New generation refrigerants for domestic heat pumps in Sweden

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***Abstract*— The increasing concentration of greenhouse gases (GHG) in the atmosphere arising from human activities is unquestionably taking its toll on the environment with severe consequences that cannot be ignored. It is therefore necessary to take measures to avoid or reduce GHG production and release. In this scope, the 2015 F-gas regulation aims at limiting the contribution of the refrigeration industry to the global warming effect by phasing-down refrigerants with high global warming potential (GWP).**

This work contributes to this effort by analysing a number of refrigerants, i.e. R32, R152a, R290, R1270, R1234yf and R1234ze(E) that could substitute the current R410A in a 10 kW domestic heat pump typically used in a single family house in Sweden.

An EES model was thus created and the relevant outputs chosen to compare the refrigerants' options are the volumetric heating capacity, the coefficient of performance (COP), Seasonal coefficient of performance (SCOP) – obtained by the method found in the Standard BS EN 14825:2012, the discharge temperature, and the Total Equivalent Warming Impact (TEWI) factor.

The reductions observed for TEWI are remarkable. The overall lowest TEWI is obtained by refrigerants having the lowest GWPs, i.e. the HCs, of 54.2% and 54.6% for R1270 and R290 respectively. They are followed by R1234yf and R1234ze(E), with reductions of 52.5% and 53.7% respectively. R152a entails a reduction of 53.6%, due to its higher SCOP. R32 shows a reduction of 40.8%. Other parameters such as thermodynamic characteristics and safety issues are also considered during the analysis.

***Keywords*—CO_2, GHG, Heat pump, R410A, refrigerant, TEWI**

1. INTRODUCTION

CARBON dioxide (CO_2) emissions arising from human doings, such as power production from combustion of fossil fuels, industrial activities and – in the matter in question – refrigeration, have become more of a concern as a result of their growing scale in the past decades. The increasing concentration of greenhouse gases (GHG) in the atmosphere is taking its toll on the environment with severe consequences that cannot be ignored (Andres, 2012). It is therefore necessary to take measures to avoid or reduce GHG production and release. In this instance, heat pumps (HP) come forward as an increasingly important player. In fact, they generally entail a lower electricity consumption when compared to equivalent electric radiator systems, thus emitting less GHG. The magnitude of their benefit highly depends on the local energy mix used for electricity production (Forsén, 2005) and their impact is principally related to the refrigerant used, as it affects the CO_2 emissions of the system both directly and indirectly. As the number of installed heat pump systems increases in a high number of countries worldwide, the importance of operating with a low impacting refrigerant becomes crucial.

Since 1994, more than 6.7 million heat pumps were installed, which equals almost 224 GW of thermal capacity (ehpa, 2014). Globally, in 2013 the installed capacity was nearly 53 GWt, and it is projected to reach as high as 119 GWt by 2020 (Transparency market research, 2015).

Sweden is one of the countries with the highest amount of heat pumps installed. In 2008 already, a total of 700 000 units were installed, producing nearly 15 TWh of renewable energy per year (Forsén, Roots, & Bertenstam, 2008)

Nevertheless, fluorinated gases – which represent the biggest share of refrigerants currently in use – still represent 2% of the GHG emissions in Europe (European Commission , 2014). New limitations have thus been imposed through the implementation in January 2015 of the latest F-gas regulation, which mandates the cut of consumption of fluorinated gases by 79% by 2030. The schedule can be seen in Figure 1.

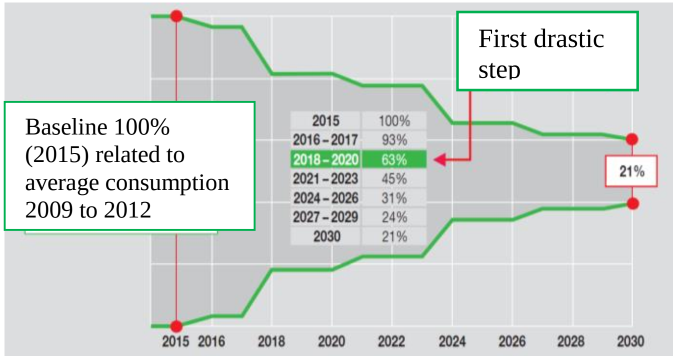

Figure 1. Phase-down of fluorinated gases' progress in time (Bitzer, 2014)

The scope of this thesis is thus to identify one or more valid replacement options to the refrigerant currently in use (R410A) in the product tested, namely the Diplomat Optimum 3G by Thermia. The main parameter evaluated is the overall environmental impact measured with the Total Equivalent Warming Impact (TEWI) factor. Other criteria have to be met as well, concerning stability in the system, thermodynamic properties, flammability and toxicity, all of which will be developed further on.

In order to accomplish the above-mentioned objective, a model has been created on EES reproducing the functioning of the studied HP, and validated against available experimental data and a second model created on IMST-ART.

The refrigerants have then been tested in such model, and results of environmental performance compared with the baseline R410A.

2. REFRIGERANTS

2.1. Characterisation of refrigerants

2.1.1. Thermodynamic characteristics

As far as thermal characteristics are concerned, low vapour heat capacity, low viscosity and high thermal conductivity should be maintained. The critical and boiling point temperatures must be chosen relatively to the application. Additionally, it would be preferable to work with a refrigerant having a small specific heat compared to the latent heat of vaporisation, in order to achieve smaller losses in the expansion valve (Rothlin, 2011).

2.1.2. Chemical characteristics

The working medium has to be stable and inert, though stability has to be reasonable – the refrigerant has to decompose in the atmosphere in a not too long time. Moreover, absence of flammability and toxicity are ideal, as well as negligible effects on the environment. Moisture levels of the system should be kept low, (1) in order to avoid corrosive compounds, and (2) to avoid freezing if present as free water. Stability relatively to the oil used in the HP has to be considered (Rothlin, 2011).

2.1.3. Safety

Concerning safety parameters, ASHRAE Standard 34 classifies refrigerants based on their toxicity and flammability levels. Specifically, classes A and B categorize the toxicity – namely class A identifies refrigerants for which toxicity has not been found at concentrations less than or equal to 400 ppm; class B those with evidence of toxicity at concentrations below 400 ppm. Groups 1, 2 and 3 identify a growing flammability at given conditions of temperature (21°C), pressure (101 kPa) and heat of combustion. In addition, a class named "A2L" has been introduced in order to keep into account those refrigerants with mild flammability but high ignition power and low burning (Institut International du Froid, 2011).

2.1.4. Environmental issues

The refrigerant R410A must be benign to the environment. Nowadays, this is true when no harm to the ozone is made, and there is no contribution to the greenhouse effect. Two indicators need thus consideration. Firstly, the Ozone Depletion Potential (ODP) is used to evaluate the impact of the substance on the ozone layer. The evaluation of the ODP is made by referring the effect of a substance to the one of R11, considered having ODP=1 (IPCC, 2005). Secondly, the greenhouse effect contribution is assessed by the Global Warming Potential (GWP). The evaluation is made by comparing the ability of absorbing heat by the gas, per unit of weight, relatively to CO_2, which is considered to have GWP=1 (Global Greenhouse Warming, 2015). For a complete evaluation of the environmental impact of the refrigerant, though, these two parameters are not sufficient. A more thorough analysis can be performed with a TEWI study (Equation (17)). This value comprehends the total impact of the substance throughout its whole lifetime, and considers not only the direct impact – due to leakages of the working fluid during the operation and dismissal of the machine – but also the indirect one, mainly due to the energy used for the functioning of the system, and is thus variable in accordance to the location and its energy sources' mix.

2.1.5. Policies

In order to reduce the environmental impact of human activities, a number of protocols and regulations have been prepared along the years; such as Kyoto protocol, Montreal protocol and lastly, aiming at reducing the CO_2 emissions by the refrigeration industry, and thus motivation behind this work, the "Regulation of the European Parliament and of the Council on fluorinated gases and repealing Regulation (EC) No 842/2006" or shortly, F-gas regulation. Its second version was implemented on 1st January 2015. Its main goal is to reduce the contribution of the refrigeration industry to the global warming, by cutting the consumption of fluorinated gases by 79% by 2030, with start at 2009-2012 average level (Bitzer, 2014). The expected outcome is a reduction of 1.5 Gtonnes of CO_2 equivalent by 2030 and 5 Gtonnes by 2050 from 2009-2012 average level (European Commission , 2014).

2.2. State of the art of last generation refrigerants

Generally, the literature available is rather limited, as some of the upcoming options are new substances and mixtures and still lack vast experimentation.

2.2.1. Hydrofluorocarbons (HFCs)

The GWP is still not optimal, and thus they should be regarded as medium term solutions.

Ho-Saeng Leea (2012) performed a study in a water source heat pump system on an R152a/R32 mixture, with varying composition of R32. When compared to an equivalent R22 system, a compressor power up to 13.7% lower along with an increased COP (up to 15.8%) were found; the refrigerant charge diminishes of up to 27%. On the other hand, the compressor discharge temperature is increased up to 15.4 °C.

Products using R32 are already in the market, such as the Daikin air-to-air heat pump (Daikin Global , 2014).

When comparing an R32 HP to an R410A equivalent, Barve (2010) observed comparable cooling and heating capacities and similar COP. Though, an increase in the discharge pressure and temperature is measured which can be a concern to the compressor's lifetime. An observable positive aspect is the

Table 1. Alternative refrigerants to R410A characteristics

Refrigerant	GWP	Flammability	T critical [°C]	P critical [bar]	Nat. boiling point [°C]
R410A	2088	A1	72.8	48.6	-48.5
R513A	631	A1	101.1	36.8	-29
R450A	601	A1	105.7	40.8	-24
R1234yf	4	A2L	94.7	33.8	−29
R1234ze(E)	7	A2L	109.4	36.3	-19
R32	675	A2L	78.1	57.8	- 52
R152a	124	A2	113.3	45.2	-25
R290	3	A3	96.7	42.5	- 42.2
R1270	2	A3	91.1	45.6	-47.6

decrease in the refrigerant charge that could lead to the usage of a smaller compressor.

2.2.2. Hydrocarbons (HCs)

Their short atmospheric life results in a very low GWP and they represent the best choice when considering their cost. On the other hand, they present problems because of their flammability.

Regarding R290, research is active and products are being brought to the market from companies such as Ait-deutschland (Maul, 2013). In order to lessen its flammability, mixtures can be used. Ki-Jung Park et al. (2009) investigated for example a mixture of R170/R290 (with varying relative composition) in a HP as a substitute for R22. The results showed an increase in the COP of up to 15.4%, despite a decrease in the capacity (up to -7.5%). Other observed improvements of the system are the decrease in the compressor discharge temperature and a reduction in the refrigerant charge. Fan et al. (2013) investigated a R744/R290 mixture, finding an increased COP, along with an increased VHC when compared to an equivalent R22 system.

2.2.3. Hydrofluoroolefins (HFOs)

HFOs have been on the rise since the 2010s, as they present a very low GWP, along with zero ODP. The main disadvantages are the mild flammability and a much higher cost of manufacturing (Bathkar, 2013).

Between the possibilities offered by HFOs, the most promising solutions are offered by R1234yf, R1234ze(E), and their mixtures. The studies conducted by Zhang et al. (2014) obtained an increase of the heat transfer coefficient of R1234yf in the evaporator, and a decrease in the condenser when compared to an R134a equivalent system. To overcome some of the limitations imposed by HFOs, mixtures can be used. Mota-Babiloni et al. (2014) investigated the performance of R450A (R1234ze(E)/R134a mixture, 58/42 in mass percentage) as a transitional replacement of R134a. The outcome is a higher COP, mainly due to a much lower compressor power consumption, and lower discharge temperature, in spite of a lower cooling capacity. Concerning R513A (R134a/R1234yf, 44/56 in mass percentage), the available literature is very limited. Schultz and Kujak (2013) performed drop-in tests in a R134a chiller, obtaining a comparable capacity, but an efficiency reduction of 3-4%.

2.3. Selected refrigerants for further analysis

A low GWP is considered for a preliminary round of elimination, as the alternative refrigerant has to have a GWP at least inferior to R410A (2088). As far as safety is concerned, toxic refrigerants (classified B by the ASHRAE 34) will not be considered as options, whereas flammability will not be a deterring in this phase of the research. The main thermodynamic aspects to be kept into account can involve the operational envelope; considering the pressures (1) the lower limit is set to the atmospheric pressure, as operation under such limit could imply absorption of air form the surroundings and thus possible damage to the system, and (2) the upper limit is set in relation to equipment limitations such as pipes' thickness etc.

With the considerations above, a shortlist of eight refrigerants is elaborated and is illustrated in **Error! Reference source not found.** and Figure 2.

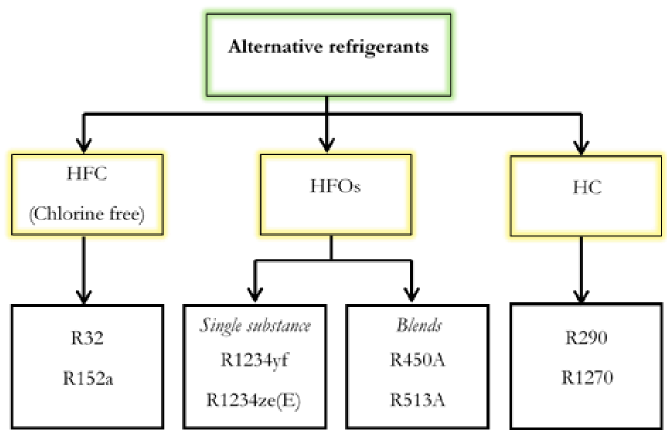

Figure 2. Selected alternatives to R410A

3. ANALYSIS OF THE DEVELOPED MODEL

3.1. Heat pump model

The product considered in the scope of the testing is the Thermia Diplomat Optimum 3G. The modelling of the said heat pump has been conducted in the EES software and validated against experimental data and a second model prepared in IMST-ART, to obtain a basic model as close as possible to the real case. The components of the system are illustrated in Figure 3; the pumps for the secondary fluids are not considered.

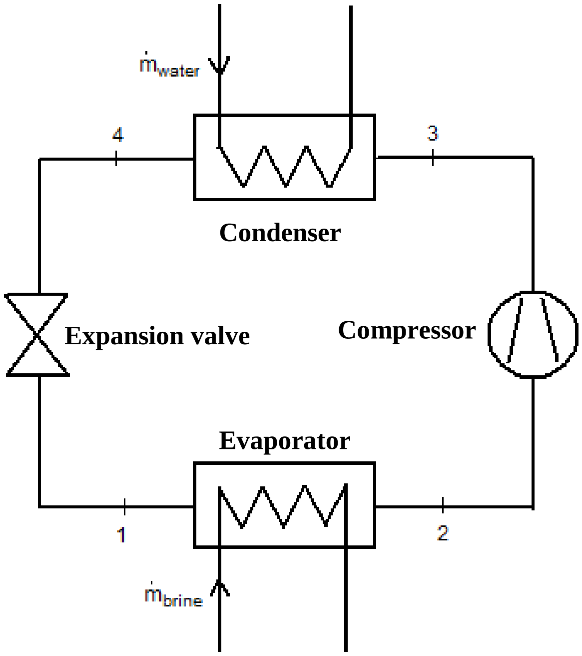

Figure 3. Heat pump system

In Figure 4 the basic vapour compression cycle – followed by the HP considered – is represented in the pressure-enthalpy diagram.

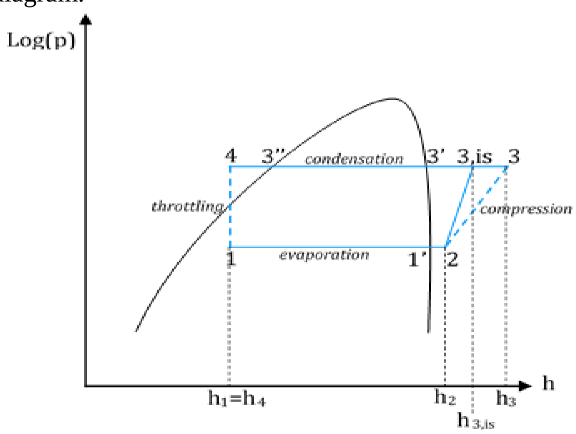

Figure 4. Vapour compression cycle in log(p)-h diagram

The performance of a refrigerating cycle is evaluated through the coefficient of performance (COP). To represent the difference in the purpose of the heat pump and the refrigerating machine, two different COP can be expressed (1), (2). COP_1 is used for HP applications, whereas COP_2 for the refrigerating machine.

$$COP_1 = \dot{Q}_1/\dot{W} \qquad (1)$$

$$COP_2 = \dot{Q}_2/\dot{W} \qquad (2)$$

where $\dot{Q}_1$ is the rejected heat during the condensation process, $\dot{Q}_2$ the absorbed heat during evaporation and W is the required net work input, in kW given by (3), (4), (5).

$$\dot{Q}_1 = \dot{m}(h_3 - h_4) \qquad (3)$$

$$\dot{Q}_2 = \dot{m}(h_2 - h_1) \qquad (4)$$

$$\dot{W} = \dot{m}(h_3 - h_2) \qquad (5)$$

where h_1 and h_2 are the enthalpy at the inlet and outlet of the evaporator, h_3 and h_4 the enthalpies at the inlet and outlet of the condenser, in J/kg and $\dot{m}$ the refrigerant mass flow, in kg/s.

It is to be noted that the considered enthalpy at the exit of the compressor differs from the isentropic one, as the losses in this component cannot be neglected. The isentropic efficiency of the compressor can thus be defined as (6).

$$\eta_{is} = \frac{h_{3,is} - h_2}{h_3 - h_2} \qquad (6)$$

3.1.1. Basic assumptions

A set of experimental data obtained from runs of the real heat pump were available and were used as starting points for the creation of the model. The testing of the heat pump was performed by Thermia, in accordance with the Standard EN 14511. Specifically, the runs were classified by inlet temperature of brine in the evaporator (0, 5, -5 °C) and outlet temperature of water in the condenser (35, 45, 55 °C), creating a combination of nine runs. The brine is a mixture of water and ethyl alcohol at fixed concentration of 29%.
Overall, the input data chosen or determined by the systems' components or operation are as seen in Table *2*.

Table 2. Input data

Amount of superheat	[ºC]
Amount of subcooling	[ºC]
Inlet temperature (brine)	[ºC]
Outlet temperature (water)	[ºC]
Mass flow (brine/water)	[kg/s]
Specific heat (brine/water)	[kJ/kg-°C]

3.1.2. Evaporator

The evaporator used in the Diplomat Optimum 3G is the SWEP QA80.
In order to define the performance of the evaporator, the UA value needs to be determined. As a first approach for the modelling, the UAs have been inserted as inputs. The inserted values could be obtained from a second, separate model, after defining all the thermodynamic properties of the major points of the cycle. As the values for temperatures and pressure were available as experimental outputs, the enthalpies could be determined. Data for mass flows and specific heat were also present, thus allowing the calculation of the UAs for evaporation and superheat. The calculated values were used as input in the main model of the heat pump. The system solved in EES is illustrated by equations (7), (8), and (9)- here stated in a general way and in the software applied to both evaporation and superheating.

$$\dot{Q} = \dot{m}_{ref} * \dot{\Delta h}_{ref} \quad (7)$$

$$\dot{Q} = \dot{m}_{brine} * \dot{cp}_{brine} * \Delta T_{brine} \quad (8)$$

$$\dot{Q} = LMTD * UA \quad (9)$$

Where $\dot{Q}$ is the heat in kW, $\dot{m}_{ref}$ the mass flow of refrigerant in kg/s, Δh_{ref} the enthalpy difference the refrigerant undergoes in the considered process in kJ/kg, $\dot{m}_{brine}$ the mass flow of brine in kg/s, cp_{brine} the specific heat of the brine in kJ/kg-°C, ΔT_{brine} the temperature difference experienced by the brine in the considered process, LMTD the logarithmic mean temperature difference in °C.

In a second approach, two correlations for the UAs have been obtained through a regression analysis (10), (11), applied to the UA values obtained from the model. . It is important to point out that due to the scarcity of available experimental data (only 9 set of points), this procedure essentially consists of an interpolation, rather than a statistical study (hence the neglect of presentation of statistical characteristics such as standard deviation and R2).

$$UA_{SH} = 22.2465 - 253.989 * k_{1'} - 298.449 * k_2 - 1511947 * \mu_{1'} - 284923 * \mu_2 - 2.2384 * \rho_{1'} - 0.4587 * \rho_2 + 0.10934 * P_{evap} \quad (10)$$

$$UA_{evap} = 323.01 + 19158.8042 * k_{1'} - 49773038.92 * \mu_{1'} + 88.612 * \dot{m}_{ref} + 0.4796 * T_{evap} \quad (11)$$

Where $k_{1'}$ and k_2 are the thermal conductivity of the refrigerant in points 1' and 2 in W/m-°C, $\mu_{1'}$ and μ_2 are the dynamic viscosity of the refrigerant in points 1' and 2 in Pa/s, $\rho_{1'}$ and ρ_2 are the density of the refrigerant in points 1' and 2 in kg/m^3, P_{evap} [kPa] and T_{evap} [°C] are the evaporation pressure and temperature.

3.1.3. Condenser

The condenser used is the H62L-CX, micro-plate heat exchanger.

As done in the evaporator case, the UAs have been calculated with two successive approaches. Firstly, they were calculated from experimental data in a separate model by (7), (8), (9) - applied to the three stages occurring in the condenser - and then used as input in the main model. Secondly, these UA values were used in a regression analysis which issued the following correlations (12), (13), (14).

$$UA_{des} = -2.6208 + 2.4611 * \dot{m}_{ref} + 287997 * \mu_{3'} - 0.0372 * T_{cond} \quad (12)$$

$$UA_{cond} = 124.1569 - 1092.0017 * k_{3''} - 69.6615 * \dot{m}_{ref} - 0.7917 * T_{cond} \quad (13)$$

(14)

$$UA_{sub} = -450.716 + 2941.455 * k_{3''} - 1005.03 * k_4 + 0.027 * P_{cond} + 0.1058 * \rho_{3''} - 30.3314 * \dot{m}_{ref} + 1.5201 * T_{cond}$$

Where $k_{3''}$ and k_4 are the thermal conductivity of the refrigerant in points 3'' and 4 in W/m-°C, $\mu_{3'}$ is the dynamic viscosity of the refrigerant in point 3' in Pa/s, $\rho_{3''}$ is the density of the refrigerant in points 3'' in kg/m^3, P_{cond} [kPa] and T_{cond} [°C] are the condensation pressure and temperature.

3.1.4. Compressor

The compressor in the Diplomat Optimum 3G is the Emerson scroll compressor ZH09K1P-TFM-524.

The correlation for the isentropic efficiency (15) has been found by fitting of data obtained on the Emerson online software (Emerson climate technologies, 2015). The values obtained with (15) were compared with experimental data available, and a relative difference between the two set of values calculated.

$$\eta_{is} = 0.0024 * P_{ratio}^{\ 3} - 0.0363 * P_{ratio}^{\ 2} + 0.1377 * P_{ratio} + 0.5531 \quad (15)$$

The calculated difference allowed to obtain values closer to the real case, from the manufacturer's data. It is assumed to be applicable to the alternative refrigerants as well. Hence, the provided correlations for R1234yf, R290, R1270 and R152a – obtained for other conditions - could be adjusted to be applied to the studied HP case (Figure 5).

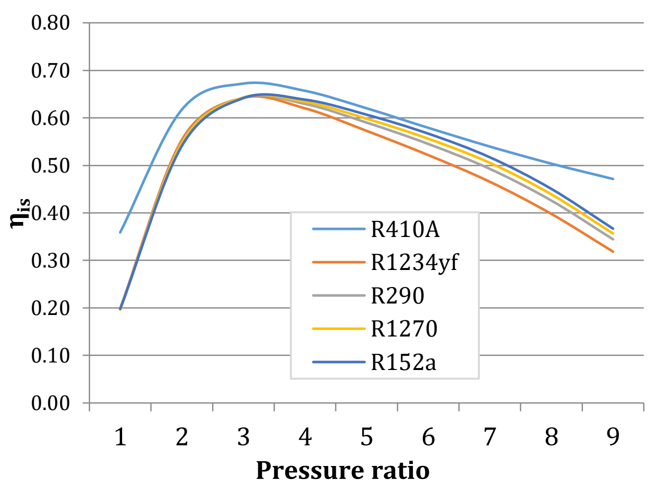

Figure 5. Isentropic efficiency of compressor for some alternative refrigerants

For the refrigerants for which the correlation was not available to start with, assumptions were made. Specifically, it was assumed that R32 could operate with the same compressor efficiencies as R410A, due to its similarities in temperature and pressure levels. Concerning R1234yf, it was assumed to operate with the same correlation as R1234ze(E). No assumptions regarding the compressor could be made for R450A and R513A, because of the limited access to data. For this reason, the analysis of these two refrigerants is left for the future steps of the work.

3.1.5. Expansion valve

The expansion valve present in the considered heat pump is a Grundfos Magna 25-100.

The throttling process is considered to be isenthalpic, and thus $h_1=h_4$.

3.2. SCOP calculation

The calculation of the SCOP has been performed following the Standard BS EN 14825:2012. Specifically, the model was structured following the Standard directives in three main aspects:

(1) Characterisation of heating season: number of hours at which every temperature occurs and heating demand

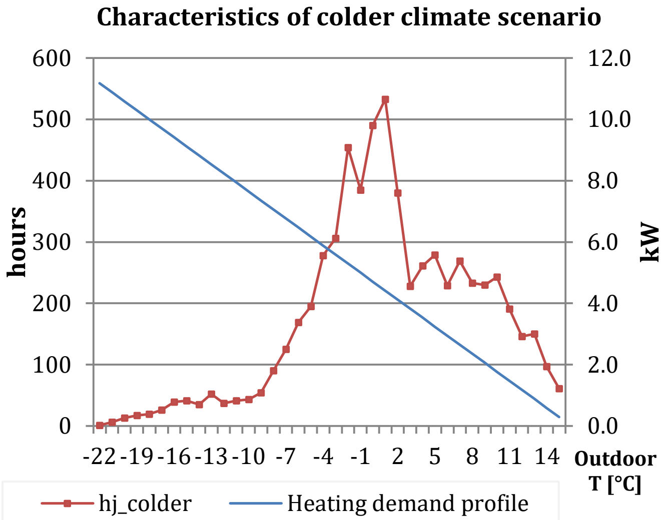

Figure 6. Temperature profile in colder climate and heating demand profile

(2) Requirements for secondary fluid (brine and water) temperatures as in Table 3.

Table 3. Requirement for secondary fluids' temperatures

		T_water_out [°C]	
Outdoor T[°C]	**T_brine_in [°C]**	**Floor heating**	**Radiators**
-22	0	35	55
-7	0	30	44
2	0	27	37
7	0	25	32
12	0	24	28

(3) SCOP calculation equation

$$SCOP_{ON} = \frac{\sum_{j=1}^{n} h_j * Ph(T_j)}{\sum_{j=1}^{n} h_j * (\frac{Ph(T_j)-elbu(T_j)}{COP_{partial}(T_j)} + elbu(T_j))} \quad (16)$$

3.3. TEWI calculation

The TEWI factor is calculated according to (17).

$$TEWI = (GWP * L_{annual} * N) + GWP * m * (1-\alpha_{recovery}) + \rightarrow \text{Direct} \quad (N * E_{annual} * \beta) \rightarrow \text{Indirect} \quad (17)$$

where GWP is the global warming potential of the refrigerant (kg_{CO2}/kg_{ref}), L_{annual} the leakage rate in the system (kg_{ref}/year), m is the refrigerant charge (kg_{ref}), N the life of the system (years), $\alpha_{recovery}$ the recycling factor, E_{annual} the energy consumption per year (kWh/year), β the CO_2 emission factor (kg_{CO2}/kWh) (AIRAH, 2012) (Mohanraj, 2011).

The values of the input parameters used are summarised in Table 4.

Table 4. Input parameters for TEWI calculation

Input data	
Annual leakage rate	2%
Lifetime [years]	15
β [kg CO_2/kWh_e]	0.023
α recovery	0.7
Charge [kg]	2.3
Annual heating demand [kWh]	27547.35

4. RESULTS AND DISCUSSION

4.1. Coefficient of Performance

The general trend of the COP (Figure 7) is common to every refrigerant, and consists of its increase with increasing outdoor temperature, thus increased heating capacity. The rates of growth though, are different within the options thus making the identification of their relationships not straightforward.

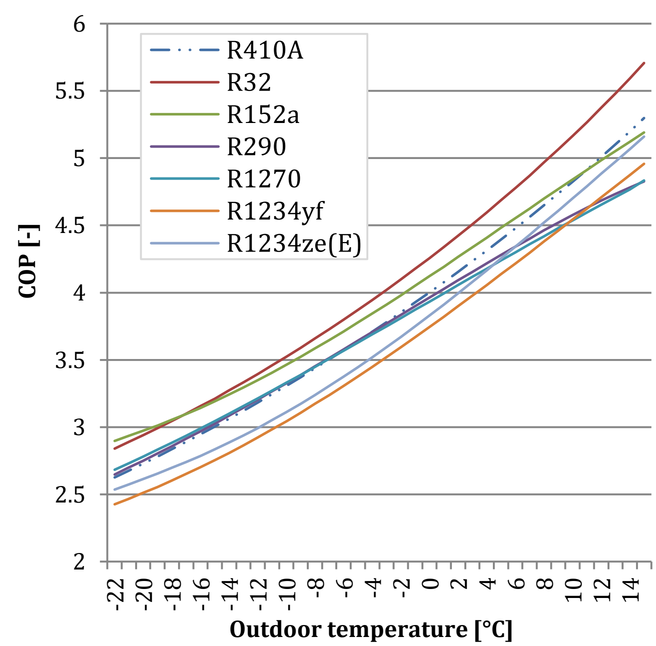

Figure 7. COP vs. outdoor temperature (radiator heating)

Generally, it can be remarked that R32 outperforms every other substance from -17 °C. It reaches the maximum increases compared to R410 at the lowest and highest temperatures of the range, respectively of 8.19% and 7.72%. The divergence is reduced in the central values of the range. Below -17 °C, the best performance is offered by R152a, whereas above that, as mentioned, by R32. Up to -7 °C, the baseline R410A is also

outperformed by both the hydrocarbons, R290 (up to -5 °C) and R1270. In the case of propane, the relative difference is below 1%. R1270 presents a slightly higher COP along all the range, and reaches an increase up to 2.2% compared to R410A at the lowest temperature evaluated. Along the range they maintain a very close growth trend. Concerning the HFOs, R1234yf and R1234ze(E), it appears that they present the lowest COPs overall. The latter maintains along the range a higher COP, with a relative difference with R1234yf almost constant.
At the highest temperatures, propane and propylene experience a slowdown in their growth rate, thus being outperformed by every other refrigerant along the range. Specifically, R1234ze(E) outperforms R1270 and R290 respectively at 5 °C and 6°C, and R1234yf and 11 °C and 12 °C. They register the highest decrease compared to R410A at the highest temperature of the interval, equal to -8.89% for R290 and -8.76% for R1270. At 12 °C, R410A outperforms R152a, thus offering the best performance at the highest temperatures, excluding R32.

4.2. Seasonal Coefficient of Performance (SCOP)

When evaluating the COPs of the refrigerants, it was evident that an overall ranking of performance was difficult to elaborate, as their relationships varied with varying conditions. The SCOP is then a powerful tool that allows the assessing of the performance along the whole heating season.
In Figure 8, the results, obtained from the method developed in section 3.2, are presented for both the cases of radiator and floor heating.

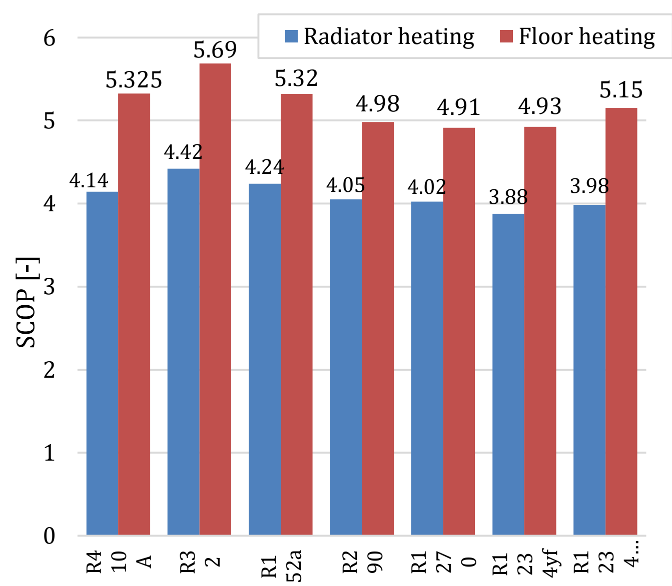

Figure 8. SCOP by refrigerant over the heating season for colder climate scenario (radiator and floor heating)

As expected, R32 presents the highest seasonal performance (+6.46% compared to R410A), as it maintains the highest COP in almost all the range of temperatures comprising the heating season. It is followed by R152a (+2.28%), both outperforming R410A. The lowest SCOP is obtained by R1234yf, with a total reduction compared to R410A of -6.4%, followed by R1234ze(E) with -3.87%. The hydrocarbons entail a smaller reduction, with -2.9% for propylene and -2.2% for propane.

4.3. Total Equivalent Warming Impact (TEWI)

In Figure 9 the values calculated for Sweden in the case of radiator heating are shown.
It is evident that the Hydrocarbons, R290 and R1270, along with R1234yf and R1234ze(E) present an almost inexistent direct effect, due to the low GWP. The indirect effect, instead, doesn't show a major variation between the options.
The lowest reduction is obtained by R32 (40.8%). The HCs offer overall the lowest TEWI, thus the best environmental performance with reductions of 54.2% and 54.6% for R1270 and R290 respectively. They are closely followed by R1234yf and R1234ze(E) that register respectively -52.5% and -53.7%. Due to the good SCOP and the low GWP, R152a also entails a good reduction, equal to 53.6%.

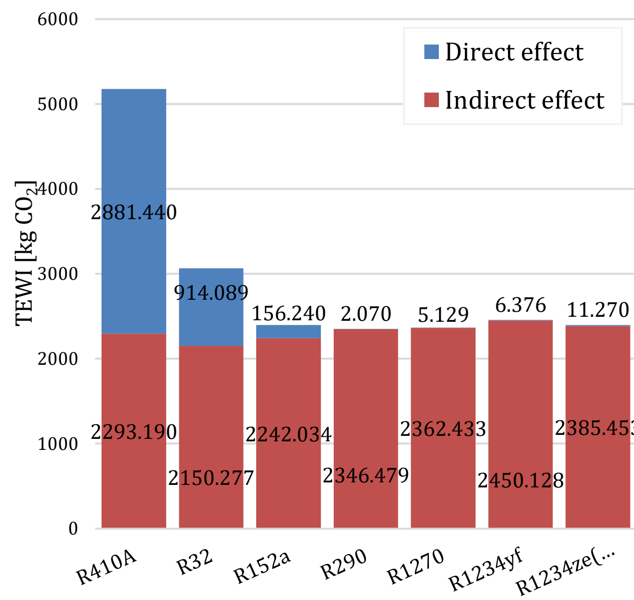

Figure 9. TEWI - Direct and indirect effect (radiator heating)

4.4. Summary for every refrigerant

4.4.1. R32

Overall, at the evaluated conditions R32 offers a better performance than R410A in terms of COP, at every outdoor temperature. Moreover, the VHC is higher, thus entailing a possible smaller charge to be used in the system, and consequently a smaller size of compressor.
On the other hand, the TEWI factor is subjected to the lowest reduction (when assuming a constant charge). Though, the TEWI could be reduced by using a smaller quantity of refrigerant (possible thanks to higher VHC) and thus having a smaller direct effect. Furthermore, at certain conditions the discharge temperature can reach level above the safety threshold set for the protection of the compressor.
Flammability issues must also be kept into account, as R32 belongs to the A2L category. The system could then be required to comply with restrictions in terms of amount of charge and location of the heat pump.

4.4.2. R152a

The use of R152a results in an increase of the SCOP, when considering radiator heating applications, and a decrease in the TEWI bigger than in the case of R32. At the very low temperatures the discharge temperature is reduced compared to R410A, and the COP is increased. At the highest temperatures, instead, the refrigerant at the exit of the compressor reaches higher temperatures than R410A, though remaining below the safety limit. The COP worsen. Furthermore, the VHC is lower, thus implying the necessity of using more charge to obtain the same heating effect.

R152a is also flammable, thus requiring restrictions as in the previous case.

4.4.3. R290

Propane offers the overall lowest TEWI between the alternatives. This is mainly due to its much lower GWP. The SCOP is subjected to a reduction thus implying a slightly higher indirect effect. The discharge temperatures are significantly lower than with R410A, which could benefit the compressor's lifetime. The considerably lower VHC would cause the use of higher charge if the heating capacity wants to be preserved.

At the lowest temperature this hydrocarbon has a higher COP than R410A, but undergoes a decline along the temperature range; at the highest temperatures, it obtains the lowest COP.

Though, R290 is highly flammable and thus subjected to strict regulations in terms of charge quantity and location of the heat pump.

4.4.4. R1270

The considerations on propylene are very similar to the ones made for propane. The reduction in COP is comparable. Though, the VHC is higher. Between the two options therefore, R1270 can be seen as a better choice in terms of producible heat, or, if constant heating capacity, in terms of smaller charge needed. The discharge temperatures reached are generally slightly higher than with R290, but still well below the safety limit and R410A values.

This refrigerant is also highly flammable, and concerns can be raised on its chemical stability at high temperatures. It could then be seen as a better solution for restricted low temperature applications.

4.4.5. R1234yf

The use of R1234yf entails a reduction of all the parameters involved. The reduction in TEWI is remarkable, though as its SCOP is the lowest between the evaluated alternatives, its indirect effect is the highest one.

The discharge temperature is also decreased when compared to R410A, bringing benefits to the compressor's lifetime.

The VHC is very low, so the same charge considerations seen previously can be applied in this case as well.

In addition, R1234yf belongs to the flammability category A2L, that in many countries' regulations is still considered equal to A2, thus requiring restrictions more severe than necessary.

Being a relatively new substance, moreover, its cost is still quite high.

4.4.6. R1234ze(E)

R1234ze(E) has very similar performance as the previous R1234yf. It can be noticed that its VHC is even lower, thus putting more strain on the size of the system. Its COP is generally higher than the other HFO, and therefore the final TEWI is comparable to the HCs' result. Regarding the flammability, the same comment as for R1234yf can be made.

5. CONCLUSIONS

When researching alternative refrigerants to the current R410A in use in the domestic heat pump considered, 8 options were identified that could entail lower CO_2 emissions, and thus comply with the directions the refrigeration industry must take in accordance to the F-gas regulation, recently implemented.

Of these eight options, six were analysed in the model created, namely R32, R152a, R290, R1270, R1234yf and R1234ze(E).

The most relevant results considered in the scope of this study were obtained for both radiator and floor types of heating systems and consist of (1) VHC between -22 °C and 16 °C, (2) COP between -22 °C and 16 °C, (3) discharge temperature between -22 °C and 16 °C, (4) SCOP, for the overall heating season, (5) TEWI factor.

It is important to understand that the results obtained are highly dependent on the area for which they have been calculated, namely Sweden (colder climate scenario and low CO_2 emission factor of the national energy mix). Therefore, the conclusion drawn from this study are only valid in the mentioned specific area.

The performance of the HP, as seen through the COP graph (Figure 7) with the alternative refrigerants was variable. At the lowest temperatures, R410A presents a lower COP than R32, R152a and both the HCs (up to -7 °C). With the increasing outdoor temperature, the baseline performance improves but remains poorer than R32 along the whole temperature range. The worst COPs were shown by the HFOs.

The overall highest SCOP (Figure 8) was thus given by R32, outperforming R410A in both radiator and floor heating cases. R152a also presented positive outcome, having a higher SCOP than R410A in radiator heating case, and an almost identical one in the floor heating case. The other options entailed lower seasonal performances, with the worst result obtained by the HFOs (and in particular R1234yf).

When evaluating the overall environmental impact of the refrigerants, through the TEWI factor (Figure 9), a general reduction of CO_2 equivalent emissions is calculated. Whereas the indirect effect is similar for every option considered – slightly increasing with decreasing SCOP – the direct effect is drastically reduced with the decrease of the GWP. In fact, both the HCs and the HFOs obtain a remarkable reduction of the TEWI factor, in accordance to their extremely low GWP. The lowest emissions' reduction is obtained by R32, which, in spite of an improved performance in the machine, still has a considerable GWP.

Moreover, the reaching by R32 of discharge temperatures above the safety limit cannot be neglected, as it could endanger the compressor's functioning. On the other hand, this drawback could be mitigated by the use of a smaller refrigerant charge

The interpretation of these results and the selection of a most favourable option is then closely related with the direction that the manufacturer wants to take. If the preference is to keep a certain heating capacity, R32 offers the possibility of using smaller charge and thus smaller systems.

If the aim is to reduce the CO_2 as much as possible, the HFOs and the HCs represent a better option in a country with a high share of renewable, clean energies and thus a small CO_2 emission factor for the production of electricity.

It is also paramount to keep into consideration the different flammability levels of the refrigerants. None of the options evaluated in this study are non-flammable, and thus they all require some restrictions in their use, e.g. charge limitations, location of the installation restrictions. HCs are the most subject to limitations, as they belong to the A3 – high flammability – category. R32, R1234yf and R1234ze(E) belong to the A2L category. As of today, regulations do not differentiate between A2L and A2 levels. This situation could be modified in the near future, representing an advantage for these three refrigerants, as it would mean less strict limitations in use.

In this concern, the two remaining options that could not be evaluated at this stage – R450A and R513A – present the advantage over the other refrigerants as they are not flammable.

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