

DEVELOPMENT AND EVALUATION OF A REVERSIBLE CO₂ RESIDENTIAL AIR CONDITIONING SYSTEM COMPARED TO A STATE-OF-THE ART R410A UNIT

ARNE JAKOBSEN¹, GEIR SKAUGEN¹, TORGEIR V. SKIPLE², PETTER NEKSÅ¹, TROND ANDRESEN³

¹SINTEF Energy Research, Trondheim - Norway

²Student at Norwegian University of Science and Technology – NTNU Trondheim – Norway

³Ph.D Student at NTNU Trondheim – Norway

ABSTRACT

A reversible CO₂ prototype RAC split unit system has been designed, built and installed in two calorimetric test chambers. The unit is based on the Lorentzen Cycle with a receiver and a suction line heat exchanger. Both the indoor and the outdoor heat exchangers have tube-in-fin design. The hermetic compressor has two compression stages and is inverter driven.

In heat pump mode the CO₂ prototype unit can either operate as a conventional one-stage circuit or as a two-stage system. In air-condition mode, there is an option for cooling of the compressor first stage discharge gas.

Experimental tests are under preparation. Realistic simulations have been carried out with the advanced in-house simulation program CSIM using input data for all components of the prototype. CSIM is a steady state circuit simulator (primarily) for trans-critical CO₂ cycles, which uses well calibrated models for all components.

A state of the art inverter driven R410A split unit has been tested experimentally as a baseline unit. Obtained results have been compared to simulation results for the CO₂ prototype.

Results indicate that the prototype CO₂ unit will perform almost equal as the R410A baseline unit in AC mode at 27.8°C and 35°C ambient temperature. In heat pump mode, the prototype CO₂ unit is predicted to perform around 30 to 40% better at –5°C and 5°C ambient temperature.

1 INTRODUCTION

The yearly world market for residential air condition units (RAC split type) is around 40 mill. units with an estimated value of 30-40 billion USD (Nowacki, 2002), and further market growth is expected. Hydro Fluor Carbons (HFCs) are the dominant working fluids in residential split units today. These are known to have relatively high global warming potentials and may also have other unforeseen long-term effects since they do not belong in the biosphere.

Carbon dioxide (CO₂) is a purely natural substance which is non flammable a non-toxic. CO₂ technology has in recent years become commercially available in heat pump water heaters and is expected to be introduced in several other appliances in the near future.

A concern against introducing CO₂ technology in RAC systems has been that system performance will not be competitive at air-condition operation (E.g. Kim et al 2003). It is well established that CO₂ performs equal or better running in heat pump mode (Aarlien, 2001), (Richter et al. 2000).

Extensive testing on mobile air conditioning (MAC) systems has been carried out over the last ten years. Results for equally enhanced CO₂ and R134a systems show that CO₂ systems perform better than R134a at almost all relevant conditions (Hafner et al. 2004). It is also shown that CO₂ systems have lower Life Cycle Climate Performance in all investigated climates.

While Aarlien (2001) worked on a rather old fashioned RAC system and Richter et al (2000) worked on an American system with large capacity and large heat exchangers, the present work focuses on a Japanese/ European split type units and the potential of using CO₂ in such systems.

A CO₂ prototype RAC system has been designed, built and installed in two calorimetric test chambers at the SINTEF laboratories. Experimental tests are under preparation. Realistic simulations have been carried out with the advanced in-house simulation program CSIM using

input data for all components of the prototype Simulation results are compared to measured performance of a state of the art RAC R410A split unit.

2 PERFORMANCE TESTS OF R410A BASELINE SYSTEM

2.1 The baseline system

A commercially available state of the art inverter driven RAC split unit with R410A as working fluid has been installed and tested in two calorimetric chambers. According to the data given by the manufacturer, the nominal capacities for the unit working at 230V and 50Hz are 3500W cooling capacity at 35°C ambient temperature and 3140W heating capacity at -5°C ambient temperature.

2.2 Test Facility

The test facility consisted of two insulated chambers where relative humidity and temperature could be maintained at specified levels. Thermal conductivity of the walls was 0.3 W/m²K and the volumes of the rooms were 18.6m³ and 32.7m³ for the outdoor and indoor unit respectively. A variable speed wind tunnel with glycol circuits connected to each room emulates the heat demand and cooling demands. Figure 1 shows as a schematic view of the test chambers.

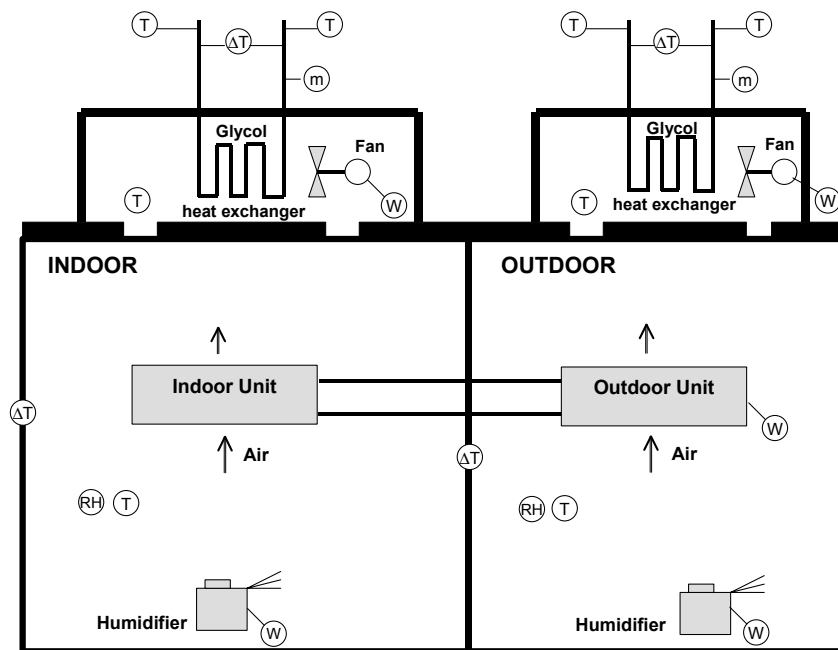


Figure 1: Schematic view of test chambers

Thermocouples of type T were used to measure all temperatures. The power consumption of the baseline unit, fans and humidifiers were measured with watt-meters. Relative humidity was measured in each room. The heat losses to the surroundings and between the chambers were calculated from measured temperature differences over the walls. All parameters were logged every 20 seconds.

2.3 Performance tests

The performance of the baseline system was determined experimentally by calculating the heat balance of the test chambers from the measured parameters for each experiment.

According to measurements at 35°C ambient temperature, the average cooling capacity and COP are 3680W and 2.84, see Table 1. Rated data from the manufacturer at the same operational conditions are 3500W and 3.0.

Preliminary results indicated that the COP is strongly dependent on the compressor speed at heat pump operation. Especially at full speed, the COP drops significantly. Included here are results for close to maximum speed at -5°C outdoor temperature, and lower speed at 5°C.

According to measurements at -5°C ambient temperature, the average heating capacity and COP are 3500W and 2.2, see Table 2. Rated data from the manufacturer at the same operational conditions are 3140W and 2.38.

Table 1: Air condition performance of the baseline R410a system

Test no.	Outdoor		Indoor		Power consumption [W]	Cooling capacity [W]	Cooling COP [-]
	T _{amb} [°C]	RH _{out} [%]	T _{ind} [°C]	RH _{ind} [%]			
	1	34.4	23.9	26.7			
2	35.0	23.1	26.8	51.1	1310	3660	2.79
3	34.5	23.5	26.7	51.5	1300	3570	2.74
4	34.1	22.4	26.6	51.0	1290	3890	3.02
5	34.3	23.4	26.5	51.6	1290	3640	2.82
6	27.7	31.7	26.9	51.3	1140	4120	3.60
7	27.6	31.5	26.8	51.0	1160	4120	3.54
8	27.7	30.7	26.6	51.5	1140	4080	3.58

Table 2: Heat pump performance of the baseline R410a system

Test no.	Outdoor		Indoor		Power consumption [W]	Heating capacity [W]	Heating COP [-]
	T _{amb} [°C]	RH _{out} [%]	T _{ind} [°C]	RH _{ind} [%]			
	9	4.9	62.3	22.0			
10	4.5	59.6	21.0	39.7	1430	3810	2.66
11	4.3	59.4	21.1	38.8	1410	3830	2.72
12	-4.9	60.5	20.7	36.8	1600	3480	2.17
13	-4.4	59.2	20.9	30.6	1600	3390	2.12
14	-4.8	57.0	19.8	33.4	1580	3650	2.32

3 CO₂ PROTOTYPE SYSTEM

3.1 Description of system

The prototype CO₂ unit is a reversible residential air conditioning system. The unit is based on the Shecco™ circuit with a receiver after the evaporator and a suction line heat exchanger. Figure 2 shows the flow sheet of the unit, mounted in a two-chamber test rig. All components except for the indoor heat exchanger is mounted in the outdoor chamber.

In heat pump mode the prototype unit can operate as either conventional one stage circuit or as a two stage system with a sub-cooler. An option at AC operation is cooling of the compressor first stage discharge gas.

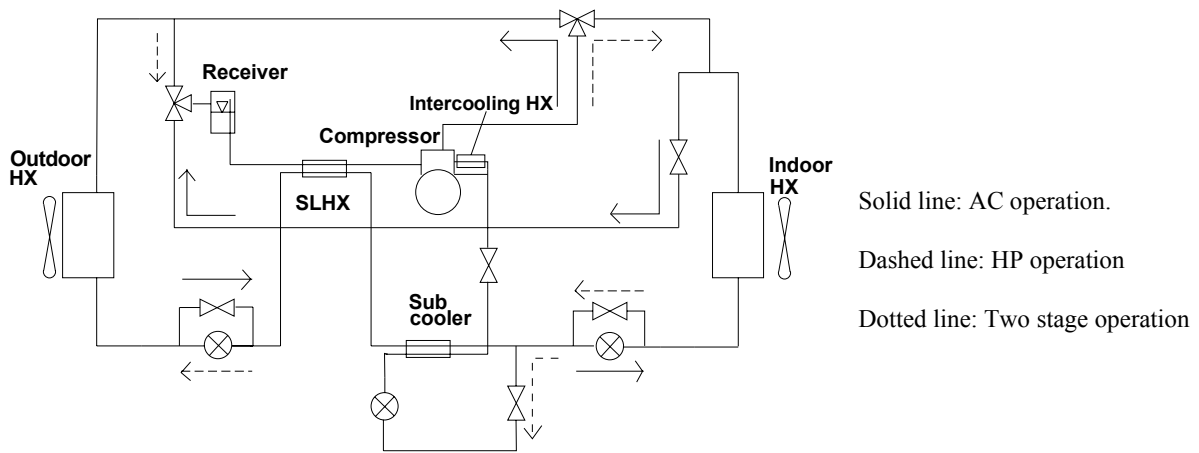


Figure 2: Sketch of prototype RAC CO₂ unit

3.2 Components

The prototype unit has a SANYO two stage rolling piston compressor model no: C-CV113HOW with a displacement of 3.33cm³. It is inverter driven with a voltage of 230V and with frequency range of 30-120Hz. A non-soluble oil (PAG 100cST) is used. Efficiencies for the compressor are discussed in chapter 4.2.

Both the indoor and the outdoor heat exchangers are designed as tube-in-fin with copper tubes and aluminum fins. Data are given in Table 3. In heat pump mode, the refrigerant flow and air flow are (cross) countercurrent. After the first tube row, the fins are split in order to prevent heat conduction from the warm refrigerant inlet. In AC mode, the refrigerant flow is reversed and the airflow and refrigerant flow becomes (cross) co-current.

Table 3: Outdoor and indoor heat exchanger data

	Outdoor HX	Indoor HX
Core size (W x H x D):	600 x 600 x 43mm	650 x 300 x 87mm
Tube size (ID/OD):	6.16/7.6mm	6.16/7.6mm
Fin size (Pitch/ Thickness):	1.4/0.105mm	1.4/0.10mm
Tube configuration:	1 circ., 48 tubes, 2 rows	1 circ., 48 tubes, 4 rows
Tube pitch (H/V)	25/21.65mm	25.0/21.65mm
Air side surface:	21.04m ²	22.78m ²
Refrigerant surface:	0.593m ²	0.638
Surface ratio:	35.7	37.7

The SLHX and the Sub Cooler are counter-flow tube in tube heat exchangers. Outer tube: 16x2mm. Inner tube : 8x2mm. Fins: H: 2mm, W: 1mm, No : 12. Total heat transfer length is 620mm.

The intercooler is a coaxial heat exchanger with CO₂ in the inner tube and water in the outer tube. Water cooling is preferred due to good ability for capacity regulation.

The receiver has a volume of one liter with inspection glasses for visual control of the refrigerant and the oil level. The oil is drained from the bottom of the receiver and the CO₂ gas is sucked from the top.

All piping between the components are made of 1/4" copper tubes with a wall thickness of 0.85mm. The expansion devices are manually regulated needle valves.

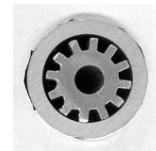


Figure 3: Cross section of SLHX and Sub Cooler

3.3 Indoor heat exchanger size

Due to comfort requirements, there is a maximum “allowable” airflow through the indoor unit. Increased heating capacity at heat pump operation will then imply increased air temperature (at maximum airflow). In order to increase the air temperature in conventional systems, like the baseline R410A system, the condensing pressure has to be increased due to the temperature pinch in the condenser, see Figure 4. Increased heat exchanger size cannot affect this need for higher condensing pressure. It may in best case reduce the condensing pressure slightly with a possible reduction of the temperature pinch. This may be a reason for the relatively small indoor heat exchangers used in conventional RAC systems. An inherent consequence of high condensing pressure at high heating capacity is a relatively poor COP.

For trans-critical CO₂-systems, size of the indoor heat exchanger as gascooler will affect the performance significantly. The larger size, the lower optimum high side pressure, and the temperature profiles of the CO₂ and air will fit closer together, see Figure 4. The heat exchanger loss will be reduced to a minimum. Increased heat exchangers size will hence improve COP. This indicates that indoor heat exchangers for CO₂ RAC systems should be increased in size compared to conventional systems.

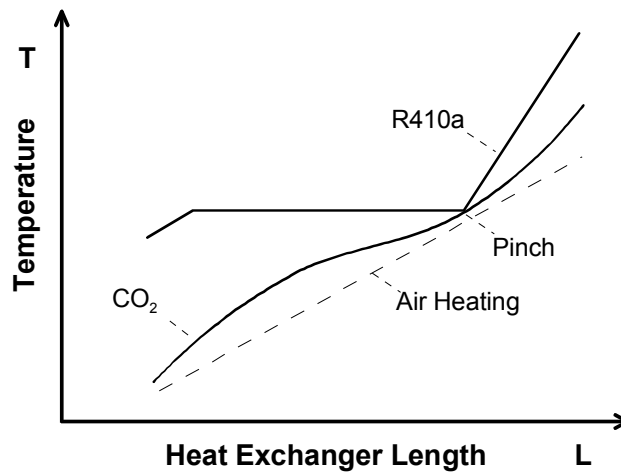


Figure 4: Principle drawing of the temperature profiles of RAC indoor heat exchangers for R410a and CO₂ systems

4 SIMULATION OF THE CO₂ PROTOTYPE SYSTEM

The performance of the laboratory CO₂ prototype system is simulated with the in-house simulation program, CSIM, and compared to the R410A measurements in the same rating points. In addition maximum and part load capacities are simulated.

4.1 Description of the simulation program CSIM

CSIM is a steady state circuit simulator for analyzing the behaviour of trans-critical CO₂ cycles. The program uses as a framework a circuit containing one indoor and one outdoor heat exchanger, a compressor, an internal heat exchanger, an expansion device, a receiver and the necessary refrigerant lines connecting the components. The heat exchangers can operate as evaporator, condenser or gas cooler depending on whether the system works in heating or cooling mode. Either air or water can be used as heat source/sink, thus covering hot water heating, air conditioning-, refrigeration and heat pump systems. CSIM includes fairly detailed heat exchanger models using the actual geometry as input and calculating local values for the heat transfer

coefficient, pressure gradient and void fraction on the refrigerant side. The heat exchangers can also be pre-calculated using heat exchanger design tools like HXSIM by Skaugen (2000) with characteristic data provided as tables or curve-fitted functions over a specific range.

The core of CSIM uses a general optimisation package (NLPQL) by Schittkowski (1985). Use of optimisation techniques can offer a lot of possibilities, both for energy efficient operation and, if data are available, also cost optimal design and operation. CSIM and its models are further described by Skaugen, et al. (2002).

The total number of free variables for optimisation routine is for this problem 218 with a total number of 480 constraints.

4.2 Assumptions and simulation results

The simulation of the CO₂ prototype system has been done for the heating and cooling mode at -5°C and 35°C ambient temperature. Requiring the same capacity as the R410A baseline unit, the coefficient of performance is maximized by varying the high side pressure and compressor speed. Simulations are also done for 5°C and 27.8°C ambient temperature, requiring the same capacity as the baseline unit, although this unit is rated at maximum capacity. This is higher than the needed capacity.

Isentropic and volumetric efficiencies are kept constant at 0.62 and 0.82 based on experience of hermetic CO₂ compressors. The values from the ARTI-21 project (Hubacher and Groll 2002) are assumed to be a bit too high. In the baseline CO₂ system, all simulations are done with a low pressure receiver and a suction line heat exchanger. The evaporator outlet vapour fraction is 0.99.

The internal heat exchanger length was increased from 0.62m in the prototype to 1.30m. The hydraulic diameters are 6.0mm and 3.5mm for the high and low pressure side.

The air face velocities for the indoor and outdoor heat exchangers were kept 0.71m/s and 1.54m/s in all cases.

4.2.1 Cooling mode

In cooling mode at 35°C ambient temperature the measured baseline R410A capacity was 3860W with a cooling COP of 2.83. The predicted CO₂ cooling COP at the same capacity was 2.87. The high side operating pressure was 91.4bar. Maximum cooling capacity at 35°C for the CO₂ unit was 4880W at a high side pressure of 97.7bar.

At 27.8°C ambient temperature the measured baseline R410A capacity was 4110W with a cooling COP of 3.57. The predicted CO₂ cooling COP at the same capacity was 3.67. The high side pressure was 81.4bar.

In a real installation, the cooling demand at 27.8°C ambient temperature should be less than the cooling demand at 35°C . An estimate based on a 3140 W heating demand at -5°C and a 3500W cooling demand at 35°C , a cooling demand in the range of 2200-2500W at 27.8°C is more reasonable. When running the R410A tests it was not possible to control the system in order to provide that capacity. Capacity control in the CO₂ system can be obtained both by varying the high side pressure, and the compressor speed. The optimum high side pressure is mainly governed by the ambient temperature. Only a small reduction in pressure compared to the full capacity can be expected. Using 2500W cooling demand at 27.8°C , a maximum COP was found to be 6.2 with the high side pressure reduced from 81.4bar to 77.8bar.

4.2.2 Heating mode

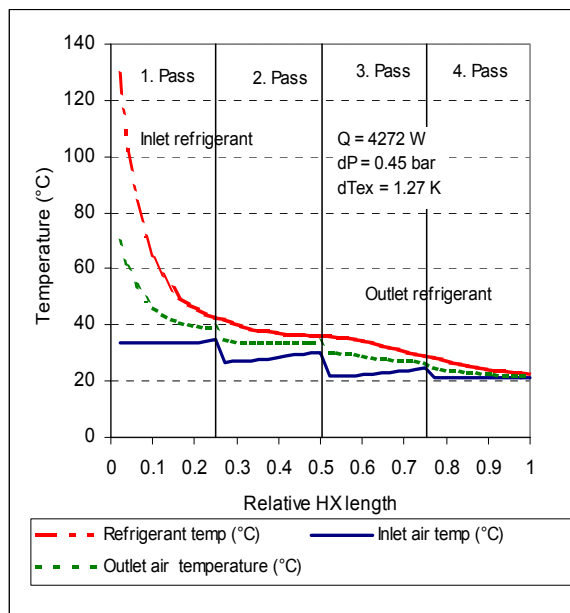
In heating mode, the predicted performance at -5°C and 5°C ambient temperature and 21°C indoor temperature are compared to the measured baseline R410A unit. The measured heating capacities for the R410A system at these conditions are 3540W and 3790W respectively, with a heating COP of 2.23 and 2.78. Again, the rated maximum heating capacity at 5°C is not very interesting for a real installation where the heating demand is less than the heating demand at -5°C . An estimate of 2500W heating demand is used for the part load calculation for CO₂ at 5°C ambient temperature.

The predicted heating COP for CO₂ using the same rated capacity as for R410A was 3.14 and 3.85 which is approximately 40% above the values for R410A. As discussed in Section 3.3, CO₂ benefits from the increased indoor heat exchanger size. The high side pressure was 79.7bar and 81.3bar. The predicted maximum heating capacity at -5°C was for the CO₂ circuit 4270W with a high side pressure of 82.2bar. This solution was constrained by the maximum allowable compressor discharge temperature of 130°C. At part load operation at 5°C the predicted heating COP was 4.54 for a 2500W heating capacity. The high side pressure was 75.0bar.

4.2.3 Heat exchanger performance

Since the heat exchanger models in CSIM are fairly detailed, a collection of results for the indoor heat exchanger operating both as evaporator and gascooler is shown in Figure 5.

Indoor HX heating mode - gascooler



Indoor HX cooling mode - evaporator

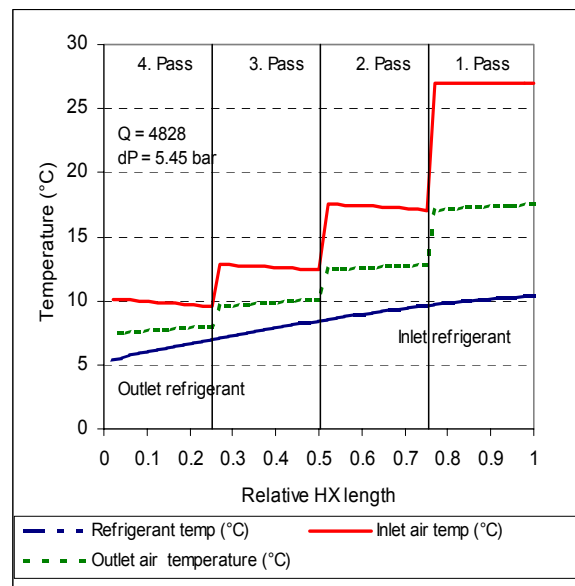


Figure 5: The indoor heat exchanger for maximum heating and cooling capacity at -5°C and 35°C ambient temperatures

The CO₂ temperature and the inlet and outlet air temperatures along the heat exchanger from the CO₂ inlet are shown. The indoor heat exchanger has four rows represented as passes 1-4 in Figure 5. The graphs illustrate how the available temperature difference in different tube rows is utilised.

In heating mode, the indoor heat exchanger is a gascooler with the inlet air temperature of 21°C shown as the lowest curve in pass 4 in Figure 5. The outlet air temperature from the pass 4 is shown as the middle curve, and mirrored as inlet air temperature to the third pass and so on. The outlet air temperature from the gascooler is thus the middle curve in the section for pass 1.

In cooling mode, the indoor heat exchanger is acting as an evaporator and has reversed refrigerant flow. The inlet air temperature of 26.7°C is entering across the 1. pass, and is cooled to 8°C across the four tube rows. The pressure drop at maximum compressor speed and maximum cooling capacity is calculated to 5.45 bar, and the advantage of having the air and the refrigerant flowing in co-current direction is illustrated in the figure.

4.3 Summary of results from the CO₂ prototype simulations and the R410A measurements

The main results from both cooling and heating mode are shown in Figure 6. The measured COP and capacities for R410A is compared to the predicted values for the CO₂ laboratory prototype. As seen, in cooling mode the CO₂ laboratory prototype have equal cooling COP as the baseline unit,

even at the artificial high capacity operation at 27.8°C. When utilizing full part load control for CO₂, the COP values can be increased by more than 70%.

For heating mode, the COP for the CO₂ unit is approximately 40% higher than the R410A unit at equal capacity. When running in part load operation, the CO₂ unit has 63% higher COP than the R410A unit running at full capacity. The R410A unit was not tested at this low part load.

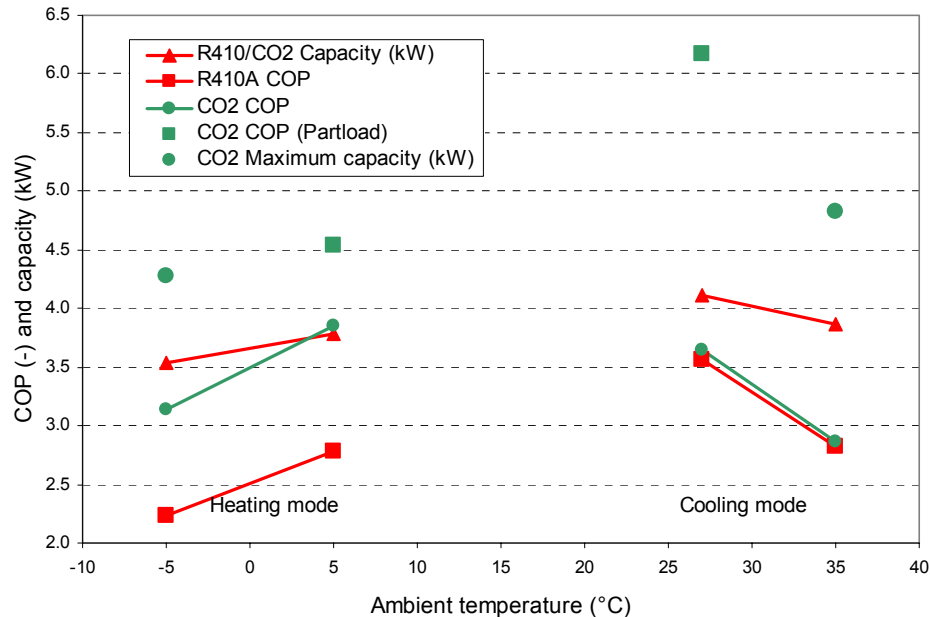


Figure 6: Main results from the comparison between the CO₂ laboratory prototype simulations and the R410A RAC unit measurements in cooling and heating mode operation

5 CONCLUSIONS

A reversible CO₂ prototype RAC split unit system has been designed, built and installed in two calorimetric test chambers at the SINTEF laboratories. Experimental tests are under preparation.

Realistic simulations of the CO₂ prototype system have been carried out with the advanced in-house simulation program CSIM using input data for all components of the prototype. A state of the art inverter driven R410A split unit has been tested experimentally as baseline unit. Obtained experimental results have been compared to simulation results for the CO₂ prototype.

Simulation results indicate that the prototype CO₂ unit will perform almost equal compared to the achieved R410A baseline measurements in AC mode, running at 27.8°C and 35°C ambient temperature. In heat pump mode, the prototype CO₂ unit is predicted to perform around 35 to 40% better at -5°C and 5°C ambient temperature. Future experimental work will reveal whether the potential for this new CO₂ system can be achieved.

Simulation of the CO₂ prototype heat exchangers revealed that the air and refrigerant flows should be (cross) co-current in operating as evaporator and (cross) counter-current operating as gas cooler. Furthermore, increased indoor heat exchanger size will reduce the optimum high side pressure for CO₂ systems in heat pump mode, leading to improved performance. Increased heat exchanger size in conventional HFC-systems cannot reduce the need for high condensing pressures if high outlet air temperature or capacity is required. The COP will suffer.

6 REFERENCES

- Aarlien R., 2001, On Design, Efficiency, and Market Potential of Residential Air Conditioning and Heat Pump Units with CO₂ as Working Fluid, *Thesis, Norwegian University of Science & Technology NTNU, ISBN82-471-5309-2*
- Hafner A., Jakobsen A., Nekså P., Pettersen J., 2004, Life Cycle Climate Performance (LCCP) of Mobile Air-Conditioning Systems with HFC-134a and R-744, *VDA Alternate Refrigerant Wintermeeting 2004, Saalfelden, Austria*
- Hubacher B., Groll E., 2002, Measurement of Performance of Carbon Dioxide Compressors, *ARTI Report No.: 21CR/611-10070-01*
- Kim M.-K., Pettersen J., Bullard C. W., 2004, Fundamental process and system design issues in CO₂ vapor compression systems, *Progress in Energy and Combustion Science, Vol 30 pp 119-174*
- Nowacki J.E., 2002, Single Room Heat Pumps for Cold Climates. *Analysis Report HPC AR14, IEA Heat Pump Programme.*
- Richter, M.R., Song, S.M., Yin, J.M., Kim, M.H., Bullard, C.W. and Hrnjak, P.S., 2000, Transcritical CO₂ Heat Pump for Residential Application, *Natural working fluids; Final proceedings of the 4th IIR-Gustav Lorentzen conference on natural working fluids at Purdue*, International Institute of Refrigeration; 2000, p. 59-67.
- Schittkowski, K., 1985, NLPQL: A FORTRAN Subroutine Solving Constrained Nonlinear Programming Problems, *Annals Operations Research*, vol. 5: p. 485-500.
- Skaugen, G., 2000, Simulation of Extended Surface Heat Exchangers Using CO₂ as Refrigerant, *Natural working fluids; Final proceedings of the 4th IIR-Gustav Lorentzen conference on natural working fluids at Purdue*, International Institute of Refrigeration; 2000, p. 306-314.
- Skaugen, G., Nekså, P. and Pettersen, J., 2002, Simulation of trans-critical CO₂ vapour compression systems, *5th IIR-Gustav Lorentzen Conference on Natural Working Fluids*, International Institute of Refrigeration 2002.

RESUMÈ

Un système fendu réversible d'unité du prototype RAC de CO₂ a été conçu, établi et installé dans deux chambres calorimétriques d'essai. L'unité est basée sur le circuit de Shecco™ avec un récepteur et une canalisation d'aspiration échangeur de chaleur. Les échangeurs de chaleur d'intérieur et extérieurs ont "tubé la conception dans aileron". Le compresseur hermétique a deux étapes de compression et est inverseur conduit. Les essais expérimentaux sont en cours de préparation. Des simulations réalistes ont été effectuées avec le programme avancé CSIM de simulation en utilisant des données d'entrée pour les composants du prototype. CSIM est un simulateur de circuit d'état d'équilibre (principalement) pour les cycles trans-critiques de CO₂, qui emploie les modèles bien calibrés pour tous les composants. Une unité fendue de R410A conduite par inverseur du dernier cri a été examinée expérimentalement comme unité de ligne de base. Des résultats obtenus ont été comparés aux résultats de simulation pour le prototype de CO₂. Les résultats prouvent que l'unité de CO₂ de prototype exécutera légèrement meilleur que l'unité de ligne de base de R410A dans le mode à C.A. à la température 27.8°C et 35°C ambiante. Dans le mode de pompe à chaleur, l'unité de CO₂ de prototype exécute autour 30 à 40% mieux à la température -5°C et 5°C ambiante.