were determined for the refrigerants by assuming a negligible heat transfer duty in the refrigerant superheating zone.

In the condenser, refrigerant inlet and outlet conditions were measured. From the previously determined values of refrigerant flow rates, the ratio of desuperheating duty to overall duty could be calculated. By assuming the subcooling duty to be negligible, and based on the measured inlet and outlet air temperatures, the air temperature at exit of the refrigerant condensing zone could be determined. Refrigerant pressure at inlet of the condensing zone was set equal to the measured pressure at condenser inlet. Hence, a saturation temperature (dew point for mixtures) could be calculated at this point. Refrigerant pressure at exit of the condensing zone was set equal to the measured condenser exit pressure and temperature could be calculated as the saturation temperature (bubble temperature for mixtures) at that point. With the condensing duty and the air and refrigerant temperatures, overall condensing "UA" products could be calculated. "U" ratios based on the R22 values were then computed for the condensing zone.

It is worth noting that the water flow rate was not constant during the experiments with all three fluids. While it was essentially the same for R22 and R410B, a smaller flow rate was used for R407C (approximately -13% compared to that with R22). Considering that the overall heat transfer coefficient depends to some extent on water flow rate, this reduction will have to be taken into consideration.

Figure 7 shows the differences in overall HTCs in the evaporator and condenser two-phase regions for R407C, as compared to R22. The significant heat transfer degradation in the evaporator (with losses ranging from 27 to 32%) results in evaporation at a lower pressure level, thus decreasing the amount of refrigerant pumped through the compressor. Refrigerant flow rates were therefore reduced by 8 to 12% compared to R22, resulting in correspondingly smaller evaporator duties, as shown by Figure 5.

Condensing HTCs are not degraded as much as in the evaporator, with losses between 4 and 15%. This is due to the fact that the air side heat transfer resistance is predominant in the condenser.

Figure 8 shows similar plots for R410B. Depending on water outlet temperature, refrigerant flow rate is increased between 0 and 20%. Improved evaporator performance correspond to flow rate increases larger than 17%. Evaporator overall HTC is degraded at small flow rates, with losses higher than 25%. Overall condensing HTCs dropped, with decreases between 4 and 15%. In contrast with evaporating HTC, the condensing HTC does not depend much on refrigerant flow rate, and practically equal values are obtained regardless of water outlet temperature at a given ambient temperature.

It is noteworthy that despite decreases in both evaporator and condenser performance, the unit EER "only" degraded by about 10% with regard to R22. The explanation can be found by calculating the compression isentropic efficiencies for both fluids. For R22, the calculations resulted in isentropic efficiencies ranging from 60 to 63%, whilst R410B lead to efficiencies ranging from 68 to 71%. This meant a relative increase of 10 to 17%. This largely made up for the losses experienced in the heat transfer coefficients, and limited the degradation in EER.

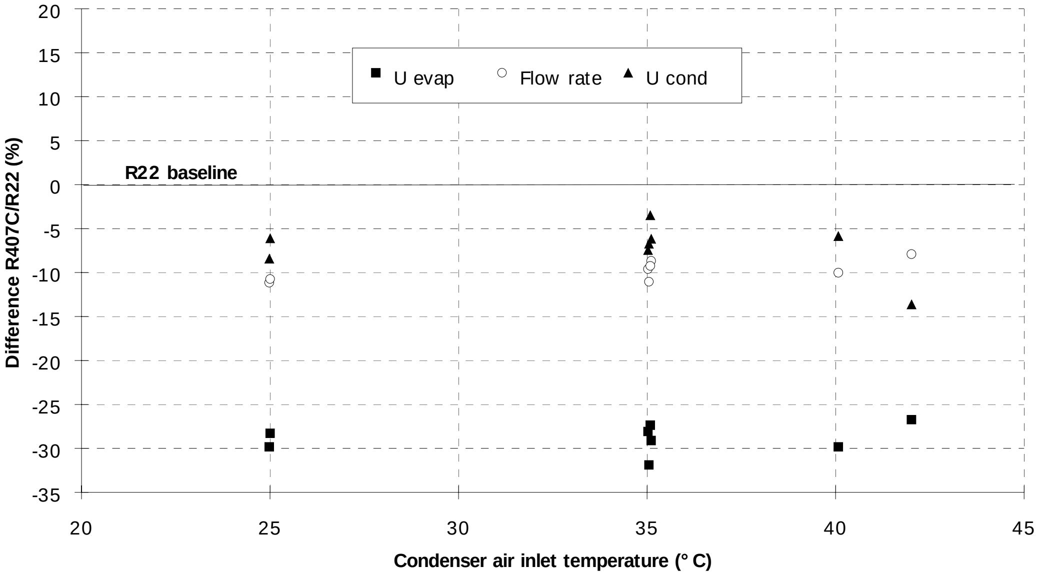

Figure 7: Differences in overall two-phase
heat transfer coefficients between R22 and R407C in Chiller n°2

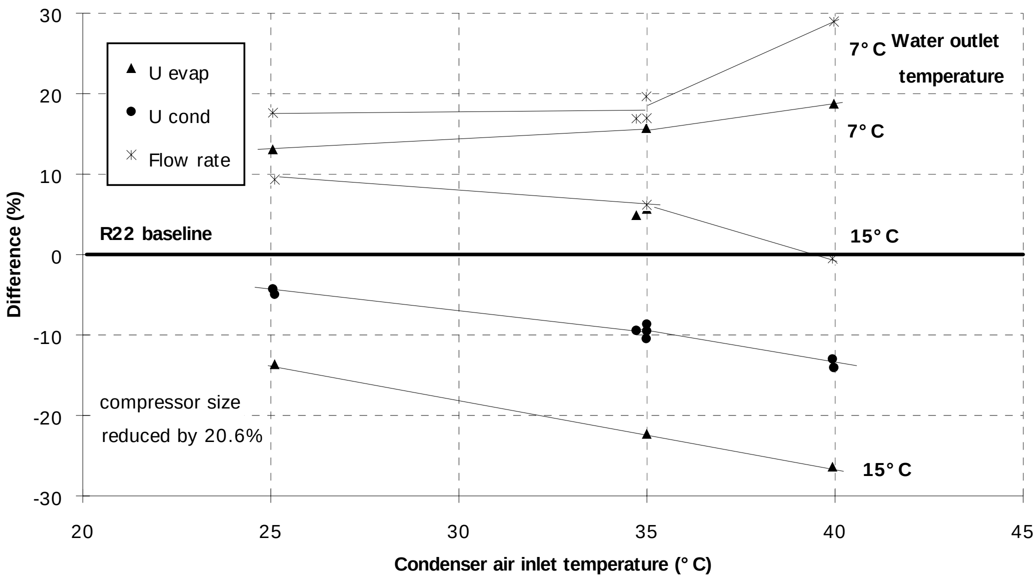

Figure 8: Differences in refrigerant flow rates and overall two-phase
heat transfer coefficients between R22 and R410B in Chiller n°2

Room Air Conditioner

Both capacity and efficiency were maintained with R407C for the room air conditioner. The governing phenomena are the heat transfer processes and fluid thermodynamic properties:

1. for both indoor and outdoor coils, the air side constitutes the dominant resistance to heat transfer; consequently, and if heat transfer coefficients were reduced for the R407C non-azeotropic mixture, the effects on overall heat transfer coefficients and therefore on unit overall performance would be moderated;

2. the indoor coil uses grooves tubes, with tube circuiting which, in the cooling mode, progressively increases the flow area. When refrigerant is evaporating, flow area increase causes the vapour velocity not to increase monotonously due to liquid evaporation; this maintains the vapour and liquid phases together and minimises slip. In the heating mode, flow is reversed, so that tube circuiting causes flow area to reduce, thereby increasing the vapour velocity as the refrigerant condenses; again, it minimises phase slip. In the case of mixtures, the refrigerant heat transfer coefficient is dependant to some extent on diffusion between phases. Phase slip can thus unbalance thermodynamic equilibrium because phases do not flow together, thereby causing a degradation in heat transfer coefficient. In addition, the grooved tubes have beneficial effects on heat transfer because of the increased turbulence and mixing. Both of these phenomena contribute to a better approach to thermodynamic equilibrium, thus enhancing heat transfer. These beneficial effects can be observed through the small temperature pinches measured in the evaporator (of the order of 2°C for R407C). The 'average' evaporating temperatures for R407C and R22 were observed to be practically the same, which yielded similar suction pressures, refrigerant flow rates and duties.

3. isentropic efficiency was better for R407C; for the six runs in the cooling mode, efficiency for R22 was around 66%, while it varied between 70 and 74% for R407C.

In heating mode with cyclic operation, capacity was increased by up to 18%. Under unsteady-state conditions, all energy terms required to maintain the indoor room test set conditions are calculated during three cycles. Unit heating capacity is then calculated as an integrated 'average' over the three-cycle period. Tests with R407C were carried out with only a change in compressor (with the same swept volume), so that the devices triggering the defrosting cycle or their set point were not altered at all. Longer outer unit frosting times were observed for R22 than for R407C (2hr 10min vs.1hr 25min). Typical unit cyclic operation is shown in Figures 9 (R407C) and 10 (R22). As it can be observed, the heating duty for R22 decreases exponentially with time, and heating cycle is maintained during more than 30 minutes at a low duty level. For R407C, the frosting cycle time is shorter and the indoor coil supplies a quasi-constant heating duty throughout the cycle. Then the control device shuts the cycle off and triggers defrosting. The heating duty is therefore integrated at a higher level than for R22, thus leading to better performance.

It is noteworthy that whilst allowing improved heating performance, the present setting of the control devices causes more frequent cycle reversals.

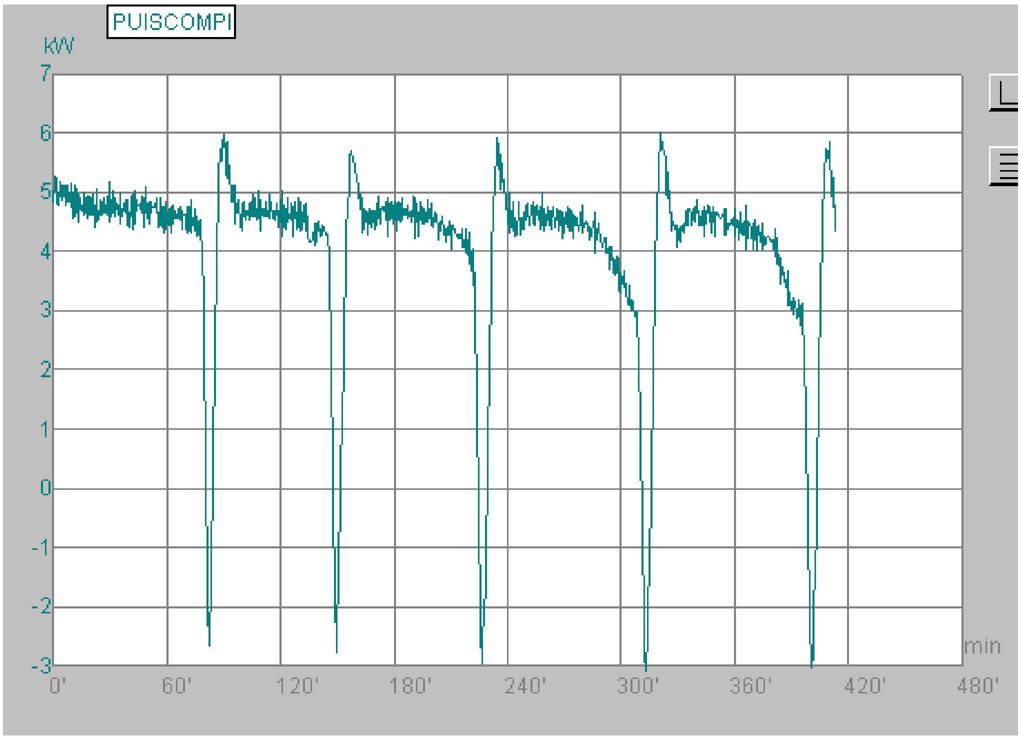

Figure 9: Variation of heating duty for R407C (cyclic operation with frosting/defrosting)

Figure 10: Variation of heating duty for R22 (cyclic operation with frosting/defrosting)

CONCLUSIONS

Cooling tests conducted on two different chillers using R22 and R407C without major hardware changes have shown the following:

- optimum refrigerant charge is the same for both fluids;
- capacity and efficiency can be maintained within ± 5 % when using R407C with a counter-flow evaporator;
- without counter-flow heat exchange, losses in both cooling capacity and efficiency were experienced; depending on water temperature at evaporator outlet, evaporator duty was decreased from 12 to 16% and EER losses ranged from 8 to 11%; overall evaporating HTCs were decreased by 30%, overall HTC in the condensing zone fell by about 10%.

Tests conducted on the second chiller with R22 and R410B, using a smaller-size compressor and filling the expansion valve sensing bulb with R410B have shown the following:

- with a compressor swept volume reduced by 20.6%, the cooling duty changes from +8 to -5%, with EER changing from -7 to -12%;
- a 10 to 17% improvement in isentropic efficiency was obtained with R410B, which largely contributed to offset the decrease in EER;
- no optimisation study was conducted to adapt the expansion valve to the new fluid; in order to fully optimise the chiller operation, such a study will be required;
- R410B yields smaller saturation temperature changes upon pressure drop; this can be exploited to design different heat exchanger circuitry, using higher mass fluxes and resulting in more efficient heat transfer;
- R410B yields better efficiencies at lower condensing temperatures.

Cooling and heating tests conducted on a reversible room air conditioner using R22 and R407C without major hardware changes (the two compressors used were of equal size and of the same type) have shown the following:

- required refrigerant charge for R407C is slightly higher than for R22 (about 10%);
- in the cooling mode, capacity and EER with R407C are practically maintained, with only a marginal degradation in capacity (4% at most) and EER (2%);
- in the heating mode with no cyclic operation, capacity and efficiency are maintained with R407C (± 1 %, and a maximum 5% degradation in efficiency); under cyclic conditions, frosting period is shorter for R407C and heating duties are improved (+18%).

These tests have shown that heat exchanger geometry and configuration are important when applying large glide refrigerant mixtures such as R407C. Proper design is therefore required to ensure that performance is maintained, particularly when the refrigerant heat transfer coefficient plays an important role in the overall heat transfer resistance, such as in chiller applications.

The high-pressure blend R410A/B presents new challenges in terms of equipment (e.g. compressors) but does hold promises such as reduced equipment size and improved efficiency. Its insignificant glide could prove to be an advantage in case of leaks from systems. However, its implementation does require proper equipment design.

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