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ABSTRACT

One of the main objectives of the European Commission in the buildings sector, responsible for approximately 40% of total energy consumption and 36% of greenhouse gas emissions, is to identify technological solutions capable of reducing energy consumption and at the same time greenhouse gas emissions. For this purpose, ground source heat pump system (GSHPs) is a technology of particular interest that promises to considerably reduce greenhouse gas emissions of HVAC systems (up to 44% compared to air source heat pumps). In order to develop and test innovative GSHPs to be used for heating and cooling in the various European climatic zones, EU has funded the GEO4CIVHIC project, which will have a duration of 4 years and will end in 2022. As part of the project, the problem of identifying new generation low environmental impact refrigerants to be used in innovative GSHPs is tackled. In this article, we report the results of an energetic and exergetic analysis of the performance of heat pumps based on simulations carried out both on simple reverse cycles and on more complex cycles. Low pressure alternative fluids have been considered as an alternative to R134a and high pressure fluids as an alternative to R410A. The simulations were conducted at various heat sink and heat source temperature conditions, in order to evaluate the GSHPs performance

in the whole range of real conditions that can be found in Europe. Particular attention was paid to the compression phase, with the aim to simulate the compressor performance in a more realistic way than simply assuming constant isentropic efficiency.

1. INTRODUCTION

The retrofit of existing building is a relevant topic within the objective to improve the energy efficiency of the building stock. The Energy Transition initiative of the European Commission has the aim to increase the market of the building stock retrofitting from 1% to 3%, shifting the nature of the interventions from shallow towards deep retrofits. Funded within this initiative, the Geo4Civhic EU project (Most Easy, Efficient and Low Cost Geothermal Systems for Retrofitting Civil and Historical Buildings) has the objective to overcome the present barriers to the application of geothermal heat pumps for building retrofitting. The main goals of the project are developing and demonstrating easier to install and more efficient ground source heat exchangers and developing or adapting heat pumps and other hybrid solutions for retrofits. In particular, one of the main objectives regarding ground source heat pumps for domestic applications is the substitution of the present high GWP refrigerants (mainly R134a and R410A) with refrigerants characterised by medium or low GWP.

The overall number of heat pumps installed across Europe in 2018 was almost 12 million, the majority in Italy and France (Nowak *et al.* (2018)). Despite this, gas boilers still dominate the market, with 90% of share, threatening the sustainability target of Paris Agreement (United Nations, 2015). EU Directives 2010/31/EU and 2018/844 consider heat pump technology as a relevant solution in order to integrate renewable energies for satisfying heating and cooling demand. At the same time, the potential of ground-source heat pumps is still underestimated, with the greatest market share hold by air-to-air or air-to -water heat pumps (Sanner *et al.* (2013)). As refer the working fluid, hydrofluorocarbons (HFCs) dominated the refrigerant market in the last twenty years. Most HFCs are characterized by relatively high GWP and therefore they are subject to the restrictions imposed by the EU Regulation 517/2014, which enforces an ongoing annual reduction of new HFCs quantities that can be introduced to the market by each producer or importer, leading to declining HFCs availability every subsequent year. As a consequence, HFCs price will continue to increase, urging the industry to find economically and environmentally sustainable alternatives with low GWP (Mota-Babiloni *et al.* (2016)). At present, R134a and R410A are the most used refrigerants in heat pumps and chillers sector, with a GWP of 1430 and 2088 respectively. Their replacement is a research challenge: several criteria have to be met to prove that a new fluid can be the proper substitute for R134a or R410A, such as suitable thermodynamic properties, stability in the system, low flammability and toxicity (Bobbo *et al.* (2018)). In particular, it must be highlighted that the simple substitution of a high GWP refrigerant with a low GWP fluid could not be enough to reduce the system overall environmental impact, if the substitution leads to lower energetic performance of the cycle, increasing indirect emissions (Makhnatch *et al.* (2014)). Mota-Babiloni *et al.* (2015) analysed different HFC/HFO mixtures and showed good results for not standardized mixtures N-13, XP-10, and ARM-42A as substitutes for R134a, and L41 and DR-5 as alternatives to R410A. Bobbo *et al.* (2019) and Bertsch *et al.* (2008) investigated the influence of different refrigeration cycle configurations compared to the basic cycle for various refrigerants.

In this paper, the results of an energetic analysis comparing short-term (GWP<1000) alternatives for R410A and both short-term and medium-term (GWP<150) alternatives for R134a are reported. Moreover, several refrigerating cycles were considered extending the set of cycles simulated in Bobbo *et al.* (2019), in order to improve the performance of low-GWP refrigerants. Aim of the analysis is to establish the most suitable refrigerants to employ as working fluids in some demo facilities under development within the Geo4Civhic project.

2. METHODOLOGICAL APPROACH

A set of low-GWP fluids was analyzed as possible substitutes for R134a and R410A in ground-source heat pumps. For this purpose, computer simulations were carried out with a software developed in Matlab environment (Matlab (2020), modelling various refrigerating cycles and implementing thermodynamic properties of refrigerants through Refprop 10 database (Lemmon et al. (2018)). Two different sets of substitutes for R410A and R134a were identified, based on similarities in thermodynamic properties such as critical temperature (T_{crit}), critical pressure (P_{crit}) and normal boiling point temperature (NBP). Tables 1 and 2 show the considered fluids with some characteristic properties. Besides a) basic and b) regenerative cycles considered in Bobbo *et al.* (2019), four new cycles were analyzed and compared: c) inter-refrigerated compression cycle with vapor injection, d) intercooler cycle, e) economizer cycle and f) auxiliary compressor cycle with liquid-vapor separator (for pure fluid and

azeotropic mixtures only). Table 3 synthetically describes each cycle, with a scheme of the circuit and the thermodynamic cycle in (T,s) diagram.

Table 1: Substitutes for R134a.

Fluid	GWP	ASHRAE Safety Class	Composition wt(%)	Tcrit [C]	Pcrit [bar]	NBP [C]
R134a	1430	A1	-	101.06	40.59	-26.07
R1234yf	4	A2L	-	94.7	33.82	-29.49
R1234ze(E)	7	A2L	-	109.36	36.35	-18.79
R513A	631	A1	R134a/R1234yf (44/56)	94.91	36.48	-29.56
R515A	393	A1	R1234ze(E)/R227ea (88/12)	108.71	35.66	-18.74
R515B	299	A1	R1234ze(E)/R227ea (91.1/8.9)	108.89	35.84	-18.80
R516A	142	A2L	R134a/R1234yf/R152a (8.5/77.5/14)	97.17	36.54	-29.43

Table 2: Substitutes for R410A.

Fluid	GWP	ASHRAE Safety Class	Composition wt(%)	Tglide [C]	Tcrit [C]	Pcrit [bar]	NBP [C]
R410A	2088	A1	R32/R125 (50/50)	0.05	71.34	49.01	-51.44
R32	675	A2L	-	-	78.11	57.82	-51.65
R454B	466	A2L	R32/R1234yf (68.9/31.1)	1.5	78.10	52.67	-50.50

3. WORKING PARAMETERS AND BOUNDARY CONDITIONS

Each cycle was simulated in steady-state conditions. The analysis was performed under the following assumptions for the working parameters.

- heating power required by the user at the condenser: 10 kW.
- superheating at the evaporator outlet: 5K.
- subcooling at the condenser outlet: 6K.
- condenser and evaporator Pinch Point: 3K
- every flux in the heat exchangers is supposed to be counter-current.
- heat exchangers are supposed to be ideal: heat losses and pressure drops are neglected.

As boundary conditions, typical values for the ground source water temperature in different climatic and soil conditions in Europe have been considered. These values have been coupled to typical values for the water temperatures at the user for different space heating terminals (from radiant systems to fan coils), as here below described:

- Ground-source water or water / antifreeze mixture temperatures (evaporator inlet): from 0°C to 10°C, with a step of 2°C
- Water temperature difference between evaporator inlet and outlet: 3°C.
- User outlet temperatures: 30°C, 35°C, 45°C and 55°C
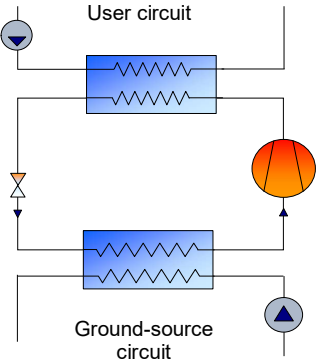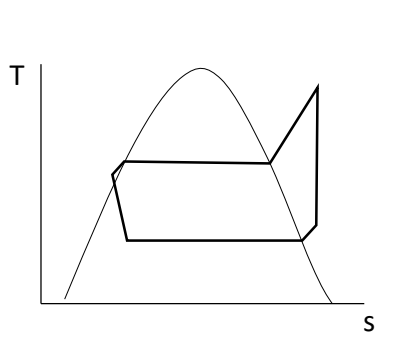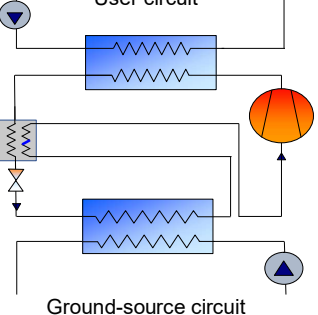- Water temperature difference between condenser inlet and outlet: 5°C.
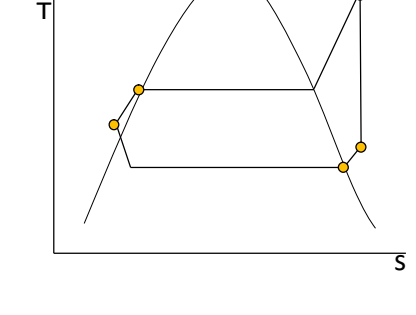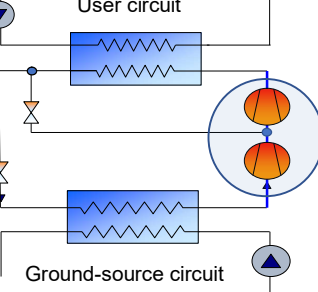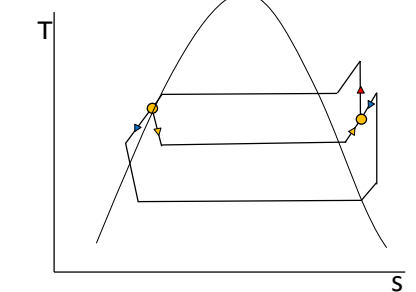
Compressors were modelled carefully due to their strong influence on the energetic analysis. In order to have reliable simulations of heat pumps, realistic data for compression work were taken from Bitzer software and used to calculate the compressors' performance. In particular, two compressor models were considered:

- Bitzer GSD60120VA scroll compressor for R410A and its alternatives;
- Bitzer 4DES-7Y reciprocating compressor for R134a and its alternatives.

For some of the fluids here considered, new on the marketplace, such as R515A, R515B and R516A, there were no data available about compressor performances. These mixtures are formed by 2 or 3 components, but one of them is predominant: R1234ze(E) for R515A (88% in mass fraction) and R515B (91% in mass fraction) and R1234yf (77.5% in mass fraction) for R516A. Thus, in these cases, the calculations to model the compressor were performed with the data of the main component of the mixture, i.e. R1234ze(E) data, were used for R515A and R515B, and

R1234yf data were used for R516A. Compressor data were treated in order to obtain the compressor performance as a function of pressure ratio.

Table 3: Cycles descriptions, simplified schemes and T-s diagram.

Cycle and features	Simplified scheme	T-s diagram
a) Base cycle		
b) Regenerative cycle An internal heat exchanger performs superheating, allowing to increase evaporating temperature.		
c) Inter-refrigerated compression cycle A vapour injection device is used to cool down refrigerant flow in order to increase compression isentropic efficiency. Intermediate pressure level and injection ratio have been optimized to maximize COP.		

Cycle and features	Simplified scheme	T-s diagram
d) Intercooler cycle A further heat exchanger is set between two compressors to cool down the working fluid stream. The rejected heat flux pre-heats the user mass flow rate before entering the condenser.		
e) Auxiliary compressor cycle with vapour-liquid separator A vapour-liquid separator (VLS) is used to reduce throttling losses. The saturated vapour phase stream is compressed by an auxiliary compressor working with a lower pressure lift, reducing the overall compression work. Intermediate pressure level was optimized. This cycle is used for pure fluid and azeotropic mixture only.		
f) Economizer cycle An internal heat exchanger replaces the VLS of the previous cycle in order to have the auxiliary compressor mass flow rate independent on intermediate pressure and to allow the usage with zeotropic mixtures too. Auxiliary compressor mass flow rate and intermediate pressure were optimized.		

4. RESULTS AND DISCUSSION

The simulated refrigerating cycles, studied for ground-source heat pump applications, were analyzed in terms of Coefficient of Performance (COP) and Volumetric Heating Effect (VHE). Performance indicators were calculated as in Bobbo *et al.* 2019 and their trends were recorded as a function of user and source temperatures. Fixing user temperature, a monotonous trend of performance indicators was noticed as a function of source temperature for each fluid and each cycle. Thus, performance hierarchy is noticed to be the same for each combination of user and source temperatures. For these reasons, a qualitative analysis can be inferred by Figures 1-2, where COP and VHE trends are shown, respectively, for R134a alternatives, in order to have a double comparison among cycles and refrigerants at fixed user and source temperatures. Figures 3-4 show analogous analysis for R410A alternatives.

Regenerative cycle (b) takes advantage, with respect to the base cycle, by using an internal heat exchanger to perform the superheating at the evaporator outlet, resulting in an improvement for both COP and VHE for all the considered fluids due to the increasing evaporating temperature. The average increase in terms of COP is 5% with respect to base cycle, while, in terms of VHE, the average increase is 7% for R134a alternatives and 6% for R32 and R410A. An exception is noticed for R454B, for which the increases are equal to 17%.

Inter-refrigerated compression cycle (c) results in lower COP with respect to the base cycle. An average decrease of 9% is noticed for both R134a and R410A alternatives. For all R134a alternatives a negligible difference in VHE is noticed, except for R134a and R1234yf for which an average 3% increase is detected. An average decrease of 5% is noticed for R410A alternatives. The worse performances of inter-refrigerated compression cycle (c) are probably caused by the vapor-injection mechanism, which increases isentropic efficiency of both stage compressors, but
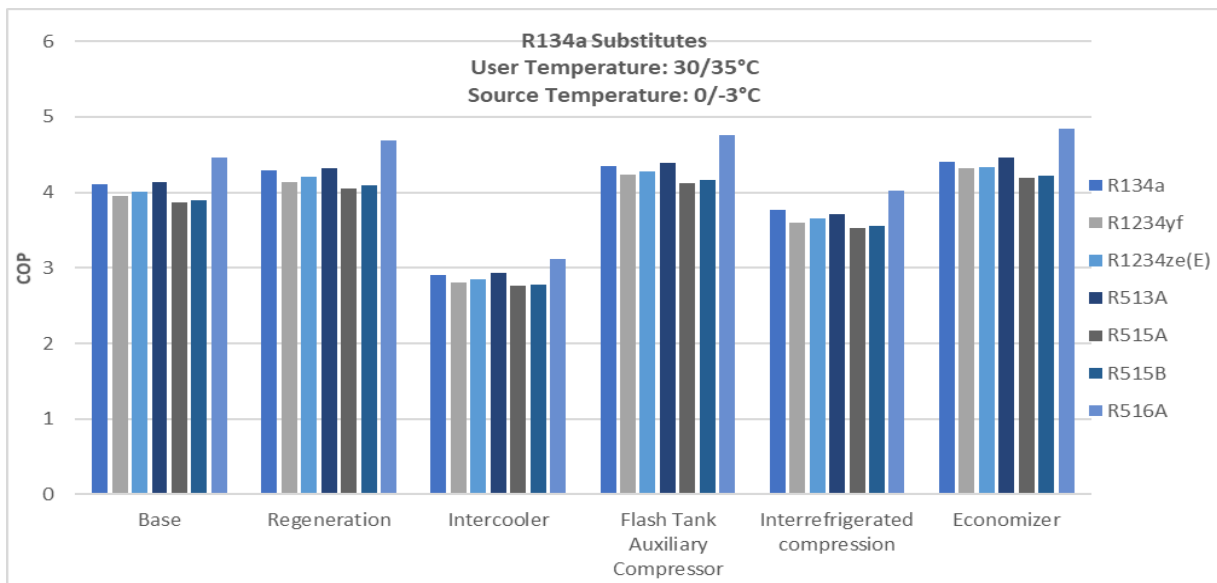

Figure 1: COP comparison among R134a substitutes and the whole set of considered cycles with fixed user and source temperatures.
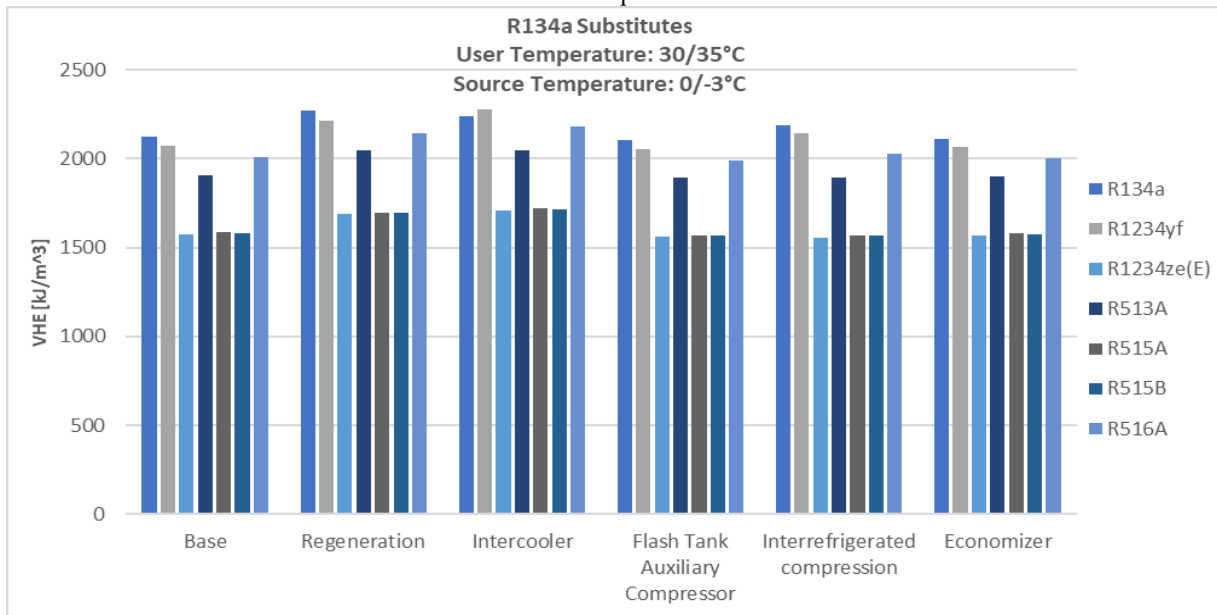

Figure 2: VHE comparison among R134a substitutes and the whole set of considered cycles with fixed user and source temperatures.

reduces the enthalpy difference at the condenser. The intercooler configuration (d) seems not to improve cycle performance in terms of COP with an average decrease of 25% among the whole set of fluids with respect to base cycle, meanwhile an average VHE increase of 5% only is observed for R134a alternatives; a light decrease in VHE is noticed for R410A alternatives. The worsening of the COP is probably caused by the increasing condensing temperature, due to the pre-heating made through the intercooler, and by a low difference between temperature levels of condenser and intercooler. This causes the worsening of isentropic efficiency at the second stage compressor, due to a very low pressure ratio.

Auxiliary compressor cycle with vapour-liquid separator (e) results in an average 6% increase of COP for R134a alternatives with respect to base cycle, while no sensible differences are noticed in terms of VHE (less than 1%). The higher COP values are due to the presence of the VLS, which reduces the mass flow rate through the main compressor and so the overall compression work.
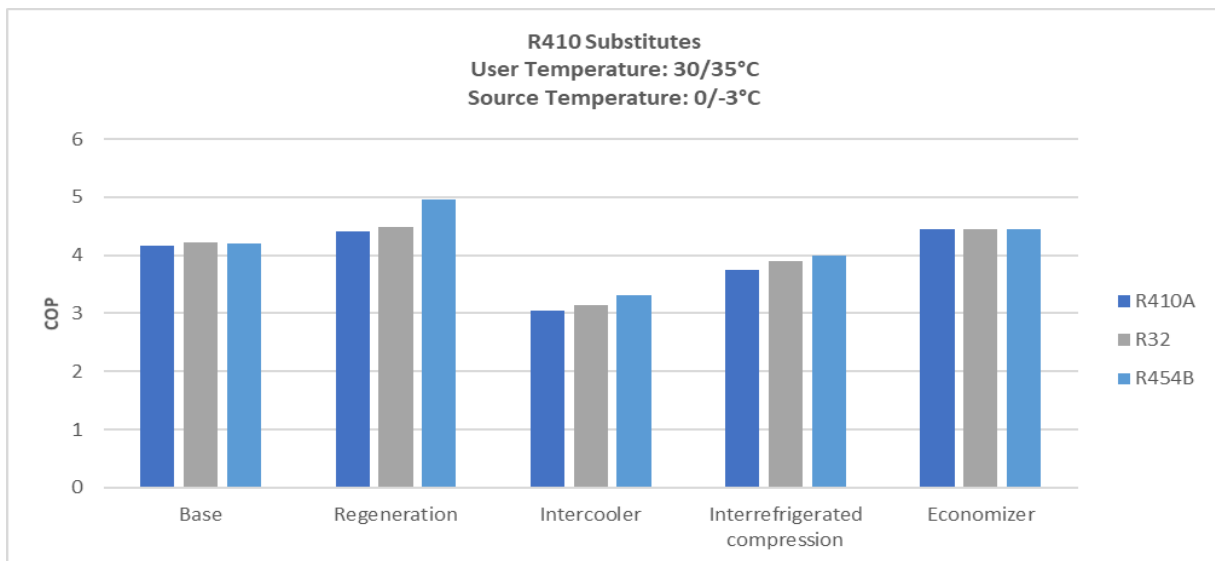

Figure 3: COP comparison among R410A substitutes and the whole set of considered cycles with fixed user and source temperatures.
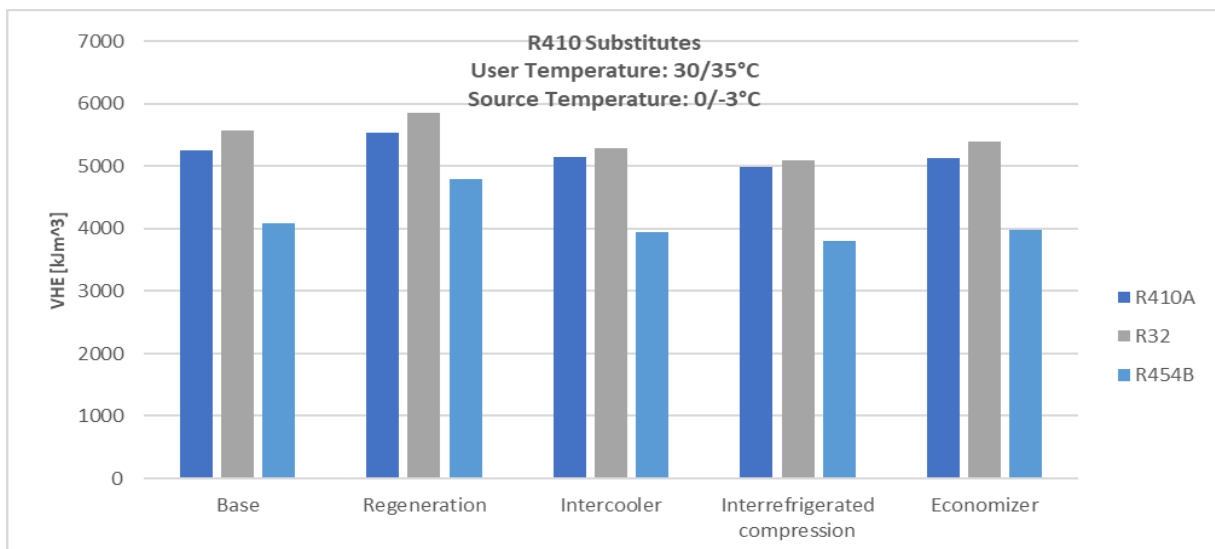

Figure 4: VHE comparison among R410A substitutes and the whole set of considered cycles with fixed user and source temperatures

The advantage of this is cycle is bounded by the dependence of auxiliary compressor mass flow rate on the

refrigerant enthalpy at the VLS inlet.

In the economizer cycle (f), an internal heat exchanger (the economizer) substitutes the VLS, so that the auxiliary compressor mass flow rate can be defined independently. This results in COP improvement for both R134a and R410A alternatives. The average increase in COP is 6% for R410A alternatives and 9% for R134a alternatives with respect to the base cycle. A light reduction of VHE is detected for R410A alternatives (2.5% average), while the detected reduction is negligible for R134a alternatives.

As far as fluids are concerned, R516A seems to be the most interesting alternative for R134a in terms of COP with an average increase of 8% among all cycles. In terms of VHE, R134a presents the highest values, followed by R1234yf, with an average reduction of 1.8%, and R516A, with an average reduction of 5%.

Instead, R410A, R32 and R454B present almost the same COP for base (a) and economizer (e) cycles, with differences lower than 1%. With inter-refrigerated compression (c) and intercooler (d) cycles, R454B presents the highest COP value, followed by R32 and R410A; the same hierarchy becomes more evident with regenerative cycle (b). Moreover, R454B presents a different cycles hierarchy with respect to the other fluids considered. For every combination of user and source temperature, regenerative cycle (b) results to be the most performing for R454B in terms of COP and VHE, while economizer cycle (f) gives the best performances for all other fluids (R134a alternatives included). As far as VHE is concerned, R32 results to be the most performing fluid, followed by R410A and R454B.

5. CONCLUSIONS

An energetic analysis was carried out to evaluate the thermodynamic performances of feasible substitutes for refrigerant R134a and R410A applied to ground-source heat pump technology. Several refrigerating cycles were studied and compared to the base cycle by simulations performed with a dedicated software developed in Matlab environment. R134a and R410A alternatives were required to have lower GWP and comparable cycle performances with respect to the corresponding substituted refrigerant, in terms of COP and VHE. Regeneration and economizer cycle were identified as the most performing cycles for the whole set of fluids considered. Moreover, refrigerant R516A results to be the most promising substitutes for R134a with an average COP increase of 8% and an average VHE decrease of 5% among all the considered cycles. R410A, R32 and R454B show similar COP values with base cycle and economizer cycle, while R454B shows the highest COP values with the other cycles. It is worth to underline the singularity presented by R454B with regenerative cycle. R454B shows an average improvement of 17% in terms of both COP and VHE with regenerative cycle. Such improvement may be attributed to the temperature glide presented by the mixture, taking into account that R454B is the only zeotropic mixture with a non-negligible temperature glide among the considered fluids. R32 results to be the most performing alternative for R410A in terms of VHE.

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