OPTIMIZING HEAT PUMP DESIGN FOR REFRIGERANTS R407C AND R410A

Eric Winandy, Director Application Engineering,
Enrico Fraccari, Product Manager Residential Comfort
Guy Hundy, Technical Consultant,
Emerson Climate Technologies, Pascalstrasse 65,52076 Aachen, Germany

Abstract: This theoretical study based on actual compressor performance and backed up by experimental evidence shows how condensing temperature can be minimised and evaporating temperature maximised by circuit design and control. The effect of temperature glide in the heat exchangers can be used to good effect with R407C and use of a suction/liquid line heat exchanger can improve evaporator utilisation. With R410A, temperature differences are limited by other factors and compressor performance is better. Seasonal analysis for space heating both underfloor and radiator, using the latest vapour injection scroll technology indicates where R410A shows advantages over R407C

Key Words: heat pumps, simulation, efficiency

1 INTRODUCTION

The technique for using compressor data for simulation of heat pump performance was reported in previous papers, Winandy and Hundy, 2008, 2007. The study reported in the IEA paper also demonstrated very good correlation between the simulated data for several rating points and the test data obtained from WPZ tests. The test points were chosen to correspond to the WPZ data at EN14511 conditions. The simulation technique has now been used to make a detailed comparison between R407C and R410A in air and ground source heat pumps. In each case the minimum practical temperature differences in the heat exchangers have been applied so that this represents a best case scenario for each refrigerant. The benefits of each refrigerant in these applications is investigated.

2 TEMPERATURE DIFFERENCES IN CONDENSERS

A visualisation of the heat exchanger temperature profiles in plate heat exchanger (PHX) condensers is given in Figures 1 and 2. For R407C the limiting condition is seen to be at the water inlet. This “pinch point” is governed by the need to maintain a positive temperature difference (2K) at this point. Any subcooling at that point will increase the temperature difference and hence increase the condensing temperature. The negative effect on COP of raising the condensing temperature is greater than the positive effect of increasing the enthalpy difference in the heat exchanger. A liquid receiver is therefore beneficial and in practice with R407C it tends to result in a small amount of subcooling due to stratification, typically 1.5K. The effect of refrigerant pressure drop is small and has been neglected.

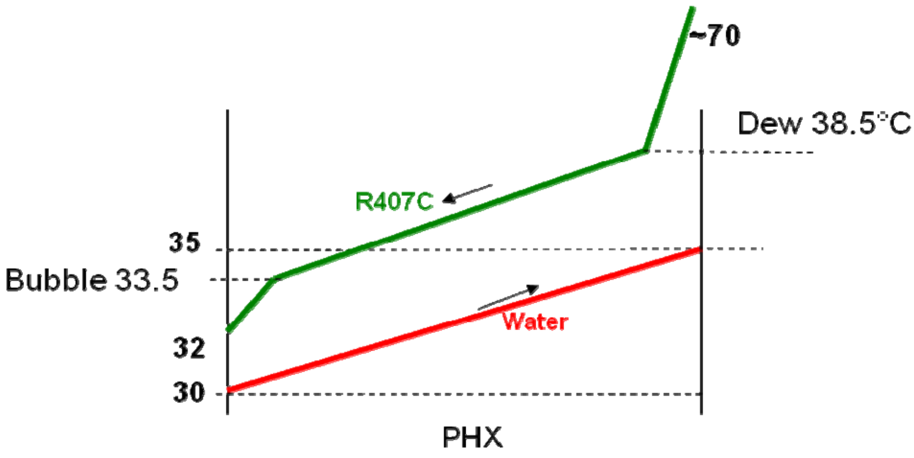


Figure 1: Condenser Temperature Profiles with R407C

On the contrary, for R410A the pinch point is at the water outlet and there is also a larger average Temperature difference through the condensation process. At the water outlet itself the high temperature gas is de-superheated, and so, with reference to the water outlet temperature, the condensation temperature difference can be as low as 1.5K. In that case, subcooling is beneficial as it increases the heating capacity without affecting directly the condensing temperature, and 3.5K subcooling is possible without liquid receiver.

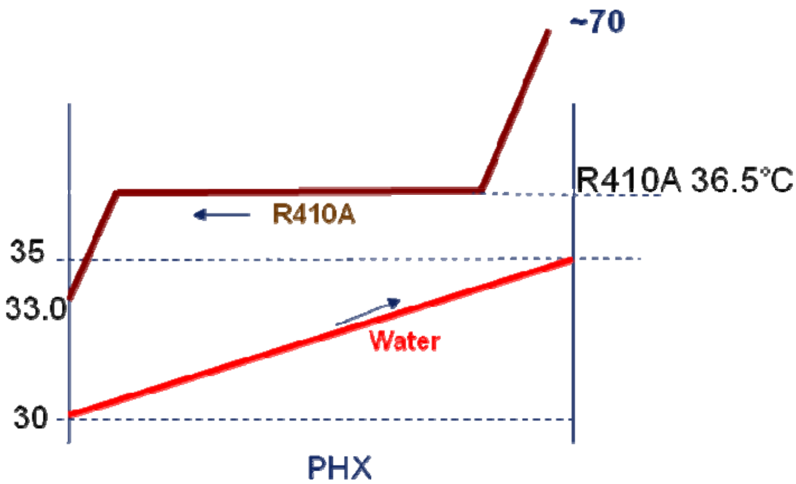


Figure 2: Condenser Temperature Profiles with R410A

3 TEMPERATURE DIFFERENCES IN PLATE HEATEXCHANGER EVAPORATORS

The temperature profiles are shown in Figures 3 and 4 for R407C and R410A respectively. For both refrigerants, with a standard expansion device requiring about 5K, the maximum evaporating temperature is limited by the superheat required (bold lines). But the inherent glide of R407C during its evaporation, a suction/liquid heat exchanger (Figure 5) allows the evaporation temperature to be increased by taking the superheated region out of the evaporator without any other constraint. In that case, an increase of 3K is possible in practice (Figure 3, dashed lines). Note that due to difficulties in keeping a stable control loop at the level of the expansion device, it is better to retain some superheat at the evaporator outlet. By using a suction/liquid heat exchanger with R407C in this way the practical maximum evaporating temperature is -3°C.

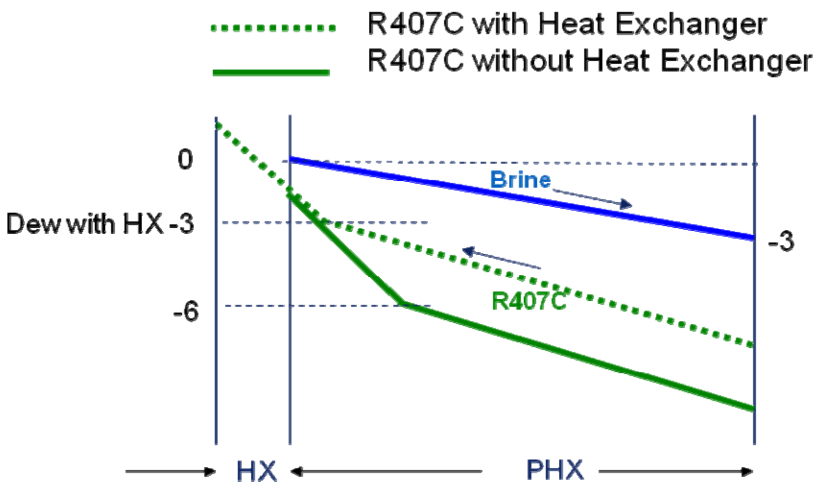


Figure 3: PHX Evaporator Temperature Profiles with R407C

In the case of R410A, a suction/liquid heat exchanger will not be as beneficial since a raise of evaporation temperature will be limited by another pinch point appearing at the brine outlet zone for the Plate Heat Exchanger due to the absence of glide with R410A. In that case, the best that can be done is to reduce the superheat to a minimum value with the help of an electronic valve for example. An evaporation of -4°C at rating conditions is possible then (Figure 4, dashed lines). So minimum temperature difference can be achieved with R410A if superheat controlled at 2K. Beyond this point the TD at the brine outlet starts to impose a limit.

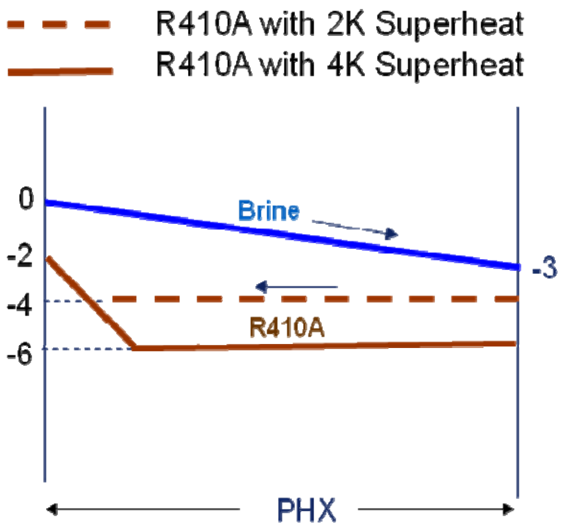


Figure 4: PHX Evaporator Temperature Profiles with R410A showing a superheat of 4K for conventional TEV and 2K for an electronic valve

The diagrams in Figure 5 show the location and effect of the suction/liquid line heat exchanger. This device would not be necessary for R410A.

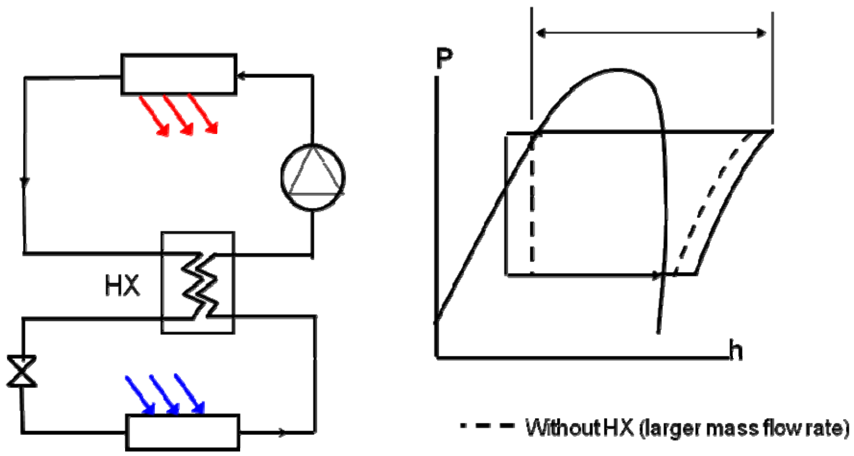


Figure 5: Suction/Liquid Line Heat Exchanger

4 TEMPERATURE DIFFERENCES IN AIR COIL EVAPORATORS

For air coil evaporators a minimum temperature difference of 8K has been chosen because it corresponds to state-of-the-art coils for both refrigerants. A slightly lower temperature difference should be achievable with R410A because of its inherently better heat transfer properties compared to R407C. Having said that, there are other considerations such as cost, dimensions and defrost capability that will influence the coil performance.

5 SUMMARY OF BOUNDARY CONDITIONS

Table 1: Ground Source Boundary Conditions

	R407C	R410A
Pump power, % compr power*	5%	5%
Superheat	4K	2K
Delta T, (Dew – Water Out)	3.5K	1.5K
Delta T, (Dew – Brine Out)	3K	4K
Subcooling	1.5	3.5

*Fixed value, based on design condition

Table 2: Air Source Boundary Conditions

	R407C	R410A
Fan power , % compr power*	10%	10%
Superheat	4K	4K
Delta T, (Dew – Water Out)	3.5K	1.5K
Delta T, (Airside)	8K	8K
Subcooling	1.5	3.5

*Fixed value, based on design condition

6 SIMULATION RESULTS FOR RATING POINT

For ground source B/W heat pumps four compressor technologies have been investigated, and are shown in Figure 6:

ZP36 Air conditioning R410A Scroll Compressor
ZH09K1P Heating Optimized R410A Scroll Compressor
ZHI11K1P Heating Optimized R410A Scroll Compressor with vapour Injection
ZH21 Heating Optimized R407C Scroll Compressor
ZH13 Heating Optimized R407C Scroll Compressor with vapour Injection

The Heating Optimized type has a higher compression ratio scroll set together with a dynamic discharge valve. This gives a flatter isentropic efficiency curve with better isentropic efficiency at the higher pressure ratios required for heating applications.

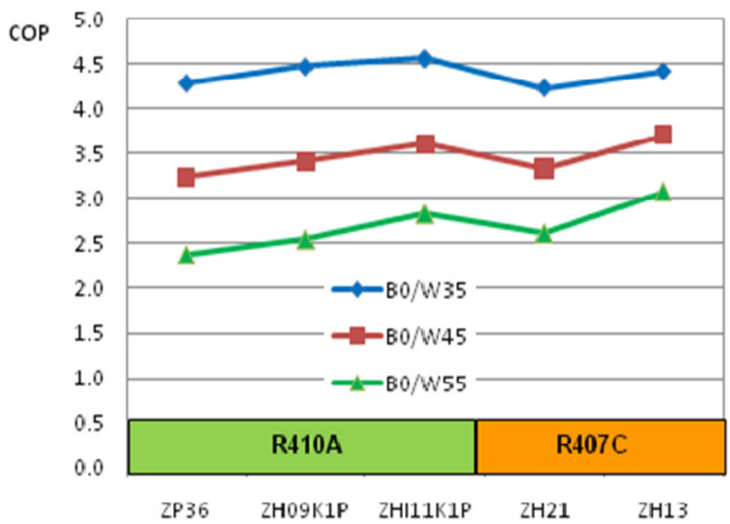


Figure 6: Ground Source HP COPs with water temperatures 35, 45, 55°C

- R410A gives better performance when all the boundary conditions are optimized.
- For underfloor heating at 35°C the benefit of vapour injection is mainly in heating capacity. The vapour injection model has approximately 18% more capacity than the standard heating model. This can be easily found from Compressor Selection Software (2010) The COP advantage is small as can be seen from Figure 6.
- There is better COP advantage for vapour injection with higher water temperatures, 45 and 55°C, Figure 6. There is also more heating capacity enhancement.

For Air Source, examples of the same scroll technologies have been analysed, with separate charts for 35°C water and 55°C water, Figures 7 and 8:

ZP42 Air conditioning R410A Scroll Compressor
ZH12K1P Heating Optimized R410A Scroll Compressor
ZHI11 K1P Heating Optimized R410A Scroll Compressor with vapour Injection
ZH38 Heating Optimized R407C Scroll Compressor
ZH13 Heating Optimized R407CScroll Compressor with vapour Injection

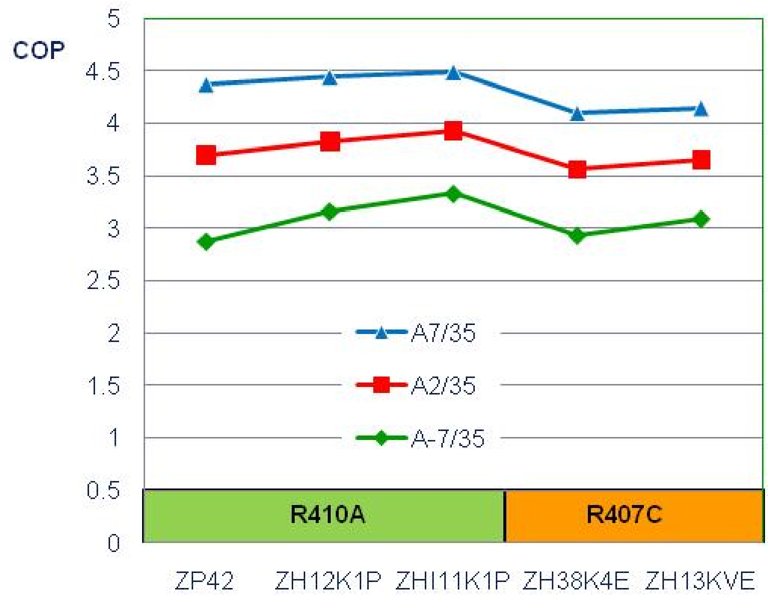


Figure 7: Air Source HP COPs with water temperature 35°C

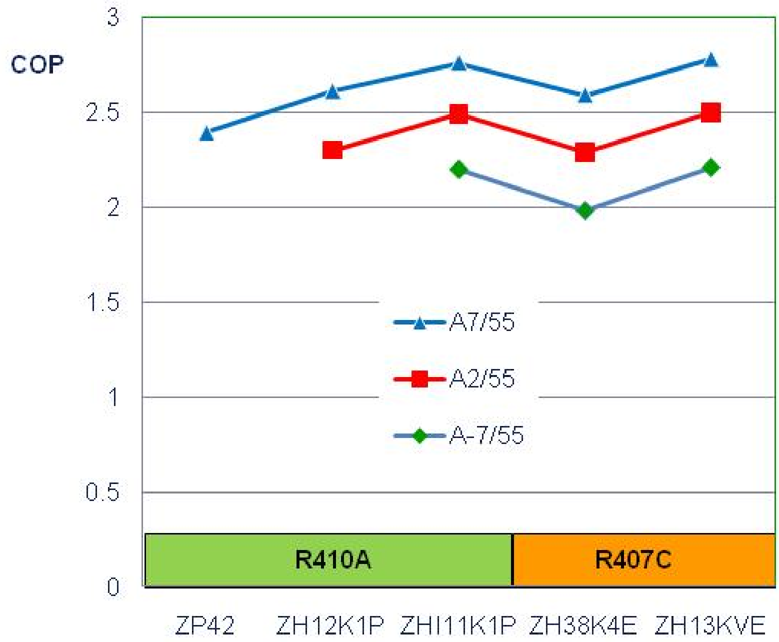
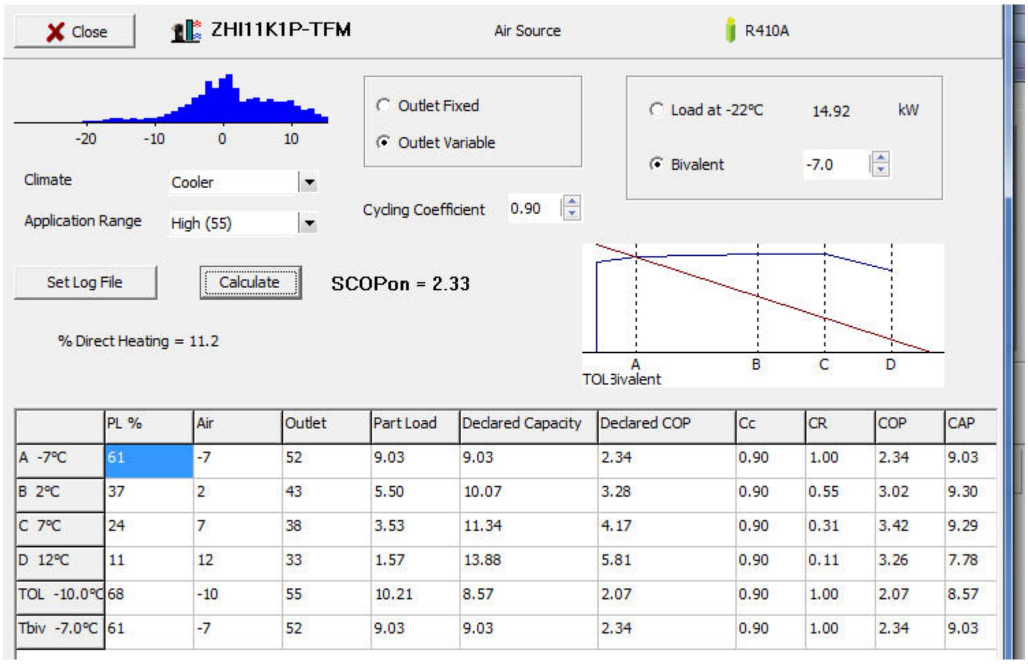


Figure 8: Air Source HP COPs with water temperature 55°C

- R410A gives distinctly better performance at the lower water temperature, and is equivalent to R407C at high water temperature where its lower critical temperature starts to gain influence.
- Vapour injection shows a distinct COP advantage, particularly at lower air temperatures.
- Operation at low air temperatures for non-vapour injected models is limited by the discharge temperature with both refrigerants. The absence of simulated COP points in Figure 8 shows the influence of this limitation. The operating envelope diagrams for all models can be found in the Compressor Selection Software (2010).

7 SEASONAL PERFORMANCE

Seasonal performance has been investigated for air source models using software based on the prEN14825 method. The term SCOP, Seasonal Coefficient of Performance, is defined as the ratio of heating annual heat delivered to annual electrical energy absorbed, including pump/fan (tables 1,2). Figure 9 shows a typical screen with the cooler climate profile illustrated at the top left. The graph visualizes the load and the HP capacity, showing the bivalent point. In this example the cooler climate with 55°C water outlet is chosen.



	PL %	Air	Outlet	Part Load	Declared Capacity	Declared COP	Cc	CR	COP	CAP
A -7°C	61	-7	52	9.03	9.03	2.34	0.90	1.00	2.34	9.03
B 2°C	37	2	43	5.50	10.07	3.28	0.90	0.55	3.02	9.30
C 7°C	24	7	38	3.53	11.34	4.17	0.90	0.31	3.42	9.29
D 12°C	11	12	33	1.57	13.88	5.81	0.90	0.11	3.26	7.78
TOL -10.0°C	68	-10	55	10.21	8.57	2.07	0.90	1.00	2.07	8.57
Tbiv -7.0°C	61	-7	52	9.03	9.03	2.34	0.90	1.00	2.34	9.03

Figure 9: SCOP calculation screen in Selection Software

Figures 10 - 13 are example results of SCOP analysis using the above method for Air/Water heat pumps. This tool enables rapid comparisons between various compressor technologies when applied in different climates and with different distribution systems. In Figures 10 and 12 the distribution is to underfloor heating at 35°C and in Figures 11 and 13 the distribution is via radiators at 55°C. SCOPs have been translated in annual running cost terms for the model sizes used in the analysis. An average electricity cost of 0.16 Euro per kWh has been applied.

The bivalent point for cold climate is -7°C, and for the average climate -1°C. Building load is 19kW at -22°C for the cold climate and 13kW at -10°C for the average climate (same building envelope). For air temperatures below the bivalent point, back up direct electric heat is added as necessary to fulfill the load requirement. The back up heater annual energy input is between 1.7 and 3% of the total annual load, except in the case of the air conditioning scroll when applied to air source, where it is 8%. This is because in the lowest ambient temperature bands this scroll type is unable to operate due to discharge temperature limits. The vapour injected type will deliver a higher proportion of the heat heat demand below the bivalent point, and hence minimize the back up heat requirement.

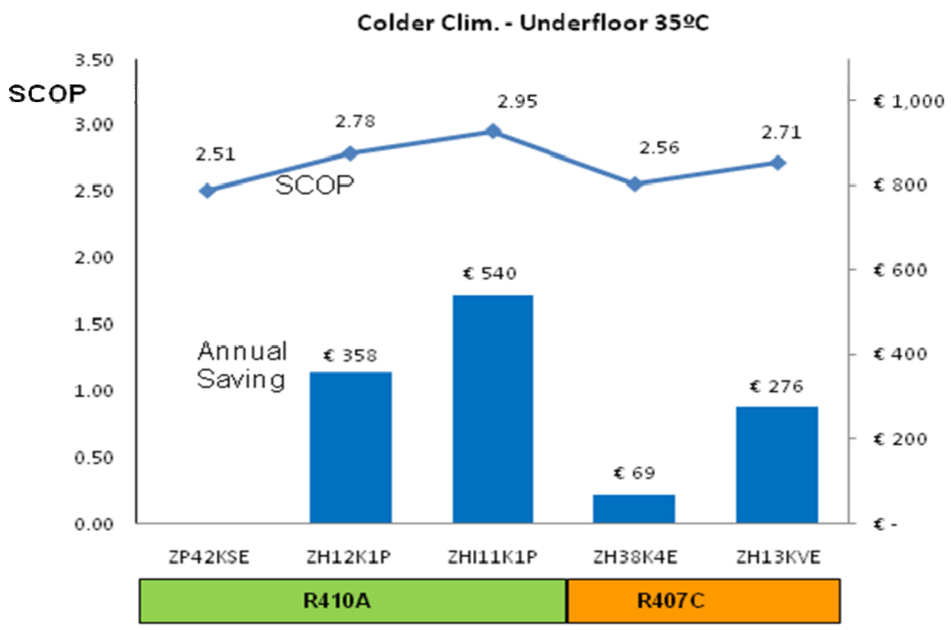


Figure 10: SCOP values and corresponding annual savings in the colder climate. The cost baseline is ZP42KSE

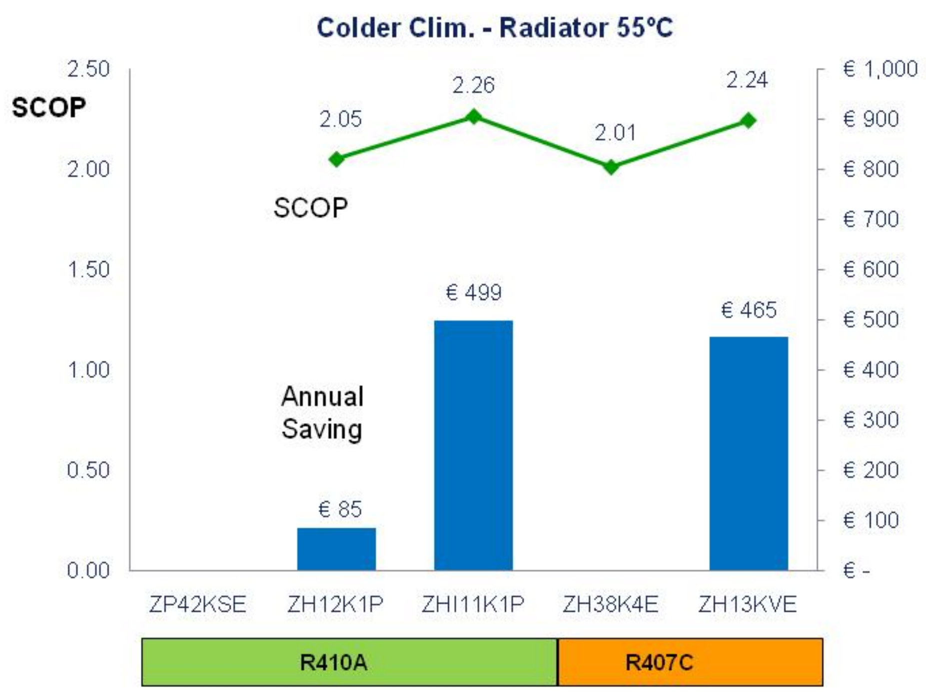


Figure 11: SCOP values and corresponding annual savings in the colder climate. The cost baseline is ZH38K4E

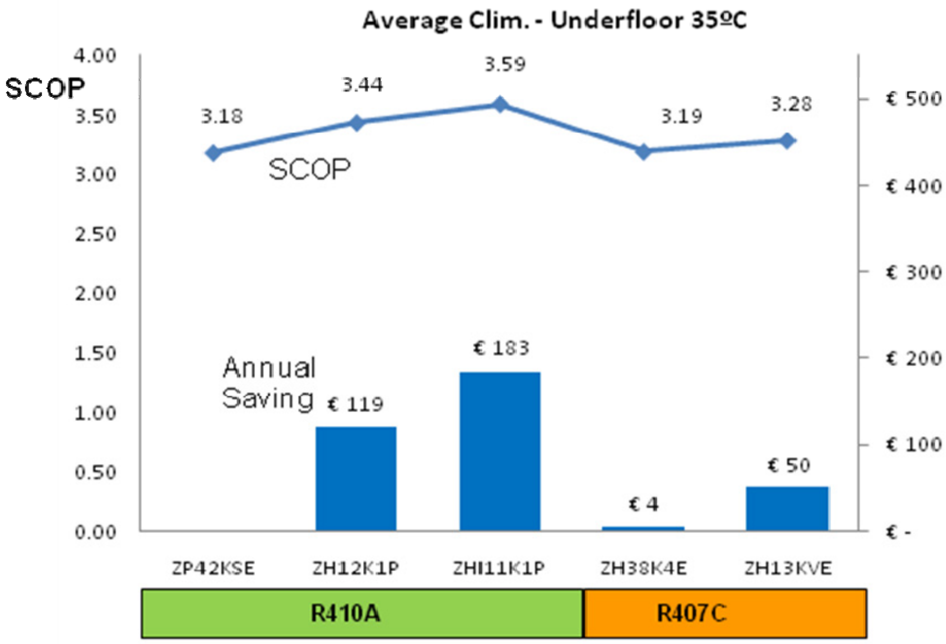


Figure 12: SCOP values and corresponding annual savings in the average climate. The cost baseline is ZP42KSE

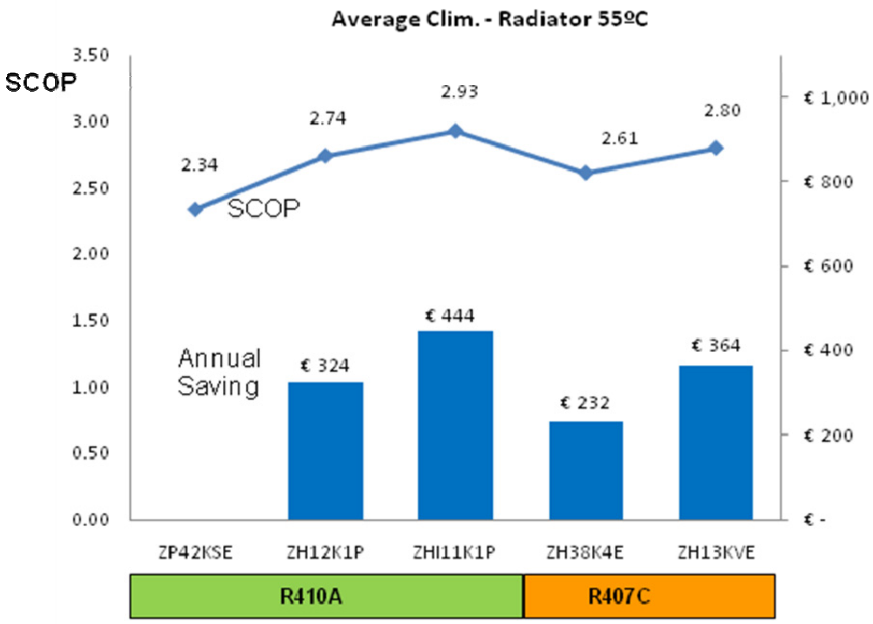


Figure 13: SCOP values and corresponding annual savings in the average climate. The cost baseline is ZP42KSE

8 CONCLUSIONS

A direct refrigerant to refrigerant properties comparison is insufficient to determine the best refrigerant for a given heat pump application. The compressor influence as well as the system influence will not vary always in the same direction and a detailed analysis is needed. In this paper, it has been shown how system design choices can be directly influenced by the inherent properties of the refrigerant and maximize their beneficial effect.

As a result, it has been shown that R410A has the potential to outperform R407C in most applications when using optimized scroll technology and when considering the most favourable boundary conditions in each case.

- Vapour injection give a good SCOP advantage at highest lift and therefore gives most benefit with air source/radiator systems

.

- When applying standard air conditioning scrolls with air source systems, direct heating is required at low air temperatures because of envelope limits, and this reduces SCOP.

9 REFERENCES

Winandy E, Hundy G, Investigation Of The Benefits Of The Vapour Injection Scroll Compressor In Heat Pumps Using A Seasonal Efficiency Model, IEA 2008

Winandy E, Hundy G, Hewitt N, Modelling Air Source Heat Pump Seasonal Efficiency, IIR Congress 2007.

Emerson Climate Technologies, SELECT 7.3, Compressor Selection Software 2010