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Comparative study of the performance of a heat pump for heating with R410A and R32 as refrigerant, using the software IMST-ART

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ABSTRACT

According to the European Commission, at this moment, heating and cooling in buildings account for half of the energy consumption, while only a 16% of the energy used in this issue is obtained employing renewable sources.

One of the possible solutions to improve the current situation is the use of heat pumps for the air conditioning of buildings. However, the refrigerants used at this moment are hydrofluorocarbons (HFCs) and they have a global warming potential (GWP) elevated so that new refrigerants which can substitute the current ones are being searched.

The most used refrigerant for heating applications is the R410A, which is a mixture of HFCs. Its immediate retrofit is one of its components, the R32, due to the compatibility of both refrigerants with the materials and its similar performance. Furthermore, the R32 has a value of GWP approximately of a third of the R410's value.

The objective of this work is to evaluate the main differences between the results obtained with both refrigerants and to find out the main changes to be done to obtain at least the same performance.

In this work, a model of IMST-ART of an air-water heat pump is presented using both refrigerants. In the model, both refrigerants are evaluated for different conditions of heating at low temperature. Lately, a summary of changes is proposed in order to cover the same functions, or more, than the original heat pump.

Keywords: *Heat pump; refrigerant; drop-in; R410A; R32; IMST-ART.*

1. Introduction

Currently, the 79% of the energy consumed in buildings in Europe corresponds to heating, cooling and domestic hot water production [1]. Additionally, the 84% of the production of this energy is done using non-renewable sources [2]. In new constructions the trend is to minimise the energy for heating and cooling requirements, but in the buildings already constructed, the tendency is to acquire a renewable source. There are several options such as biomass boiler, solar thermal and heat pumps.

In the last years, the refrigerants used for domestic heating and cooling using heat pumps are HFCs which have a value of GWP elevated.

From European directives, there are some regulations as the F-Gas regulation that predict or force a reduction of the commercialisation of these fluorinated gases and its use, making a progressive substitution of the current refrigerants to others with a reduced value of GWP. Is due to that reason

that we are in process of changing the refrigerant used, and there is enough uncertainty about which refrigerant will replace the current HFCs.
Trying to solve this question, there are some studies which assume that there is no refrigerant with good performance, non-pollutant, nontoxic, inflammable, and cheap [3]. To go deeper, the only pure subcritical refrigerants which can substitute the current HFC are at least slightly flammable [4].
In this work it is presented a study of a drop-in from a heat pump using R410A to R32. The model is done using a real heat pump model in the software IMST-ART [5].

2. Methodology

This study shows a comparison between the refrigerant most commonly used in heat pumps for domestic heating, R410A, and one of its components, which is thought to be the easiest drop-in, not only due to the similarities in their thermodynamic properties but also, their compatibility with materials and substances, such as the oil of the compressor.
This study is motivated by the high value of GWP of the current refrigerant, the heat pumps using this refrigerant is going to be banned.
The model represents a vapour compression cycle of a real heat pump used for domestic heating at low temperature.

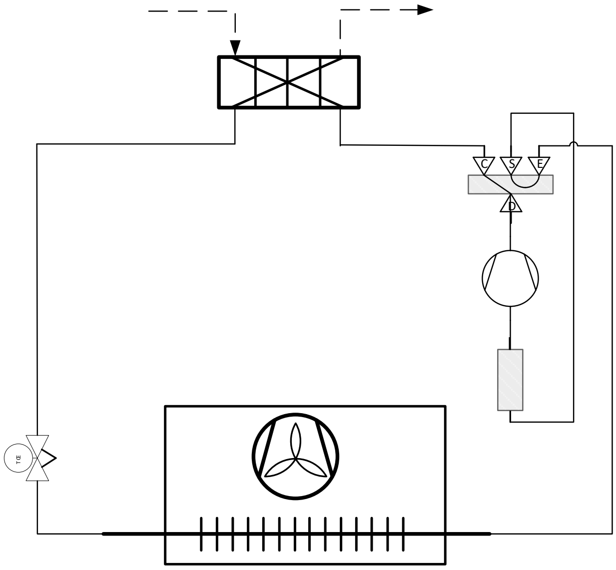

Figure 1: Refrigerant circuit of the model.

The circuit of the heat pump is a simple circuit with a suction accumulator, used to reduce the superheat and, therefore, the discharge temperature, and manage the different refrigerant needs depending on the conditions The heat pump has a capacity of 9kW on the design point of an intermediate climate and the main characteristic of its components are showed in Table 1, Table 2 and Table 3:

	Displacement (cm3)	**Type**
Compressor	17.2	Rotary

Table 1:Characteristics of the compressor.

	Width (mm)	**Height (mm)**	**Port Diameter (mm)**	**Plate Pitch (mm)**	**Number of plates**	**Flow type**
BPHE	72	329	24	1.1	34	Counter-current

Table 2: Characteristics of the BPHE (condenser).

	Tube diameter (mm)	Tube thickness (mm)	Inner surface	Transversal spacing (mm)	Longit. spacing	N rows	N. circuits
FTHE	7	0.43	Riffled	25	19	3	8

Table 3: Characteristics of the FTHE (evaporator).

The characteristics of the refrigerants are:

Refrigerant	Safety class	GWP_{100}*	Critical temperature	Critical pressure	Heaing volumetric capacity** (kJ/kg)
R410A	A1	2087	71.3	4901	203.96
R32	A2L	675	78.1	5782	304.18

Table 4: Characteristics of the refrigerants.

* GWP_{100} source: Regulation (EU) no 517/2014 of the European Parliament and of the Council [6]
**Calculated with REFPROP [7] for condensation temperature of 35ºC, evaporation temperature of 0ºC, 5K of subcooling and 3K of superheat and isentropic efficiency of 100%.

As it can be observed in Table *4*, the R32 is being used as a direct substitute of the R410A, because, while the thermodynamic properties are being conserved, the GWP is considerably lower in the case of the R32.
The model calculates the behaviour of a real heat pump, but some considerations have been considered in order to simplify the calculations:

- No pressure drops in pipes.
- Perfect distribution in heat exchangers.

In the model, the comparison which has been performed is a completely characterisation of the heat pump for heating at low temperature, for each refrigerant. The characterisation follows the process described in the standards UNE-EN14511[8] and UNE-EN14825[9], where the first step is to calculate the water mass flow and the air mass flow in standard conditions, both at the nominal point A7W35 with a ΔT in the water flow of 5K and a compressor speed of 60Hz. Then the mass flow of both secondary fluids has been put as an entry of the rest of the tests. Then the characterisation is done doing the following tests (Table *5*) with its partial load conditions which follow the equation (1)

$$Partial\ Load\ Factor = \left(\frac{T_{air} - T_{lim}}{T_{design} - T_{lim}}\right) \tag{1}$$

Where:
T_{lim}=16ºC. It is the limit temperature above which heating is not required.
Tdesign= -10ºC. It is the air temperature at which the heat pump is working at full partial load factor.

TEST	PARTIAL LOAD FACTOR
A-10W35	100%
A-7W34	88%
A2W30	57%
A7W27	35%
A12W24	15%

Table 5: Partial load conditions.

After knowing the mass flows, the partial load test campaign can be calculated. It is assumed that the bivalent temperature is -7ºC. Thus, the first test is A-7W34 and all the heating capacities are calculated using it as a reference.
With the results of each test and the number of hours of every condition, according to the standard UNE EN 14825[9], the seasonal COP is obtained. It must be considered that in the conditions where the test cannot be performed as stationary, it has been assumed a degradation of the performance considering the wall temperature and the mass flow of the water condensed from the air.

3. Results

The first comparative has been done between both refrigerants in nominal conditions.
The first variable observed is the heating capacity where the R32 case has a bit less than 10% more heating capacity. This heating capacity increase is similar to the volumetric capacity difference, due to the fact that the working conditions are very similar. This difference results in a slightly higher design capacity after the drop-in, without changing any component, and, for the partial loads campaign test, the air and water mass flows are going to be a little higher (1160 kg/h vs 1080 kg/h).
Once obtained the mass flows, the next step is obtaining the heating capacity of the bivalent point. In this case the assumption is that the bivalent point is -7ºC, and it corresponds with 88% of the partial loads. Therefore, the heating capacities of the different conditions must be near the values shown in the Table *6*.

Condition	**Partial Load (%)**	**Heating Capacity**	
		R410A (W)	**R32 (W)**
A-10W35	100	8.91	9.67
A-7W34	88	7.84	8.51
A2W30	54	4.81	5.22
A7W27	35	2.74	2.98
A12W24	15	0.72	0.78

Table 6: Partial load conditions and heating capacities.

With these requirements, the heating capacity of each test condition is obtained by varying the compressor speed between the values which have been considered as the minimum and the maximum compressor speed i.e. 30rps and 120rps respectively. In Table *7* the results of both partial load campaigns are represented.

	Compressor speed (rps)	**Heating Capacity Partial Loads (W)**		**Heating Capacity (W)**		**COP**	
		R410A	**R32**	**R410A**	**R32**	**R410A**	**R32**
A-10W35	120	8909	9670	7199	7837	2.71	2.74
A-7W34	120	7840	8510	7838	8509	2.98	3.01
A2W30	58	4811	5222	4776	5260	4.91	4.95
A7W27	30	2744	2979	3437	3644	5.87	5.84
A12W24	30	722	783	4070	4272	8.09	8.01

Table 7: Heating capacity and COP on each condition.

As it can be seen, the demand curve (partial loads) is fitted from the bivalent point until almost 7ºC of air temperature. It means that even though the compressor has installed an inverter, at ambient temperatures above 7ºC, the heat pump would need to work being switched on and off. The load and capacity curves of the heat pump can be seen in the Figure *2*.

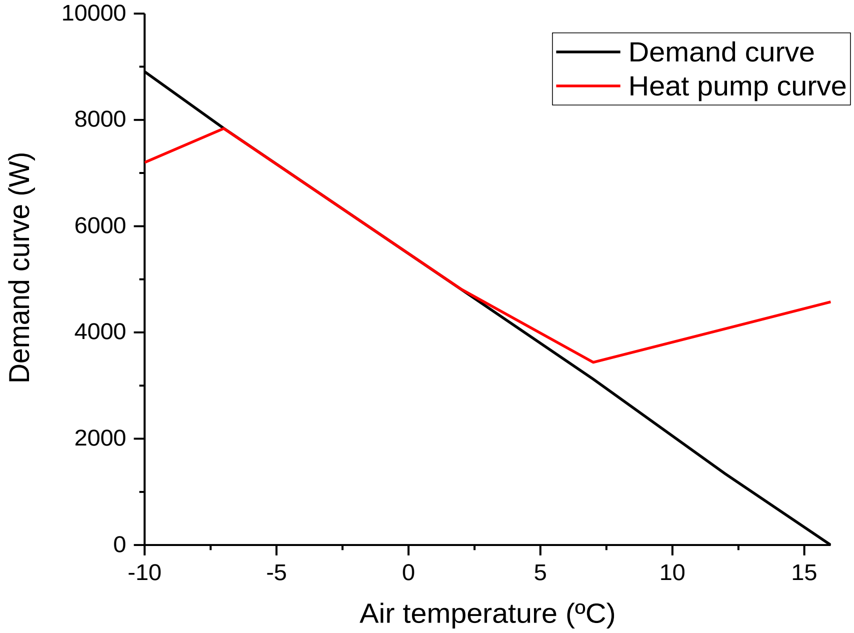

Figure 2: Demand curve and heat pump curve depending on the air temperature.

3.1 Main thermodynamic properties

Regarding the different discharge temperatures, there is a point that cannot be reached using the R32 as refrigerant with the current model of heat pump, which is the point of A-10W35. In this point the discharge temperature is above 120ºC, and in the envelope graphs of the compressor, the maximum temperature allowed is 110ºC. So, the easiest way to compare both heat pumps is either switching off the heat pump and connecting an electric resistance or connecting a resistance in the evaporator to increase the air temperature before the evaporator. Both considerations have been considered and the temperature of the air must be -6ºC to produce hot water at 35ºC. The discharge temperatures of both cases can be observed in the Figure *3*:

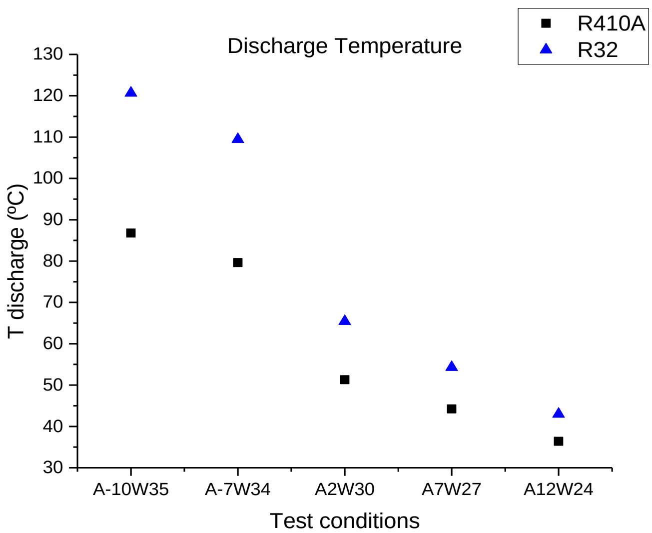

Figure 3: Discharge temperatures of the tests.

As it is known, the discharge temperature is considerably higher in the R32 case, and it could be very difficult to do this drop-in for domestic heating at high temperature. However, due to this superior discharge temperature, as presented in the Figure *4*, this fact induces to condensation temperature reduction due to the pinch point occurrence.

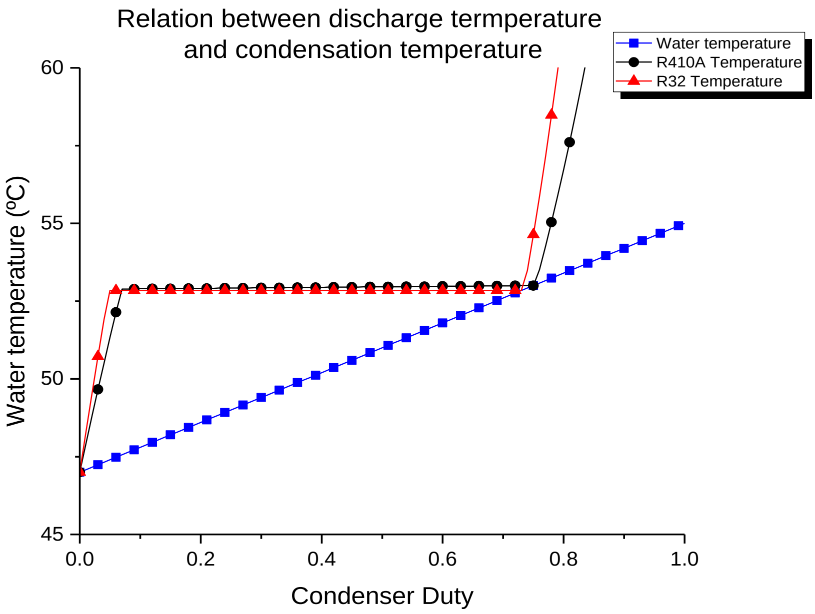

Figure 4: Relation between discharge temperature and condensation temperature.

Therefore, comparing the condensation temperatures, in all conditions, the condensation temperature of the R32 case is on a small scale lower than the condensation temperatures of the R410A case. This difference can be observed in the Figure *5*

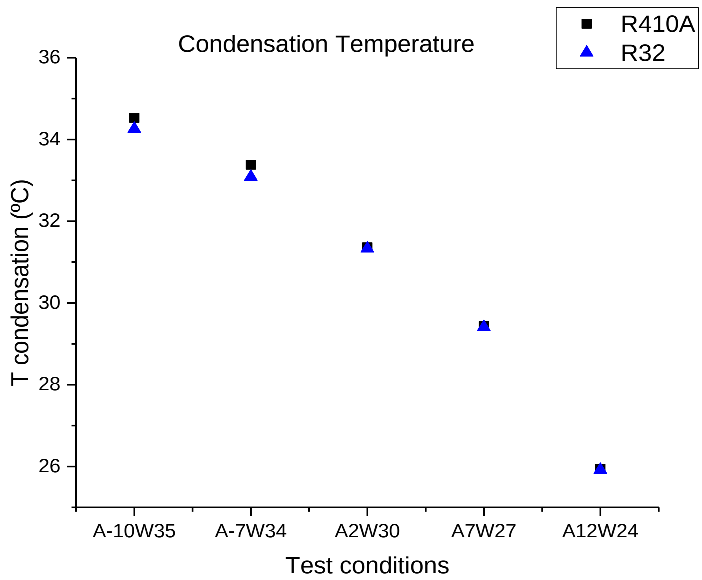

Figure 5: Condensation temperatures of the different tests.

Seeing then the evaporation temperature, Figure *6*, the evaporation temperature for the R32 case is slightly lower in all the conditions, which makes the wall temperature be lower. Additionally, the condensates mass flow is higher in the R32 case Therefore, the defrost periods are going to be longer and more power consuming for the R32 than for the R410A. Thus, the degradation of performance due to the frost formed in the evaporator must be higher in R32 case.

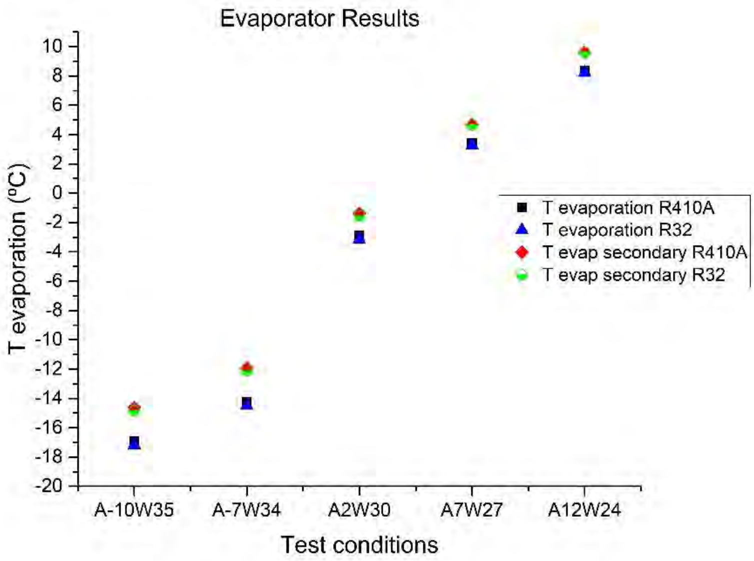

Figure 6: Evaporation temperatures of the tests.

3.2 Partial loads and energy efficiency

After the study of each relevant thermodynamic variable, a global performance study has been done. As mentioned before, the R32 has a slightly higher heating capacity and COP but by contrast, there is one partial load test that cannot be performed without being out from the envelope of the compressor. In this point is needed a resistance to heat the air before entering the evaporator.

After the correction the partial load test results are showed in Table *8* and Table *9* for both refrigerants.

	Compressor speed (rps)	Partial Load Needs (W)	Heating Capacity (W)	COP HP	Global Power Input HP (W)	Power Input Auxiliar (W)	COP bin
A-10W35	120	9670	8743	2.4	2894	4222	1.23
A-7W34	120	8510	8509	3.01	2829	1.4	3.01
A2W30	58	5222	5260	4.95	1170	0	4.50
A7W27	30	2979	3644	5.84	624	0	5.84
A12W24	30	783	4272	8.01	533	0	8.01

Table 8: Results of the campaign of R32.

	Compressor speed (rps)	Partial Load Needs (W)	Heating Capacity (W)	COP HP	Global Power Input HP (W)	COP bin
A-10W35	120	8909	7199	2.71	2654	2.71
A-7W34	120	7840	7838	2.98	2634	2.98
A2W30	58	4811	4776	4.91	1069	4.47
A7W27	30	2744	3437	5.87	585	5.87
A12W24	30	722	4070	8.09	503	8.09

Table 9: Results of the campaign of R410A.

With these values and the number of hours of each temperature in an average climate, can be obtained the seasonal COP.

SCOP	4.75
ηs	1.898

Figure 7: SCOP and efficiency of the heat pump using R410A.

	Etiquetado
A+++	X
A++	
A+	
A	
B	
C	

Figure 8: Energy labelling for the heat pump using R410A.

SCOP	4.54
ηs	1.814

Figure 9: SCOP and efficiency of the heat pump using R32.

	Etiquetado
A+++	X
A++	
A+	
A	
B	
C	

Figure 10: Energy labelling for the heat pump using R32.

As it can be seen, the problem with very low ambient temperatures does not affect massively to the global performance, as the number of hours at year working at those temperatures is low. The difference between both COP is around 0.2 where the design capacity has increased a little (650W).

4. Discussion

To conclude, it has been proved that the drop-in in a heat pump for domestic heating at low temperatures, from R410A as refrigerant to R32 can be directly done, even without doing a retro-fit.

For average climates, this kind of heat pumps can be used with R32 without several problems but for heating at high temperatures. Additional measures must be considered to decrease the discharge temperature of the compressor.

It should be mentioned that the value of GWP of R32 case is still elevated (675[6]) and this drop-in could be a short-term solution of the current heat pumps, but for the long-term strategy the value of the GWP of this refrigerant should not be low enough to be a future solution.

5. Acknowledgements

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REFERENCES

[1] COMMISSION, T H E E. *6.9.2013*. 2013, (812), 1–82.

[2] COMMISSION, E. *Heating and cooling*. Available at: https://ec.europa.eu/energy/en/topics/energy-efficiency/heating-and-cooling

[3] KUJAK, S; SCHULTZ, K. *Insights into the next generation HVAC&R refrigerant future*. *Science and Technology for the Built Environment*. 2016, **22**(8), 1226–1237.

[4] MCLINDEN, M O; BROWN, J S; BRIGNOLI, R; KAZAKOV, A F; DOMANSKI, P A. *Limited options for low-global-warming-potential refrigerants*. *Nature Communications*. 2017, **8**, 1–9.

[5] IIMST-ART V3.9. *Simulation Tool to Assist the Selection, Design and Optimization of Refrigeration Equipment and Components*. B.m.: Universitat Politècnica de Valéncia, Instituto de IngenieríaEnergética, Spain (2010). 2018

[6] PARLAMENTO EUROPEO. *Reglamento (UE) nº 517/2014 sobre gases fluorados de efecto invernadero*. 2014, **2014**, 195–230.

[7] LEMMON, E W; MCLINDEN, O M. *NIST Reference Fluid Thermodynamic and Transport Properties — REFPROP*. *Database*. 2002.

[8] EN, N E; UNE-EN, N. *UNE-EN_14511-3=2014*. 2014.

[9] EN, N E; UNE-EN, N. *UNE-EN_14825=2016*. 2016.