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Energy Efficiency of a Chiller Using R410A or R32

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ABSTRACT

An advanced simulation model of a packaged air cooled water chiller was developed. The nominal cooling capacity of the chiller is about 70 kW at 35°C dry bulb outdoor air temperature, and it consists of a single refrigerating circuit with two identical scroll compressors. The compressor was characterized by its experimental performance curve according to EN 12900 for both R410A and R32. Off-the-shelf copper tubes and louvered aluminum fins were considered for the condenser and typical brazed plates configuration for the evaporator. The finned coil condenser and the brazed plate heat exchanger evaporator were modeled by an elementary finite volumes technique, previously validated in other papers; for each volume the heat transfer coefficient and the pressure drop were calculated using semi-empirical correlations chosen among the most accurate available in the literature. The aim of the paper is to compare the performance of R410A with that of R32 which is considered a possible HFC substitute with lower GWP (675 instead of 2088). The condenser heat exchanger was optimized for R410A and R32 independently with regard to the number of internal circuits according to the Total Temperature Penalization Performance Evaluation Criteria (Cavallini et al., 2010), without changing the overall heat exchanger dimensions. Furthermore a seasonal efficiency analysis is carried out for different finned coil circuit configuration in terms of European Seasonal Energy Efficiency Ratio (SEER) calculated according to the European Standard EN 14825 for water chillers. Basing on the proposed modeling work, it is possible to conclude that R32 system efficiency performance is acceptable as alternative to R410A for water chillers and further simulation analyses will include evaluating the HFO blends.

1. INTRODUCTION

R32 has been considered as a good refrigerant since "The Day After CFCs" (Cavallini, 1995). However, it has been used for more than 15 years only as a component in refrigerant mixtures. The impact of fluorinated refrigerants on global warming has led to the so-called "mildly" flammable refrigerants becoming well accepted and, R32 has been revaluated as a serious candidate in comfort applications.

Huang et al. (2011) studied a household reversible air conditioning unit and found that the performance of R32 is close to R410A at standard cooling condition and superior at standard heating condition. The capacity of an R32 heat pump is higher than that of an R410A heat pump with the same compressor displacement.

Xu et al. (2013) investigated the performance difference in a heat pump system with vapor injection by using R410A and R32. They found that the capacity and coefficient of performance (COP) with R32 increase up to 10% and 9% respectively, compared to an identical cycle using R410A. They concluded that R32 heat pump performance would be further enhanced by an optimization of the system. One of the major issues reported in the literature is related to the higher discharge temperature when operating the system with R32 instead of R410A and to the miscibility gap of R32 in POE lubricants developed for R410A (Bella and Kaemmer, 2011 and 2013). Xu et al. (2011) analyzed a "quasi two stage" heat pump to mitigate the increase of refrigerant discharge temperature when using R32 instead of R410A. Bella and Kaemmer (2013) proposed the injection of dry refrigerant vapor at intermediate pressure to mitigate the discharge temperature. They found that this strategy reduces significantly the R32 discharge temperature, increases the R32 compressor efficiency and cooling capacity.

When looking for the substitution of a refrigerant in a particular application, the more optimistic scenario would be "drop in": i.e. one simply replaces the refrigerant by only tuning the refrigerant charge and adapting the throttling expansion valve flow coefficient or its control law. Pham and Rajendran (2012) presented an experimental drop-in performance comparison with R32 and R410A in a 3-ton reversible unit in cooling and heating mode. They found up to 3.3% higher cooling capacity, up to 3.9% higher heating capacity with R32. The EER in cooling mode was 1.5% lower whereas the COP in heating mode was between -0.8 and +0.9%, depending on the working conditions. They found a slightly higher evaporation temperature with R32 (despite of the higher evaporator heat duty, both in cooling and in heating mode). However, the consequence was an increase in the condensation temperature.

In principle, assuming that the unit design is optimized for the original refrigerant (in this case R410A), there is no evidence that the same design would be so for the new refrigerant (R32).

Cavallini et al. (2010) demonstrated that it is possible to optimize the circuit length for a finned coil air heat exchanger of given overall dimensions for a specific refrigerant. Usually, the designer has to optimize the number of circuits and their lengths based on the refrigerant mass flux.
This choice is an optimization between two opposite effects on the heat exchanger. The higher the mass flux, the greater the heat transfer coefficient and thus the heat exchanged. On the other side, the higher the mass flux, the greater the pressure drop. In a heat transfer process such as evaporation or condensation, the pressure is directly related to the saturated temperature, so a fall in pressure means a fall in temperature. Hence, these refrigerant temperature drops decrease the average driving temperature difference with the secondary fluid.
Cavallini et al. (2010) introduced a new method to optimize the finned coil circuitry. The authors defined two performance evaluation criteria termed penalty factor (PF) and total temperature penalization (TTP) and demonstrated that TTP is a useful parameter to optimize the heat exchanger circuitry. The optimal circuit length should have the minimum TTP value. The TTP is the sum of the temperature difference between the refrigerant and the wall temperatures (the "driving" temperature difference) and a half of the saturation temperature drop. Both temperature differences can be evaluated with reference to the average temperatures at the average vapor quality of 0.5.
The reader may refer to the above cited reference for a detailed description of the TTP concept. The TTP is a useful tool, it gives the designer the opportunity to optimize the air cooled finned coil circuitry (that means to change the curves lay-out of the condenser), without modifying the overall dimensions, the fins, spacing and the fans. Basing on the above conclusions, a "drop-in" operation will be successful from an efficiency point of view if both the refrigerants (the original and the substitute) display same or lower TTP with same circuit length or by simply changing the circuits lay-out
In the present paper the TTP approach is carried out for the finned coil heat exchanger of a water chiller unit. Afterwards, a comparison of efficiency at full load and at seasonal efficiency according to the European standard EN 14825 is performed using a simulation tool for the whole refrigeration unit operating with R410A and with R32.

2. IDEAL VAPOR COMPRESSION REFRIGERATION CYCLE

In order to gauge the performance potential of R32, this section compares R410A and R32 in an idealized vapor compression refrigeration cycle. The cycle chosen is meant to be representative of typical comfort cooling operating conditions, and is specified by a constant evaporation temperature of 5°C, a condensation temperature of 45 °C, a superheat of 5 K and a sub-cooling of 5 K
The thermodynamic properties of R32 and R410A were evaluated with REFPROP 9.1 (Lemmon et al. 2013).
Table 1 is comparing the main performances with R410A and R32. The volumetric cooling capacity (VCC) is the product of the suction vapor density and the evaporator specific enthalpy change.

Table 1: Cycle performance with R410A and R32

	VCC (kJ/m3)	COP	Dis. Temp. (°C)
R410A	5613	5.39	67
R32	6172	5.52	80
Variation	+10%	+2.4%	13

In an ideal cycle, it is possible to make the following observations:

- First, the volumetric cooling capacity with R32 is about 10% higher. The R410A suction vapor is some 38 % denser while the refrigeration capacity per mass of R32 is some 52% greater, leading to 10% higher volumetric cooling capacity.
- Second, the compressor work is some 48 % higher with R32; whereas the refrigeration capacity per mass is larger for R32 is 52% larger leading to 2.4 better COP with R32.
- Finally, R32 discharge temperature is some 13 K higher, implying a larger superheat loss for R32. This is a direct result of larger slope for the saturated vapor line for R410A;

In the next section, a more detailed simulation is adopted for exploiting non-ideal effects, such as compressor losses and exergy losses in heat exchangers in an air cooled water chiller.

3. "REAL" VAPOR COMPRESSION REFRIGERATION CYCLE PERFORMANCE

In the present work the performance of R410A and R32 inside an air cooled water chiller is carried out by means of a numerical analysis. The studied unit consists of a single refrigeration circuit with two scroll compressors in parallel, two parallel air cooled finned coil condensers and one brazed plate heat exchanger evaporator.
The numerical approach described in Casson et al. (2002) has been used to the design the finned coil heat exchanger (condenser).

The heat exchanger is discretized in a finite volume, each element is calculated as small cross flow heat exchanger, and the conditions of both the primary refrigerant and secondary coolant are passed to the adjacent volume.
The correlations used for the refrigerant-side such as heat transfer coefficient, pressure drop, and void fraction re listed in Table 2. The air side heat transfer coefficients are calculated according to semi-empirical correlations developed from an experimental data base collected in Padova University. The air side heat transfer coefficients are assumed to be constant and uniform in the whole heat exchanger. The main geometric characteristics are reported in Table 3and are chosen according to atypical air-conditioning European production. The overall frontal dimension of the chosen finned coil is 1.9 m height and 1.5 m length. Two parallel finned coil heat exchangers are used in the simulated tool.
Once fixed the overall geometry of the heat exchanger as reported in Table 3, the single circuit length is changed and the correspondent TTP is calculated for both R410A and R32 at nominal full load conditions, as per Table 4.
It is worth noticing that the liquid subcooling is assumed identical for both R410A and R32, in line with the analysis proposed by Casson et al. 2003. In that work it is demonstrated thermodynamically that the optimal condensate subcooling (i.e. the one giving the maximum COP) is identical for R410A and R32 in a range between 2 and 6 K, depending on the working conditions.

Table 2: Correlations used for the evaporator and the condenser

Refrigerant-side heat transfer coefficient for the evaporator	Longo and Gasparella (2007)
Refrigerant-side heat transfer coefficient for the condenser	Cassonet al. (2002)
Refrigerant-side pressure drop for the evaporator	Longo and Gasparella (2007)
Refrigerant-side pressure drop for the condenser	Cavallini et al. (2002)
Refrigerant void fraction	Rouhani and Axelsson (1970)
U-bend pressure drop	Equivalent length = 30*D*
Water-side heat transfer coefficient for the evaporator	Muley and Manglik (1999)
Air-side heat transfer coefficient for the condenser	Manufacturer proprietary data

Table 3: Finned coil condenser Geometry

Tube arrangement		Staggered
Tube material		Cu
Finned coil length	(m)	1.5
Outside tube diameter	(mm)	10
Inside tube diameter	(mm)	8
Longitudinal tube spacing	(mm)	19
Transverse tube spacing	(mm)	25
Fin geometry (material)		corrugated (Al)
Fin spacing	(mm)	2.1
Fin thickness	(mm)	0.11
Number of rows		3
Face air velocity	(m/s)	1.5

Table 4: Full load operating conditions

Condensate Sub-cooling	(K)	4
Vapor Superheat	(K)	5
Air inlet (dry bulb)	(°C)	35
Water in/out temperatures	(°C)	12/7

Table 5: Brazed plate evaporator geometry

Fluid flow plate length	(mm)	600.0
Plate width W	(mm)	200.0
Corrugation type		Herringbone
Corrugation angle	(°)	60
Corrugation amplitude	(mm)	2.0
Corrugation pitch	(mm)	12.0

The results of the numerical analysis showing the TTP as a function of the circuit length are reported in figure 1. The TTP values are slightly lower for R32 which indicates a tendency of R32 to have higher heat transfer performance compared to R410A during the condensation under the same working conditions. Furthermore, the TTP curves are rather flat for circuits longer than 24 m and display the lowest values for both fluids. As a consequence a 27 m circuit length is chosen as an optimal for both R410A and R32.
Regarding the brazed plate heat exchanger design, it was decided to use a single heat exchanger as the evaporator to avoid problems of uneven two-phase fluid distribution and oil return. Due to a limited number of brazed plate types suitable for the 74 kW nominal cooling capacity; a brazed plate with 0.6 m height and aspect ratio 0.3 has been selected. The main geometrical characteristics of the brazed plate heat exchanger are listed in Table 5.The chilled water flows in counter-current to the refrigerant and the heat transfer area of each plate is divided in several small stretches. An iterative procedure conceptually similar to the finned coil condenser simulation is used. The correlations used for the refrigerant-side such as heat transfer coefficient, pressure drop, and void fraction are listed in Table 2.

Several simulations have been first run with R410Aand afterwards with R32 for a different number of plates at the nominal conditions as per table 4. Figure 2 shows the calculated evaporation temperature with different number of plates having a capacity of 75 kilowatt. It is important to highlight that the evaporation temperatures for R32 and R410A are similar. Furthermore, the heat transfer area increase has a positive effect, since it increases the evaporation temperature.
However, for a number of plates higher than 100, the evaporation temperature remains almost constant. This is a consequence of an increased number of passages on the water side; the water flow rate being constant. The water side heat transfer coefficient decreases because of the decreased mass flux. This negative contribution on the heat transfer process tends to counterbalance the positive effect of the heat transfer area increase, under the fixed cooling capacity hypothesis.

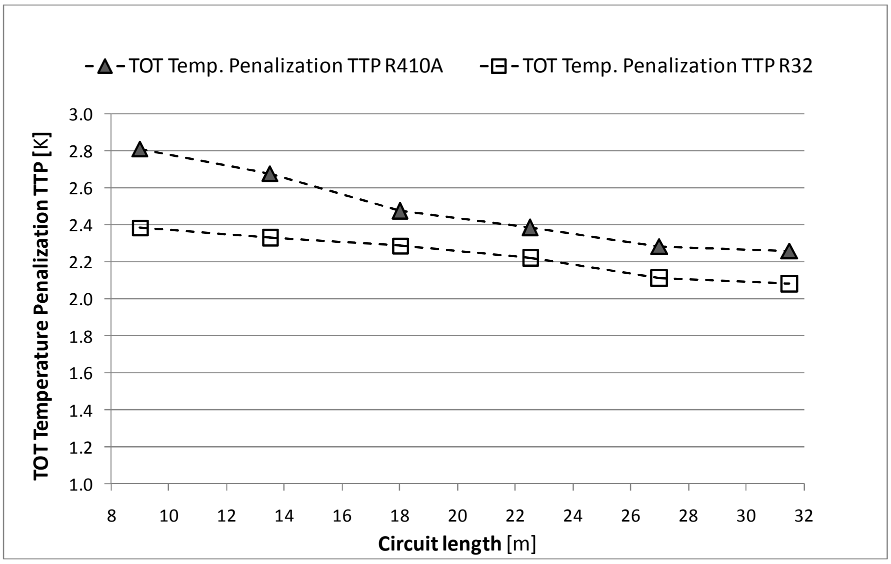

Figure 1: Total Temperature Penalization for the finned coil

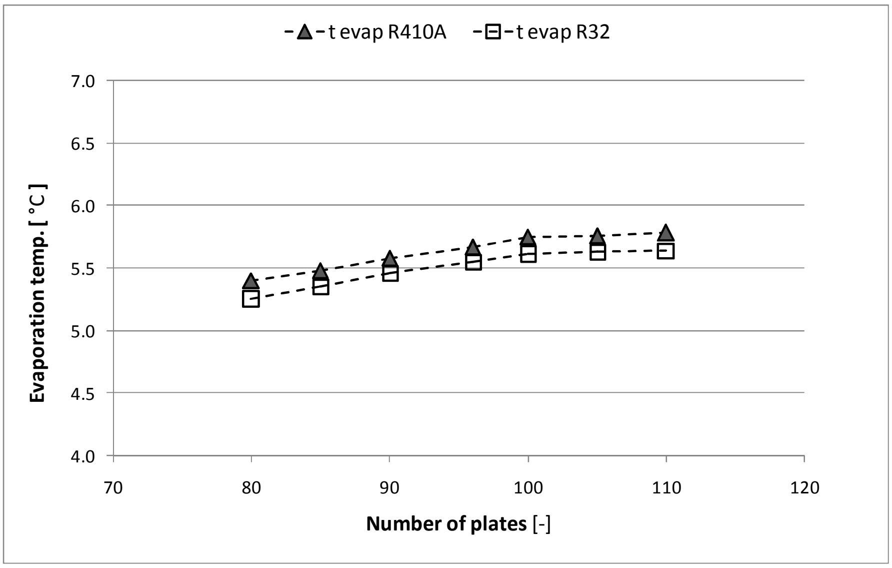

Figure 2: Effect of the number of plates on the evaporation temperature

As a consequence, a heat exchanger with 96 plates was identified as the optimal one for the evaporator. A larger number of plates could generate a poor distribution of the two-phase refrigerants over all the passages, without increasing the evaporation temperature. The same simulations with R32 confirmed that the same number of plates adopted for R410A is the optimal for R32.
One off-the-shelf scroll compressor designed for R410A having a displacement of 24.5 m^3/h at 50 Hz was experimentally analyzed by means of performance test rig. The test rig is designed in accordance with EN 13771-1 method D2 where the refrigerant flows is measured with a Coriolis mass flow meter at the discharge side. The uncertainty of the absorbed electrical power is within +/-1% and the expected uncertainties for the cooling capacities should be +/-2.5 % based on the accuracies of the mass flow meter, pressure and temperature sensors.

The compressor was tested within the operating application envelopes and polynomial correlations of mass flow and electrical power were generated as per EN12900 for both R410A and R32. These generated compressor performance data were used for chiller simulation with both fluids.
The performance of the compressors are reported in the table 6 at the nominal conditions reported in table 4.

Table 6: Compressor Performance at nominal condition reported in table 4

	VCC	COP	Tdisch.
	kJ/m³	(-)	°C
R410A	5478	3.95	76
R32	5832	4.00	92
Variation	6%	1%	16

A simulation code of the whole chiller has been implemented based on the previously outlined simulation tools for the condenser, the evaporator and the generated compressor performance data. A flow diagram of the code is shown in figure 2.

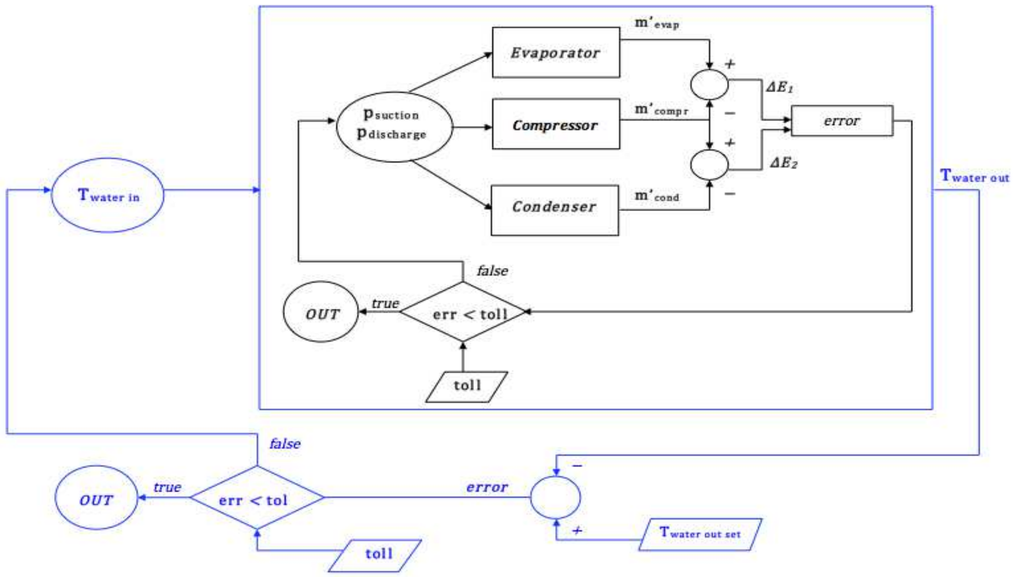

Figure 3: Simulation code flow chart

The inlet temperatures and mass flow rates of both air and water are the input parameters to the simulation programs well as the vapor superheat at the evaporator outlet. Starting from suction and discharge pressure initial guess, each component subroutine provides a refrigerant mass flow rate of the condenser, the evaporator and the compressor. When these mass flow differences are higher than the set tolerance (fixed by the user),the suction and discharge pressures are varied until the components' mass flow rates converge within the set tolerance. An external iteration loop enables the set outlet water temperature to be obtained by varying its inlet temperature. The present simulation tool was validated on the experimental performance data available in Cecchinato and Chiarello 2010. That set of data was obtained from off-the-shelf chillers similar to the configuration analyzed in the present paper. The present code was demonstrated to be able to capture satisfactorily air conditioning performances and related indicators. Figure 4 reports the variation of the condensing temperature with condenser circuit length, whilst figure 5 reports the evaporating temperature variation. It is important to highlight the influence of the circuit length on the condensing temperature, as this decreases more than 1 K from a circuit of 9 m to a circuit of 27 m. Figure 6 shows the results of the simulation of the system at full load and the impact of the temperature variation on the system performance for both refrigerants R410A and R32. The evaporator plate heat exchanger (96 plates) and the compressor are the same for all conditions in the figures.
The system Energy Efficiency COP, also denoted as EER ratio in EN standards is reported in figure 7.It is important to point out the consequences of the non-optimal choice of the circuit length, especially in terms of EER (figure 5). The EER is mainly impacted by the condensation temperature changes (the evaporation average temperature being fairly constant) and the compressor performance at those condensation and evaporation temperatures. For both refrigerants the EER increases up to 7 %, and with R32 it is showing a slightly lower sensitivity to the circuit length variation.
In Table 7, the main parameters of the "real" cycle obtained under the working conditions in Table 4 (full load) for R410A and R32 are reported, for 27 m condenser circuit length, 96 plates for the evaporator.

When using the same compressors and with optimized heat exchangers design, R32 delivers some 5 % higher cooling capacity. The average saturation condensation temperature is identical for both fluids, while the evaporation saturation temperature is some 0.2 K lower for R32. The system EER is almost the same for both fluids. The discharge temperature for R32 is some 18 K higher than for R410A.

Table 7: R410A and R32 simulation results for the working conditions in Table 4

	t cond	t sub	t disch	t evap	Q evap	$\dot{m}_{water}$	Pcomp	EER
	(°C)	(°C)	(°C)	(°C)	(kW)	(kg/h)	(kW)	(-)
R410A	46.3	42.4	78.4	4.75	74.0	12024	19.6	3.773
R32	46.3	42.2	95.2	4.54	78.2	12744	20.7	3.785

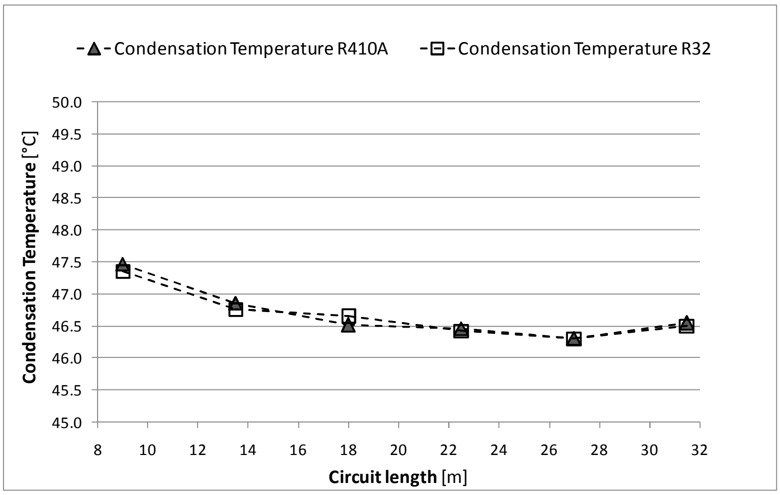

Figure 4: Condensation temperature as a function of finned coil condenser circuit length

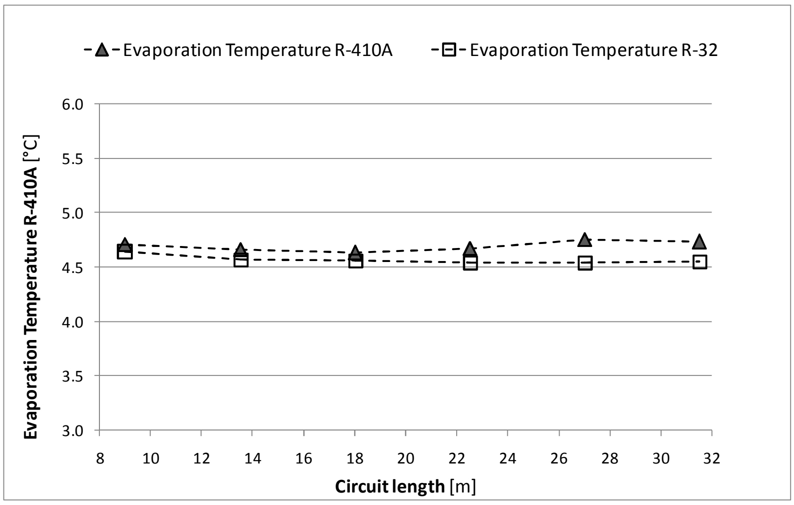

Figure 5: Evaporation temperature as a function of finned coil condenser circuit length

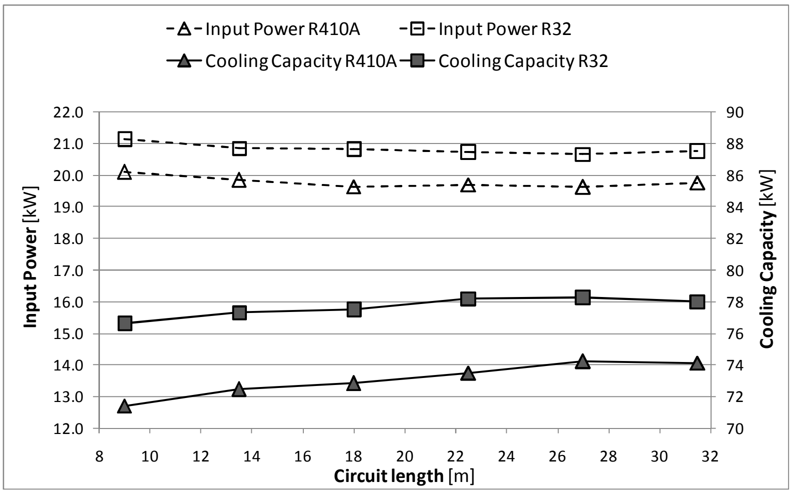

Figure 6: Input power and cooling capacity as a function of finned coil condenser circuit length

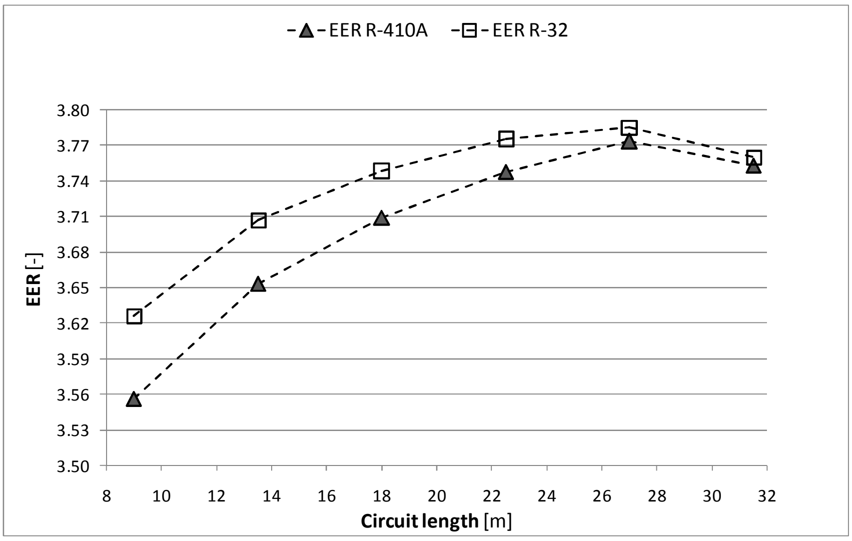

Figure 7: Energy efficiency ratio as a function of finned coil condenser circuit length

4. SEASONAL EFFICIENCY ANALYSIS.

Table 8 is reporting the main operating conditions for the seasonal efficiency according to the standard EN14825.The condenser circuit length for this analysis is 27 m according to the optimized TTP discussed previously, and the evaporator heat exchanger has 96 brazed plates.
Starting from the experimental cooling capacity and the input power (including auxiliaries) recordings during tests A, B, C, D tests, the EN 14825 fixes a calculation procedure for estimating the refrigeration unit performance with ambient temperatures from 17 °C up to 40 °C. The standard determines a “bin” of seasonal working hours per each different ambient temperature, with the aim of representing a typical European hot season climate.
According to the EN 14825 procedure, it is possible to evaluate the Seasonal Energy Efficiency Ratio SEER that is the reference annual cooling demand divided by the annual electric power consumption. The standard fixes correction factors for units with on/off modulation (i.e. fixed capacity compressors, like the ones considered in this paper).
In this work we simulated the working conditions A, B, C, D (Table 8) for the refrigeration unit under investigation. The results are reported in Table 9 for Tdesignc equal to 35°C. The cooling capacity of the chiller when operating with R32 is 5 to 6 % higher than with R410A, notwithstanding the evaporation and condensation temperatures of R32 and R410A being very close. The discharge temperature with R32 is 14.3 to 16.8 K higher. The EER of R32 is slightly better for cases A and B (i.e. at high ambient temperature), whereas R410A EER appears slightly higher for C and D (i.e. 25 °C and 20 °C ambient temperatures respectively).

Table 8: Test/simulation matrix according to EN 14825

	Part load ratio	Part load ratio	Outdoor heat exchanger Air dry bulb temperature	Indoor heat exchanger Fan coil application Inlet/outlet water temperature Fixed outlet
		%	°C	°C
A	(35-16)/(Tdesignc -16)	100	35	12/7
B	(30-16)/(Tdesignc -16)	74	30	[a]/7
C	(25-16)/(Tdesignc -16)	47	25	[a]/7
D	(20-16)/(Tdesignc -16)	21	20	[a]/7

[a] with the water flow rate as determined during "A"

Table 9: Simulation results for A, B, C, D case as in Table 6

		Tcond.	Tsub.	Tdisch.	Tevap.	Qevap.	EER
		(°C)	(°C)	(°C)	(°C)	(kW)	(-)
case A	R410A	46.30	42.4	78.40	4.75	74.00	3.773
	R32	46.30	42.2	95.20	4.54	78.20	3.785
case B	R410A	41.88	37.98	70.72	4.83	78.13	4.349
	R32	41.94	37.96	85.06	4.83	82.71	4.356
case C	R410A	37.40	33.38	63.41	4.92	82.38	5.015
	R32	37.46	33.43	78.48	4.87	87.16	4.993
case D	R410A	33.03	29.00	57.14	4.98	87.44	5.77
	R32	32.98	28.6	72.23	4.86	91.60	5.71

One should recall that the results reported in Table 9 can be considered as “rated” performances, at full load operation in the corresponding ambient working conditions. The EN 14825 fixes a procedure to evaluate the part load EER (EER_{PL}) for fixed capacity units, like the one under investigation.Tables10 and 11 report respectively the performances results for R410A and R32. The results show that under part load conditions, R32 EER outperforms R410A at lower ambient temperature.

Table 10: Simulation results for R410A at part load conditions according to EN 14825

Part Load Ratio (%)	**Cooling load demand** (kW)	**Declared capacity** (kW)	**EER at part load (-)**
100%	74	74.00	3.77
74%	57.81	78.13	4.17
47%	38.72	82.38	4.40
21%	18.36	87.44	3.95

Table 11: Simulation results for R32 at part load conditions according to EN 14825

Part Load Ratio (%)	**Cooling load demand** (kW)	**Declared capacity** (kW)	**EER at part load (-)**
100%	74	78.2	3.76*
74%	57.81	82.71	4.18
47%	38.72	87.16	4.44
21%	18.36	91.6	4.08

* not the same value as in Table 7, because of different declared capacity.

The effect of changing the circuit length of the condenser on the SEER is reported in figure 8. The graph indicates that the two refrigerants have very similar SEERon curve shape. One should consider that according to EN 14825, the number of working hours per season at higher ambient temperature is relatively low (e.g. at 35 °C, the number of working hours is only 13, whereas at 20 °C ambient temperature, it is 225).
Since the EER_{PL} for R32 is higher than R410A values especially considering the larger number of working hours, the seasonal EER (SEERon) is 2 to 3 % better for R32 compared to R410A. Both results for EER and SEERon reported

respectively in figure 7 and figure 8 confirm the suitability of the simplified TTP approach to design the condenser circuit length. The circuit length that minimizes the TTP is also the one leading to the maximum EER and SEERon.

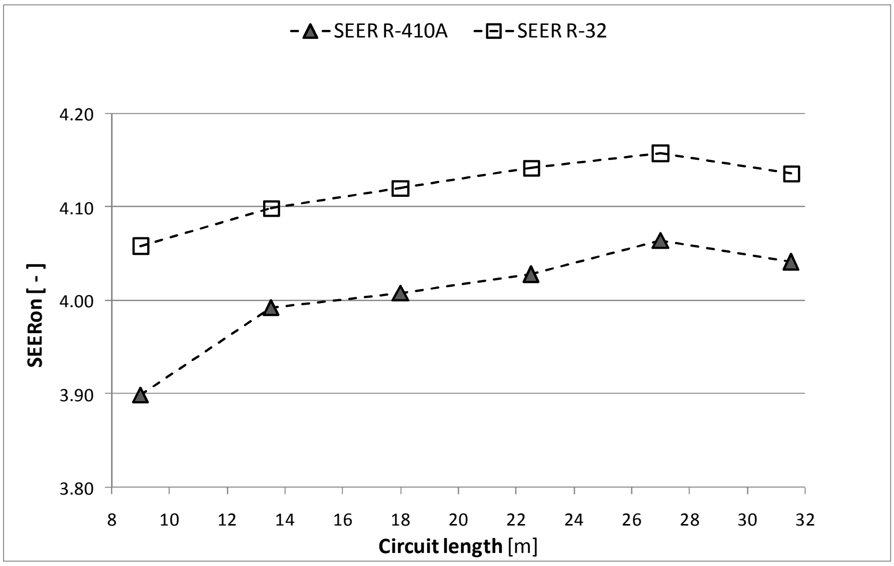

Figure 8: SEERon function of finned coil condenser circuit length

CONCLUSIONS

The performances of R410A and R32 in a packaged air cooled water chiller were compared by means of a detailed simulation model. The scroll compressor was experimentally characterized by means of calorimetric tests. The Total Temperature Penalization (TTP) performance evaluation criteria have been used to optimize the finned coil condenser circuit length. The brazed plates heat exchanger evaporator design was also numerically optimized.

The simplified approach based on the TTP was verified to be a suitable tool to optimize the design of the finned coil condenser circuits.

R32 was demonstrated to outperform R410A cooling capacity by some 5.7 % with 35°C ambient air dry bulb temperature and water temperatures 7 °C and 12 °C outlet and inlet temperatures, respectively. The EER was almost the same for both refrigerants while the SEER according to EN 14825 is 2 to 3% higher for R32, when using the same heat exchangers.

NOMENCLATURE

COP	Coefficient of Performance	(–)
EER	Energy Efficiency Ratio	(–)
GWP	Global Warming Potential	
$\dot{m}$	refrigerant mass flow	(kg/h)
P	input power	(kW)
Q	heat flow rate	(kW)
SEER	Seasonal Energy Efficiency Ratio	(–)
Tdesignc	Reference Design Temperature Conditions for cooling	(°C)
T	Temperature	(°C)
VCC	Volumetric cooling capacity	(J/m^3)

Subscript

comp	compressor
cond	condensation
disch	discharge
evap	evaporation
PL	part load
sub	subcooling

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